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COLLEGE OF ENGINEERING
DEPARTMENT OF NAVAL ARCHITECTURE AND MARINE ENGINEERING

PROCEEDINGS OF CONFERENCE

ON

Advanced Marine Engineering Concepts For Increased Reliability

Conference Director
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FOREWORD

This volume contains the papers and discussion from the Conference on *Advanced Marine Engineering Concepts for Increased Reliability* held at The University of Michigan. Since only one written discussion was submitted, the bulk of the discussion in this volume was transcribed from tape recordings made during the Conference. All those who spoke were given an opportunity to edit their remarks. In the few instances in which the participants did not do this, the conference staff corrected the obvious errors, but accepts no responsibility for sins of commission or omission.

The Conference dealt with a subject which is new to most of the marine engineering fraternity. There is little doubt that we can learn much from the aircraft industry and from other technologies concerning reliability analysis. It is hoped that the Conference and its Proceedings will stimulate an interest in the subject eventually leading to the application of suitable reliability techniques to marine problems.

The original idea for the Conference came from Commander E. P. Cochran of the Office of Naval Research, which, together with the Bureau of Ships, Department of the Navy, provided the necessary funds. Our thanks go to Lt. Commander A. J. Coyle, who replaced Commander Cochran as the contracting officer and who provided excellent support for the program from Washington. Finally, I would like to thank Miriam Levin, Ray Reilly, and Dick Voelker for their able assistance.

G. L. West, Jr.
Ann Arbor, Michigan
May 24, 1963

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INTRODUCTORY TALK

C. M. FIXMAN
Maritime Administration

I feel singularly honored by being asked to open this Seminar in Advanced Marine Engineering Concepts for Increased Reliability sponsored by the Bureau of Ships, The Office of Naval Research and the Department of Naval Architecture and Marine Engineering of the University of Michigan.

An opening address perhaps should be humorous, general in content and should lead to the papers to follow. Once given the floor, however, in a subject as important as this, I feel I must offer a few statistics and make some points.

Engineering for reliability is not new. We owe our lives to the reliability and safety devices designed and incorporated into the common things that surround us and make up our way of life.

Primarily as a result of lack of reliability in some of the newer fields of engineering, a body of theory and practice has been developed, and we in one of the oldest of the fields of engineering, the field of water transportation are learning new things.

You may consider the importance of reliability engineering as one of competitive economics, as one of safety of life and property or as one of the natural security.

I feel that reliability engineering will be a vital force in maintaining our leadership in the world by reducing our costs in redundant equipments, improving the quality of our product and reducing the man-hours of maintenance required to keep our equipment on the line.

Let us get back to transportation industry and consider some recent accounts relating to reliability.

The Washington Post of November 2, 1962, reported that the Federal Aviation Authority had collected penalties from two trunk airlines for unsatisfactory maintenance of jet aircraft. In one case an airplane had to shut down an engine in flight because of mis-matched engine parts. In the other case a plane had to return to an airport because the plane's landing gear could not be retracted — a valve had been installed in the hydraulic system backwards. These incidents sound familiar to us in the marine industry.

In another report the Civil Aeronautics Board blamed the 1961 crash of an Electra at Chicago's O'Hare Airport and the resulting loss of 37 lives on mechanics failure to securely tighten and safety wire two cables in the aileron hydraulic boost system.

On February 2nd the SS MARINE SULPHUR QUEEN a T-2 tanker converted to carry 15,000 tons of molten sulphur disappeared on a coastal voyage with 39 men aboard. She had been employed in this trade for some time having delivered over 950,000 tons of sulphur before her disappearance. Did a mechanic forget to do something on her before she left on her last voyage?

Lloyd's Register shows 189 ships of over 100 gross tons, totaling a loss of 470,000 gross tons in 1961. The United States losses were 9 ships totaling over 40 gross tons.

Recently a brand new merchant ship could not complete its trials because a boiler tube ruptured. When she returned to the building yard, a cold chisel was found lodged in a boiler tube. It prevented circulation and caused the subsequent rupture of the tube.

In the cases of the airplanes, penalties were exacted by a Government agency in addition to the loss of lives, loss of equipment and loss of revenue. In the marine cases there was no penalty other than the loss of lives, ships and revenue.

How many of the marine losses were due to the design of the equipment? How many of them were due to improper maintenance or operation?

The Trial Board of the Maritime Administration when asked to furnish information on repetitive failures on recently delivered merchant vessels reports as follows:

> The radar equipment on nearly all ships requires constant maintenance. There is scarcely a trip where a service call on this equipment is not required. Loran and radio equipment is nearly as bad.
>
> Operators are becoming more and more aggravated by the lack of reliability in electronics equipment.
>
> Although hydraulic hatch covers seem to be an answer to cargo handling, they are a constant headache aboard ship. The hydraulic systems leak and the covers are not tight. The covers also require excessive maintenance for physical damage because it seems nothing can be made strong enough to withstand the beating inflicted by stevedores.
>
> More and more operators are attempting to carry premium liquid cargoes in bulk. In an effort to minimize tank cleaning costs, they have gone to special coatings and stainless steel claddings. The latter proved to have a serious shortcoming in that salt water ballast would cause corrosion if carried for extended periods of time. Most of the coatings have not been very successful because of the extreme care required for application and the fact that they generally are not suited for all cargoes that may be carried.
>
> Recently, there have been several bearing failures on main turbines. It appears that these were not due to the usual "dirt in the oil" but may have been because of the additives being used.
>
> The Naval Architects invaribly design a low stack for esthetics — and they invaribly cause smoke and soot on the decks and in the house.
>
> Stainless steel wheels were installed on the Delta Line ships out of Avondale. Within the six month guarantee period, severe cracking of the blades was detected. In addition, several blades were lost due to breakage of the bolts in the hub. The wheels have been redesigned and the new wheels are currently being put on. Other wheels of the same material made by the same company have proven successful so far on several Lykes ships.
>
> This propeller example is particularly interesting since a cursory examination of the records of 29 American single crew cargo ships covering over 37,000 ship days of steaming since 1943, showed an availability considering casualties from all causes for the conventional solid cast manganese bronze propeller and tail shaft combination in the order of 0.997 or higher. The stainless steel propellers are an advance concept in which the reliability engineering analysis may have been questionable.

Our goal is to obtain reliable operation. Note that this goes beyond just having reliable equipment. The examples mentioned above prove this. Reliable equipment can be obtained by proper and careful design, suitable materials, quality control, proper construction, and careful testing. These are not enough. Proper operation including maintenance and repair are essential. Only within limits can we design to reduce the requirements for caution in operation. We cannot engineer equipment and systems without knowledge of the eventual operation of the equipment. Our job is not completed until we know the crew is trained to operate in normal and abnormal situations.

Once acceptance trials have been completed we tend to turn the job over to the contract people to settle the money problems. The ship is in the hands of the operators and we get vague high level reports on a few minor problems, but mostly that she is a fine ship. So we go along designing another ship witnout the feed-back of information we need to improve the next ship.

During this meeting we can be sure that we will find that we have insufficient data on maintenance to be able to assign good numbers to marine reliability factors. As engineers we need the facts so that we can improve our equipment and systems. Valid reliability numbers can be used to reduce our costs.

In the marine field, our units operate independently and seagoing people are by necessity forced to be self-sufficient. This same self-sufficiency has made it difficult to obtain the data we require. To the people operating our ships the data we desire is just some more paper work to keep the landlubbers happy. They want to devote their time to operating. Further, I am not sure just what data we need. In a recent case we attempted, with the cooperation of the Society of Naval Architects and Marine Engineers and several ship operators, to obtain some data on the number of man hours being used by ships crews in various maintenance operations. We have some numbers, but preliminary analysis does not fit them into a normal statistical pattern, and therefore, their validity can be questioned. Having operating forces take such data is expensive and provides information of questionable validity. Yet, before we can get the feed-back information so that we can improve our practice, we must have the data.

Before we can get the information we need, we must have a good knowledge of how we want to use it, then the costs of obtaining the information will be as low as possible and we will be able to check its validity.

Often, more can be learned from a failure than from successful operation. Equipment which operates satisfactorily may be overdesigned, may be on the point of incipient failure, or may be such that a little effort keeps it operating. When equipment actually fails, however, we know something is wrong. An analysis of the failure gives us an opportunity to strengthen the weak point, that is, if we will take advantage of the opportunity. We all have a tendency to make the repair and get the equipment going without really trying to determine what caused the casualty. There are too many reports of failure due to "crystallization," wiped bearings in machines that have operated satisfactorily for years, dirt in hydraulic systems and other common excuses for failure. Since, in the case of a casualty, the equipment is already down and time will have to be spent in repairing it, we should have competent and thorough analyses of the factors influencing the failure so that we can determine its cause and feed back the information into our designs. Some laboratory work on the analysis of deposits found, some calculations of stresses at points of failure, good photographs and reports would add little to the cost of the repair. Further, this might give us the information we need to advance the concept as well as prevent a repetition of the error.

Now may I summarize the general points that I have tried to make in this introduction.

First, reliability goes beyond having reliable equipment. It includes proper operation and maintenance.

Second, we need feed-back of information from the operating forces if we are to advance our concepts.

Thirdly, we need qualified analyses of casualties so that we can determine the cause of failures. When we get the feed-back from the operating forces we will be in a position to use the methods of the reliability engineers to improve our ships.

Finally, where do we go from here? Meetings such as this will help to determine the path we should follow.

I feel we have the knowledge of the theory and practice of engineering reliability and our rate of progress toward Advanced Marine Engineering concepts for increased reliability will be determined by the effort and funds we are willing to expend.

WORK STUDY IN SHIP DESIGN

CAPTAIN H. A. KAUFFMAN, USN
Bureau of Ships Work Study Coordinator

The U. S. Navy has for some time been confronted with three closely connected problems. The first and foremost of these is the continual growth of ships' complements since World War II. The increase is caused by the introduction of new, sophisticated, rapid response ships' systems which have resulted in an increasing requirement for more operational and maintenance manpower. The second problem is the increase in space required by the new systems and the increased complement, causing ship growth such that a present day DE is bigger than some World War II DD's. The new systems have undoubtedly improved the ships' "eyes", "ears" and "kill" capabilities but at the expense of bigger ships and greater cost in men and money. The third problem is one of maintenance. Sophisticated new systems have brought difficult maintenance problems and unreliability. Our methods of dealing with the maintenance problems have not advanced as rapidly as technology has introduced more complexity. We need more complete preventive maintenance data and a better Maintenance Management System to plan and control work in ships.

The Chief of the Bureau of Ships, Rear Admiral R. K. James, U. S. Navy, was very conscious of all of these problems when he made a European tour two years ago. During the tour he reviewed the efforts of the British Navy in applying Work Study in their ships at sea. The results, which he saw, showed that there were often better methods for men to do their work in ships. In some cases minor alterations to the ships and their systems were necessary to do the work with fewer men. The word "alterations" sparked his desire to do this work during the design stage thus furthering his "Dollar Stretch" program which is: To get more and better ships for the same dollars. He saw in Work Study the tools the designer needed to integrate men and equipment. He believed that the ship design engineers could make good use of all these techniques in designing all of the systems in a ship for men to operate. These tools would encourage the designer to give more consideration to the fact that men are integral links in all systems. Having the foresight to see the possibilities, Admiral James started the Bureau of Ships Design Work Study Program in May 1961.

Before describing the Program a question must be answered: What is Work Study? Most people associate the name with Time and Motion Study — Time Study (originated by Taylor) is just one of the many modern techniques for measuring work content so it has been renamed Work Measurement — Motion Study (originated by Gilbreth) is the study of human movements to find better ways for people to do work. The new name of Method Study embraces many more techniques which are not confined to people. Method Study finds better ways of doing work by people and machines and organizations. The scale is much broader than Motion Study though some of the techniques are similar. The two techniques: Method Study and Work Measurement, combine to be called "Work Study." Work Study is nothing more than a logical, systematic, fact finding method of determining what needs to be done, how it should be done, and who should do it. Determining the facts, however, is not always an easy task.

The Bureau first had to take the tools of Work Study as developed by industry and the British Navy's Fleet Work Study organization and adapt them for use in ship design. In accomplishing this task, it was recognized from the start that Work Study is no substitute for the technical ability and ingenuity of the design engineers. Rather it is a tool which helps the designer to evaluate and question his work in a logical and systematic fashion. It was, therefore, necessary to establish a Design Work Study School for training engineers. At this school the men are taught to use such recording techniques as Flow Process Charting, Multiple Activity Charting and Operational Sequence Diagramming; the new and searching approach to Critical Examination; and Preventive Maintenance Management. Through all of these techniques and others, the facts necessary to design ships that require the minimum number of men to operate and maintain them are established. This, then, is our task:

> "To design ships that require the minimum number of men to operate and maintain them without reducing the ships' fighting effectiveness."

A total of 230 men have been trained to date: Engineers from the Bureau of Ships, Bureau of Weapons, Bureau of Naval Personnel, OPNAV and private companies.

To describe Work Study in Ship Design, it can best be presented through the use of Figures 1 and 2. For any ship design, the design engineers must make a three prong attack on the ship, i.e., Operating-Maintenance-Management, as shown in Fig. 1. Starting with Operating, the engineers first examine all systems within the ship to establish the *need* for each system. In some cases it has been found that a system, subsystem or component is not needed since the *cause or reason* for having that particular system, subsystem or component is changed or eliminated. Having established the true *need* for an item, then the designer must provide the simplest, most reliable hardware to satisfy that true need. In the latter analysis, reliability of components and systems should be considered together with the desires to have the minimum number of components in a system since, generally, this leads the way to improved reliability and minimum preventive maintenance.

Having established the *hardware* system to best satisfy the *need,* the designer again critically examines the system to determine if that hardware system is, in fact, the minimum manning system under *all* operating conditions. If it is also the minimum manning system, then his basic design work on that system is complete. If it is not, then the designer must establish the hardware system that will provide for minimum manning. At this point it can be determined whether automation can, in fact, save operators under specific or all operating conditions.

After all systems in the ship have been examined as described above, then it is possible to establish the operational manning requirements under all operating conditions as is shown in the third block under "Operating" on Fig. 1. At this point in the ship design, the details of the systems are passed to the Bureau component engineers for their comments and ultimate approval and for their estimate of the preventive maintenance requirements of all components in the ship; see Fig. 1 under "Maintenance." These component engineers must not only specify what work has to be done and how often the work has to be done by a specific skill level, but must estimate how long it takes to do the work. Thus, they produce a quantitative measure of the preventive maintenance requirements for the ship under all conditions. Armed with the above operational and preventive maintenance manning data under all operating conditions, it is possible to chart these partial results as shown in Fig. 2 for each specialty. Only three operational conditions are shown in this figure, i.e., Battle, Cruising and Port; but there are actually several more conditions that must actually be charted. To draw Fig. 2, the designed manhours required to operate the ship during a 24 hour period is blocked-in under the three shown conditions. Next the preventive maintenance manhours are superimposed on the operating manhours, i.e., in this example it is assumed that no preventive maintenance manhours are performed in the Battle Conditions but do exist in the Cruising and Port Conditions.

Corrective maintenance is defined as work required by the ships' force to place defective equipment into operating condition. Other work is defined as authorized

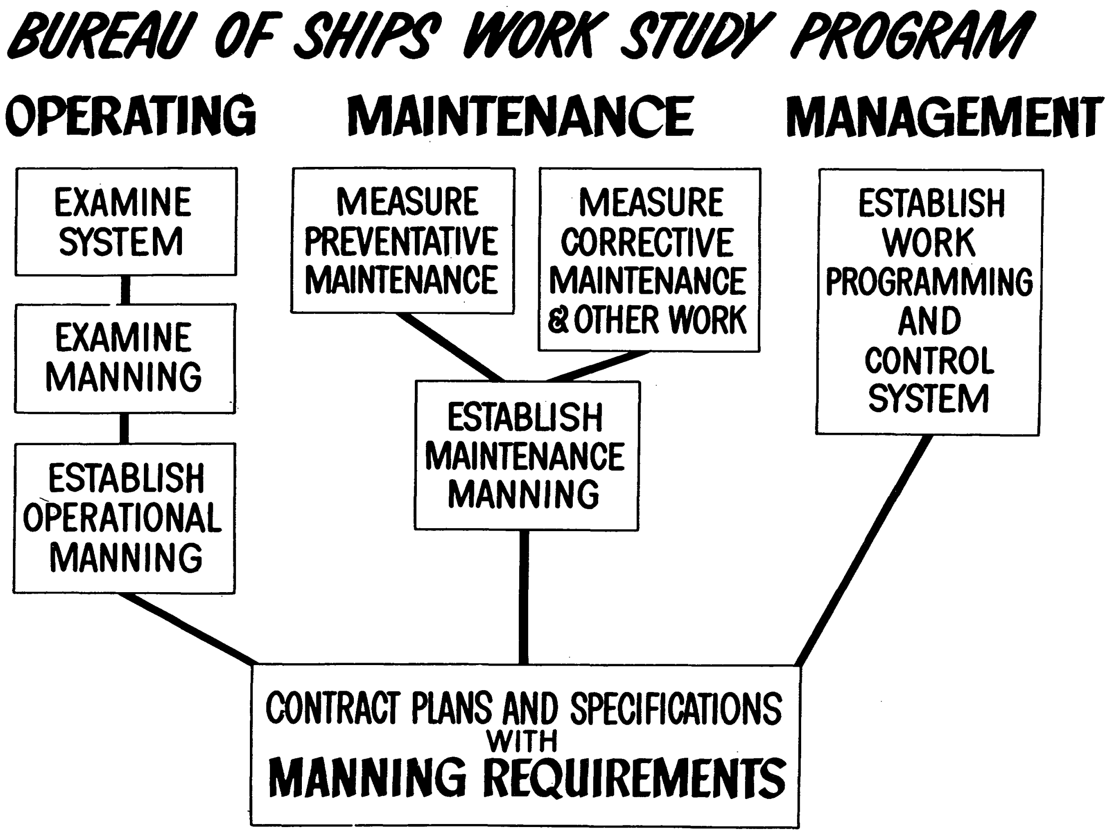


Figure 1.

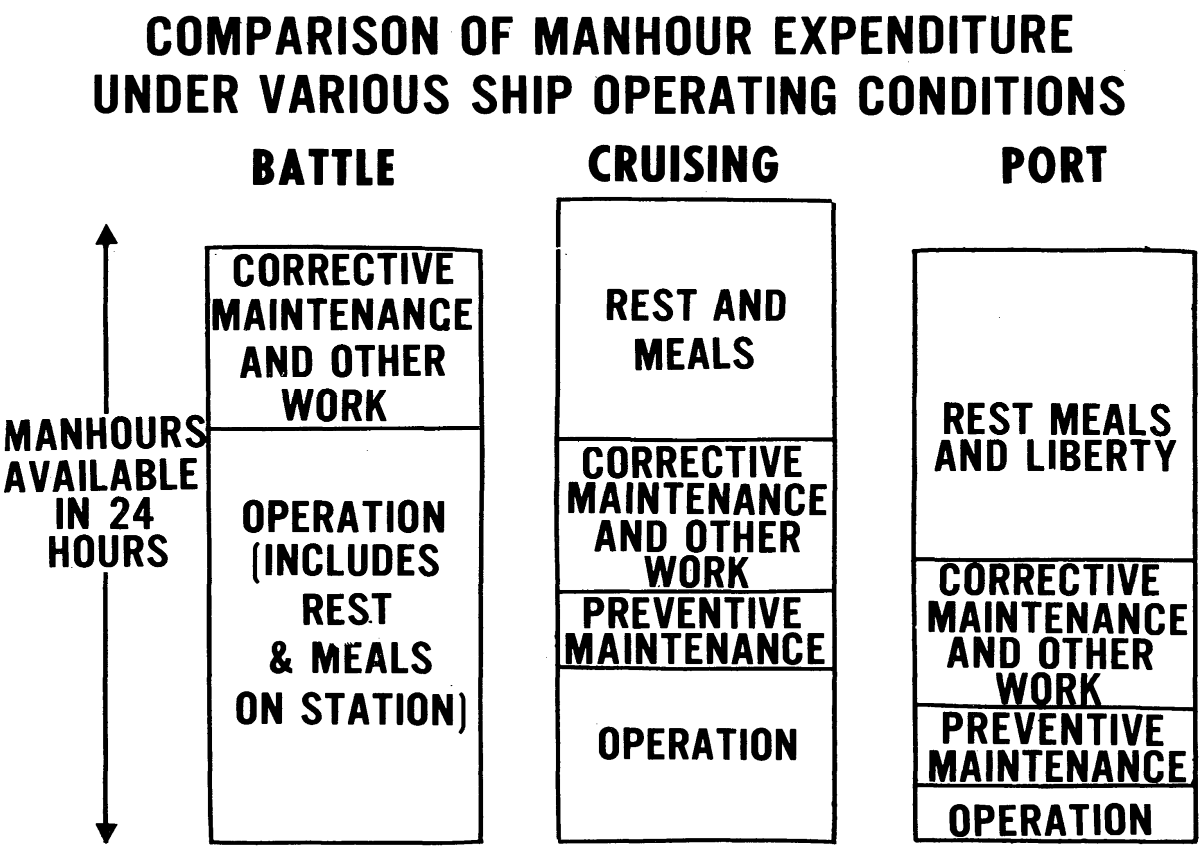


Figure 2.

tasks performed by the crew not previously defined, such as taking care of authorized personal problems, training, haircuts during working hours, office work, etc. These tasks cannot be quantitatively determined by the Bureau design engineers. Thus, these manhour requirements must be measured by Fleet Work Study Teams on similar active ships. The results of their measurements are then adapted to the ship being designed as manhour *estimates*. These *are estimates* but are more accurate than guesses. The manhour estimates for corrective maintenance and other work are again superimposed on the manhour blocks as shown in Fig. 2. To the above manhour blocks in Fig. 2 must be added the necessary manhours expended for rest, meals and liberty under all conditions. After establishing the manhours required in each condition during a 24 hour period, it is possible to see which condition is manpower governing. In the example shown in Fig. 2, it is assumed that the Cruising Condition is the manpower governing condition. Thus, this condition will govern the complement of the ship, and the ship must be designed to carry at least this number of men. Also, if further reductions in complement are required, the designer must work on the Cruising Condition until its manhour requirements are equal to or below the manhour requirements of other operating conditions.

Having established the operational and maintenance manning as described above, Fig. 1, and having determined the complement by the procedure described, Fig. 2, the Bureau can now provide the Contract Plans and Specifications with Manning Requirements. Growth complement margins can be added to the Manning Requirements by the customer — CNO in this case — as desired, but the ship has actually been designed for the minimum number of men to operate and maintain it.

This, again, is not the whole story. For if it were possible to design a perfect ship, poor shipboard management could certainly make it necessary to provide more men to do the work. Thus, as shown in Fig. 1, shipboard *Management* is another input that must be controlled. Although the Navy has efficient operational management, it must develop more efficient shipboard maintenance management to actually realize effective minimum manning. The Navy now has a major program to improve preventive maintenance management on ships. Thus, for the purposes of this paper, it is assumed that effective shipboard preventive maintenance is accomplished and that the materials for the preventive maintenance management systems, i.e., planning guides, planning boards, data, etc., are installed in the completed ship.

As can be seen in Fig. 1, all three of the main conditions — Operating Maintenance — Management — have been satisfied to design ships that require the minimum number of men to operate and maintain them. This, then, is the Bureau of Ships' Design Work Study Program.

On the first ship designed using Work Study techniques, a Destroyer Escort, it is safe to estimate that about 90% of the systems within the ship have been improved either from a simplification point of view and/or from operational manning considerations. It is believed that a few examples of how this program has effected this ship's systems will be of interest.

Since the steering systems on most Navy surface ships are similar to those on Destroyer Escorts and since their design is greatly influenced by custom, the results of Work Study Critical Examination of this system are interesting. The present steering system on DE's is shown in Fig. 3. Four (4) men are assigned per watch to operate this system. A helmsman, lee helmsman, and telephone talker are in the Pilot House. The lee helmsman is provided to split the helmsman's watch because of the fatiguing nature of the duty. The fourth man is in the Steering Machinery Room where he is ready to steer the ship when the helmsman loses control. This fourth man is in communication with the Pilot House talker.

In Fig. 3 each component of the system is represented by a circle. The black circles represent man operated components. The components in the upper portion of this figure are located in the Pilot House while those in the lower portion are located in the Steering Machinery Room. As this system is designed, no matter

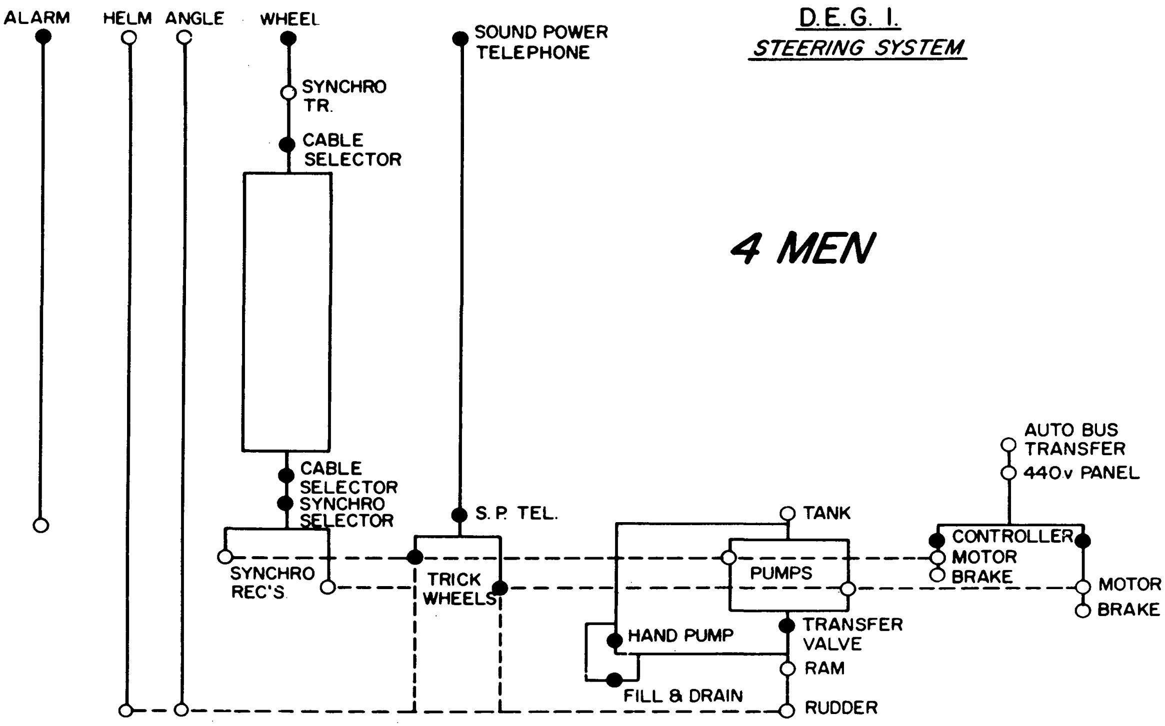


Figure 3.

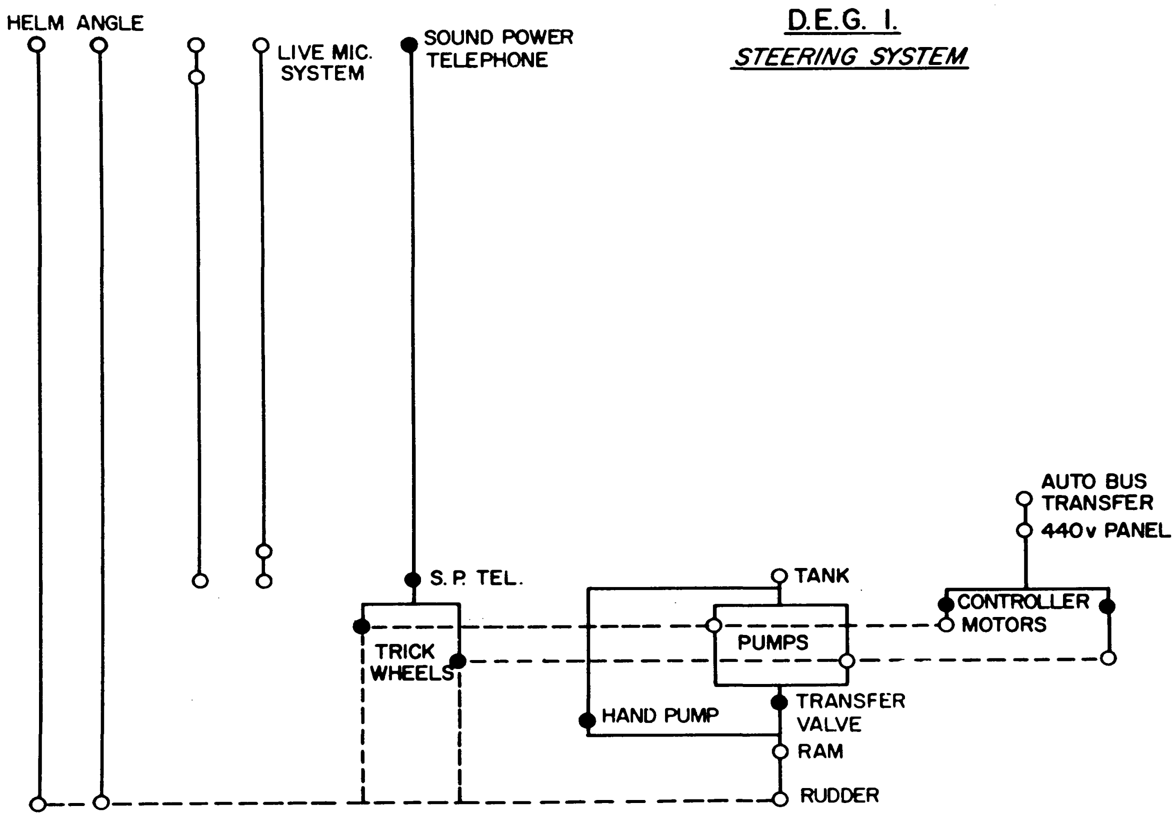


Figure 4.

what casualty occurs, the helmsman loses control; the man in the Steering Machinery Room must take control, realign the system for returning control to the Pilot House, and then transfer control to the Pilot House. Since few casualties actually occur, the man in the Steering Machinery Room has a demoralizing watch; in fact, even though he has telephone communication with the Pilot House at all times, it is still necessary to provide an alarm bell in the Steering Machinery Room, operated from the Pilot House, to alert this man whenever a casualty occurs.

As previously explained, this hardware system first is examined to provide the simplest system, i.e., the system having the least number of components having at least the same component reliability — this leads the way to minimum maintenance manpower. As a result of this critical examination, the system shown in Fig. 4 is developed. The following components are replaced with the indicated simpler ones or are eliminated:

1. The entire Synchro Signal System is replaced by a simple Live Microphone System.
2. The Fill and Drain Pump is eliminated by the use of the existing Hand Pump.
3. The Brakes are replaced by providing another position on the Transfer Valve.

The next step is the manning examination of this simpler system. This examination shows that the system of Fig. 4 could be manned by two (2) men under all conditions vice the present four (4) for the system of Fig. 3. In addition, the helmsman and lee helmsman in the Steering Machinery Room never lose control due to steering casualties. However, moving the helmsman from the Pilot House to the Steering Machinery Room is *against* Navy surface ship custom; thus, a major disadvantage. Through the manning examination, it is found that several foreign navies do not have the helmsman in the Pilot House, that the U. S. Navy's submarine Commanders operate their ships on the surface in confined waters with the helmsman down below, and that harbor pilots seldom care where the helmsmen are located as long as they hear and execute their orders. Yet, custom can well dictate the end result.

If the operators want the helmsman in the Pilot House, then how must the hardware system change to still require only two (2) men to operate the steering system? Fig. 5 shows such a system. It is immediately apparent that this system has more components than the original, Fig. 3, since it is necessary to operate all Steering Machinery Room man operated components from the Pilot House thereby eliminating the need for the Pilot House talker and the man in the Steering Machinery Room. Although the helmsman in the Pilot House does not lose control during the failure of a component of the system in Fig. 5, the entire system is more subject to battle damage. Although it is believed that under wartime cruising conditions only two (2) men would be required, during battle conditions an additional man would be required in the Steering Machinery Room. The system of Fig. 5 will also require more maintenance manpower.

This steering system example shows the effects of both the hardware system examination and the manning examination and how the use of remote or automatic controls does not necessarily mean a reduction in personnel.

To show the impact of Work Study on a typical engineering piping system, the boiler fuel oil service system is representative. In Fig. 6, the left diagram is the system as originally designed for the DEG-1. This system, as designed, can certainly be termed "standard practice." There are two service tanks (S.T.), three constant delivery pumps (P) each capable of supplying oil to one boiler (B) at full power, fuel oil meters (M), an accumulator (A) to supply oil to the boiler during momentary power interruptions, and a by-pass valve (V) to circulate excess oil from the constant delivery pumps back to the service tanks.

Critical examination of this system results in the system shown on the right

D.E.G. 1

STEERING SYSTEM

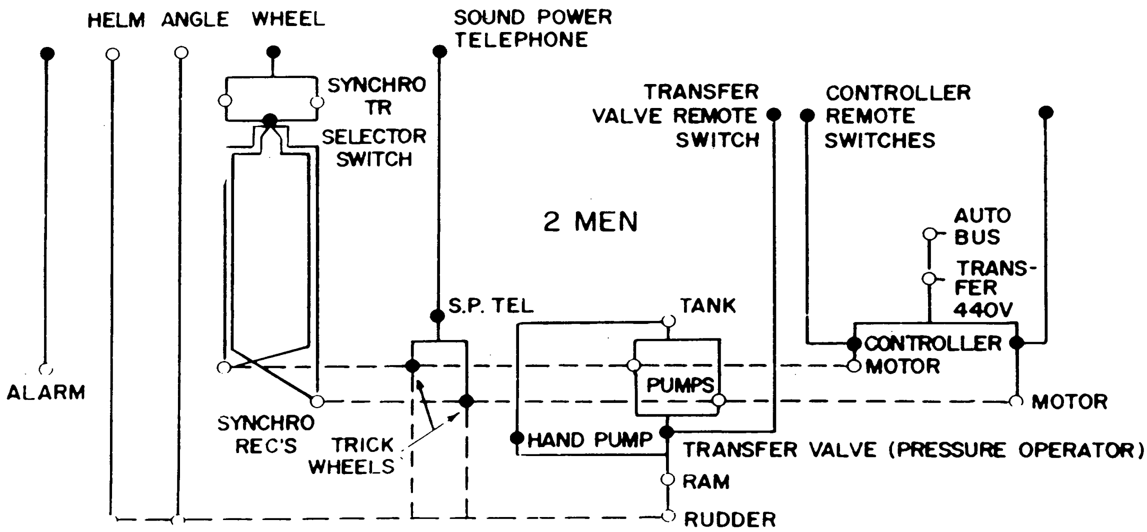


Figure 5.

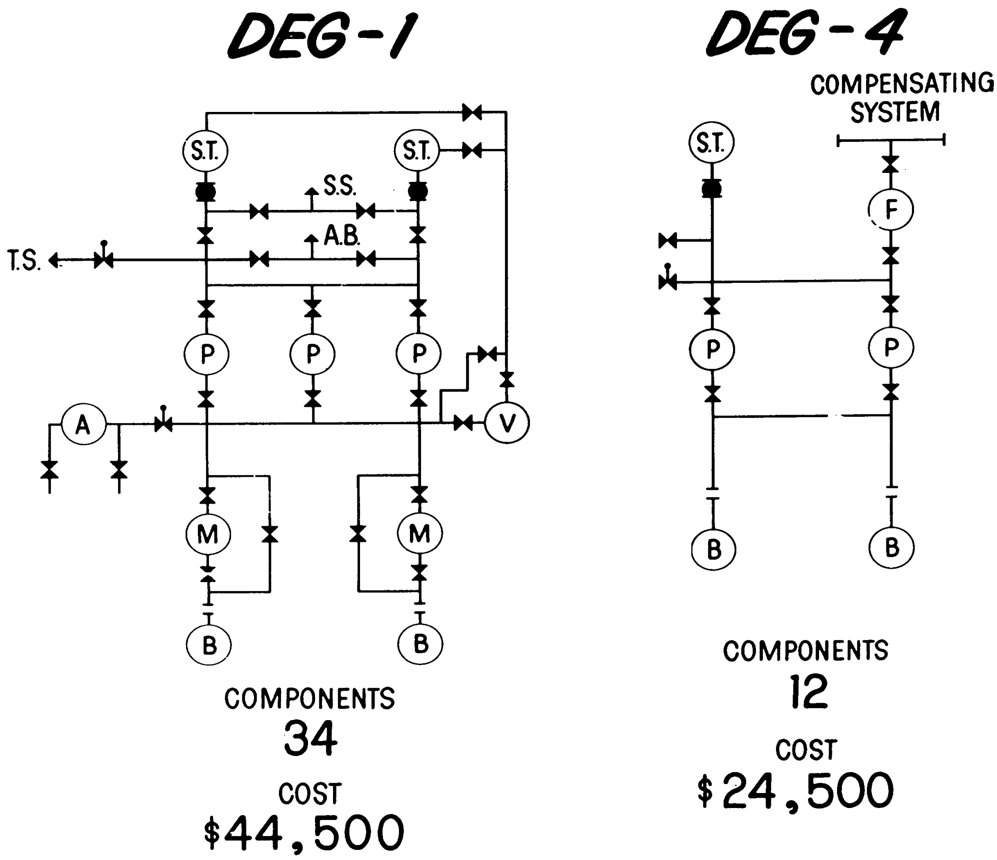


Figure 6.

side of Fig. 6. Since the ship has pressure — fired boilers, which burn either diesel oil or JP-5, a salt water — fuel oil compensating storage system was originally installed primarily for stability reasons. In such compensating storage systems, there are several tanks in series where salt water enters the tank at one end when oil is pumped from the tank at the other end. The oil in the end draw-off tank is free of water except upon replenishment. Under normal operating conditions, the fuel oil end tanks can be used as service tanks provided a filter (F) is installed in the line. In the revised system only one storage tank is required that is used while replenishing oil to the compensating storage system. Instead of installing constant delivery pumps, two variable delivery pumps are used, each capable of delivering oil to two (2) boilers at full power. This means that more standby pumping capacity is available. The use of variable delivery pumps eliminates the need for a by-pass fuel oil system to the service tanks. The accumulator is eliminated since both the fuel oil pumps and the feed water pumps are electrically driven. If a power failure occurs, it is best to stop the oil flow to the burner when the feed water is stopped. Also, by examination it was found that the meters are used only during official Bureau of Ships trials; thus, provisions can be made to temporarily install test meters.

These changes result in a savings of 22 components and in an estimated saving of $20,000 for components alone.

By applying Design Work Study to the redesign of the DEG-1 Class with the same characteristics, it was determined that the following gains could have resulted:

1. Complement decreased by about 36 men.
2. Ship length decreased 15 feet.
3. Displacement decreased by about 350 tons.
4. Stability Class I obtained.
5. Follow ship cost reduced by $700,000.

CONCLUSIONS:

The three problems of:

1. Continual growth of ships' complements.
2. Continual growth of ships' size.
3. Increased maintenance and unreliability

stimulated the demand for positive action and provided the incentive for Top Management to find new ship design techniques required to solve the problems. As can be well imagined, the effort to apply these new procedures to all ship types will demand the conserted effort and devotion of all our engineers who provide the U. S. Navy with the finest ships.

The Bureau of Ships is leading the way in developing new procedures for designing ships through the Design Work Study Program.

DISCUSSION

FRANKEL (MIT): I am very glad for your success in this Work Study program. I believe that a reduction of components in a system not only increases the reliability, but that the actual performance of a system will normally increase too. Performance means efficiency of response or the output of the system. We have found in many marine engineering systems that by increasing complexity to increase reliability, we actually decrease the performance. This is because the theoretical performance is very often far apart from the actual performance of a system.

One question comes to my mind in listening to the Work Study program and its results. Every manufacturer who supplies ships gets a couple of state points for the system he is designing, and designs it to the best of his ability without regard to the response of the output of his system to the next system in line. I am actually talking about the interaction of systems. It is not sufficient to show that the reliability of a system by itself is within limits. There is no completely independent system on a ship and, therefore, the reliability of one system must be within acceptable limits with all the possible responses from other interacting systems.

KAUFFMAN: Thank you very much, sir. You are perfectly correct and I agree with you 100%. I will admit that in the design work study program we have generally, for the first time, designed most of these systems as complete systems.

The techniques are there and in some cases we have investigated the interaction of systems. We haven't gone far enough, but the techniques are there for us to do it, but we still have lots to learn.

HIRSCHKOWITZ (U.S. Merchant Marine Academy): I'm wondering if you could answer what you mean by preventative maintenance or what you anticipate in this section?

KAUFFMAN: I think Commander Heenan will answer that completely. I'm not trying to put you off.

SULLIVAN (Maritime): Would you tell us what type of persons you used in evaluating and examining your systems. Are they different talents than the persons who would normally be designing?

KAUFFMAN: No. They are exactly the same engineers that have been designing these systems in the Bureau of Ships for a long time. Some have been there a long time, and some a short time, but they are regular engineers that do this work.

SULLIVAN: Let's say that someone who is a mechanical engineer designs the systems; would you bring electrical or industrial talents in who normally wouldn't have been involved in the design, but who would have a different outlook?

KAUFFMAN: Well, first off, there are very few systems that we find are entirely mechanical and entirely electrical or entirely hydraulic. So the first thing we had to do was take the supervisor of a group of engineers, and assign him the responsibility for this whole system. Since he had the responsibility for the whole system, he had the work and the critical examination work that were done by the electrical man, and the hydraulic man and the mechanical man, all were reviewed by the officer or supervisor responsible for the whole system, and in this way, we are getting at your problem I think.

We aren't organized in the Bureau of Ships this way, but we are actually designing these systems this way now, but it does mean interaction between sections to the extent we haven't had before.

SPATNEY (Ford Instrument): Would you go into changing general philosophy of, let's say systems. For instance, take a power plant, you normally get into a lot of analogue instrumentation. Would you go into a digital form to see if you could reduce the overall complexity of it and maybe share time on something like the MTDS or something?

KAUFFMAN: I'm trying to bring your question into context here. We use industrial computers techniques to solve some of our problems, but they are all individual problems thus far. We are actually turning to industry in the design division of the Bureau of Ships, and finding out how they are using computers in the initial designs stages and other design states on how they use computers to help them design systems. At this we are virgins.

LAWRENCE (U.S. Army): I'm wondering about these teams that go aboard the plant to pick up the data that goes into your research; in other words the feedback information. At present, is this a continuing program or is it something that is just being done in connection with a particular design effort?

KAUFFMAN: It is a continuing program in that the fleet work study people are actually getting this factural information from the fleet, on ships or aircraft, it doesn't matter. That information is available to the designer. In addition we can request that they make specific studies for us as the designer, and we have some of those things going on right now. We need more. We don't have enough fleet work study teams at the present time to obtain all the information that the designers need to know, but that is part of the program and we do work with the fleet work study people. Does that answer your question?

FRANCIS (Boston Naval Shipyard): How does the Bureau intend to handle design changes that come up after the contract plans, specifications and manning requirements have been laid down by this system?

KAUFFMAN: The Bureau of Ships, through the supervisor or the Shipyard Commander, will handle these things in this way — if you want to change a system from what has been relayed to you from the contract design stage, and incidentally you're going to have a lot more information than you had before, your changes in manning then will have to come back to the Bureau of Ships, because we have to feed it right to the Bureau of Naval Personnel, as you might affect the complement of the ship.

FRANCIS: I would like to ask one more question. In passing out the manning requirements under contract specifications, will any of the detailed information that forms the basis of these manning requirements be forwarded as well?

KAUFFMAN: Yes, some of the information will be forwarded. The information that is contained in the critical examination technique that we use which is a systematic way for the engineer to evaluate all the different possibilities which I told you about — that information won't be a part of the contract plans and specs. However, if you need that information on a particular system we will give it to you. If your people aren't trained, they will have a difficult time understanding it, however.

THAYER (Lykes Bros.): Your preventative and corrective maintenance is bound to vary with a piece of equipment as far as the material and even the manufacturer are concerned. Can you write your specifications in contract plans close enough so that you can get the exact piece of equipment and the particular manufacturer you want?

KAUFFMAN: The answer: how I wish we could! But I'm afraid that Congress and a few other people would be right on us.

THAYER: Doesn't this affect you?

KAUFFMAN: Absolutely. It can very much do so. If those estimates are wrong or the shipbuilder goes out and buys a component that is inferior, it could certainly affect the manning of that department and it may be that we have to put another man on the ship. In fact, we're not going to be fools about this thing either, particularly on the first few ships we do this on. We will have more bunks and accommodations built into the ships than our minimum manning shows. The Bureau of Naval Personnel, in any case, usually gives us an addition of 10% over and above what they normally thought their needs were anyway. Hence, we don't have to re-design the ship if we need another man.

It is certainly true, however, it would be great if we could actually go out and buy anything we wanted, from whom we wanted, but it's not going to happen.

COUTINHO (Grumman Aircraft): One of the basic problems is to get empirical data for the designer; but the designer at the beginning of a project can't quite tell you exactly what kind of data he needs, because he hasn't gone deep enough into his problems. The designer usually can only describe the required data after the problem is solved. I have seen many data collection systems, some of them quite elaborate, built up by specialists. The problem here is that the data is then often collected and processed for the convenience of the specialists who are collecting and processing the data, rather than from the point of view of what the designer really needs. The most valuable commodity in this particular area of trouble reporting is also the hardest thing to get, namely common sense. Somebody has to stand and watch this area very carefully; otherwise a great big operation with lots of paperwork will develop and won't really mean much.

KAUFFMAN: Yes sir, I think it is the number one difficulty — again in this business of management Heenan talks about feedback. However, there are a number of things that hurt you and me in trying to develop a system to get data we need. We have people that are trying to sell hardware and we have other services trying to sell us a system. We have all sorts of things that are always in our way in trying to get only the information we want, that we can use. We are faced with this right now in the United States Navy. We are fighting to get only the data we really need.

HIRSCHKOWITZ: It seems to me, being a Merchant Mariner, that one of the big problems was feedback, and I assume that a difficult area is that people do not remain as skilled practitioners in their field, but rotate, as has been the custom in the Navy. I think they may suffer by not being able to get this common sense approach that has just been so eloquently presented.

Now I'm wondering if the Navy has given any thought to the idea of doing as we do in the Merchant Marine where an engineering officer can go up the scale, come to a high prestige position as an engineer, and live engineering. He will know it whether he moves from ship to ship, which should in my mind be limited to a type of ship. I have a feeling that this feedback would be better than a lot of data. I'm always afraid with data that the engineers would get so involved with paper work, they would never get down to their engine rooms.

KAUFFMAN: Yes this has been considered. Here again I think when we talk about feedback systems this will be another problem I'll let Frank (Heenan) hit upon. He will give you a better feel for what we're thinking in this area. I agree with both of you gentlemen that this is one area and one field in which we've got to keep our head on our shoulders and not let people run away with it, because the minute the Bureau of Ships or any Bureau starts receiving tons of automatic data with which nobody knows what to do, the system will fall flat on its face anyway.

FRANKEL: The point that was raised just a minute ago is that we are unable to ever get exactly the component we require. I wonder instead of coming up with some deterministic and empirical requirements, very often based on the output of processing statistical data, if we shouldn't make much more use in defining tolerances and the interaction of tolerances as a certain probability distribution that a certain component's characteristics will come within certain limits, thus giving the designer much more flexibility in making his compromises. We are always confronted with having to make compromises, and we would rather have him make them within certain limits which we set.

KAUFFMAN: Yes. I have no doubt. You're stating here that if we get the right kind of statistical data through feedback, and if it is the kind we need; it can maybe relax tolerances and make the other component engineer's job easier and possibly improve reliability. It is possible that these things can happen, but at the same time the main problem is finding out what information you really want, so we're not feeding back tons of information that you can't use. I think this is what these gentlemen mean, and they have got to give the Navy time to do this. We've got to have time to find out what we need, and I think in doing this job right, we will possibly need industry's help in finding out exactly what we do need.

ARNOLD (United Aircraft): Do you have a feel yet for how many people or what percentage are required aboard your ships for preventative maintenance?

KAUFFMAN: Yes, we have a feel, and the Royal Navy has a better feel than we do because they are farther ahead than we are in this field. However, on the USS Lowry, they couldn't get their preventative maintenance done in the machinery and electrical area; by installing the system that Commander Heenan is going to tell you about, they get all of their maintenance done with the people off watch in an hour and a half or very close to it.

ARNOLD: This didn't quite answer the question I was looking for. Do you know how many people are added to the complement of the ship in general?

KAUFFMAN: Because of the maintenance?

ARNOLD: Because of preventative maintenance.

KAUFFMAN: Absolutely, we will know this figure as a number by rates.

ARNOLD: Do you have a feel for it now?

KAUFFMAN: No. The reason we don't have a feel for it is that we haven't analyzed or completed the analysis in these critical areas yet. They are behind everybody else. The electronic design people haven't finished their work, and so we can't get this maintenance added up until we know what all the components requirements are.

I think the closest we can come to this is in the electrical business which I think Frank Heenen can hit upon a little bit in his talk. I think we have a number there.

WELLING (Bureau of Ships): I was wondering whether you would care to comment on the work study as related to the degree of skill required of the operators. In other words do you take this into account in making your work study?

KAUFFMAN: Right now in the United States Navy the degree of skill required of operators has not necessarily been the major problem. The jobs that the electrical man has to do as a switchboard operator are simple. The jobs of the console operators in CIC, these people aren't hard to train. There are probably exceptions that if we make a mistake our Chief of Naval Personnel will call our attention to it.

Heenan is going to hit maintenance, and I'm not going to answer it.

KIN (Esso International): You speak of reliability of equipment. Are you asking in your specifications for the requirement of running a piece of equipment say from repair period to repair period?

KAUFFMAN: This is almost a subject in itself. The United States Navy, the Department of Defense, and the Secretary of the Navy right on down are tackling this problem. From the very conception of a brand new system, we are trying to specify the operational requirements all between maintenance periods. They are now trying to specify this from the CNO level right down to the technical bureaus.

It's just getting under way really. They have been working on this now for a year. We have the results of some of this work in the technical bureaus. But remember when the operators tell us what type of operational reliability they want, there is the mutual feedback that the technical bureaus must give them, because perhaps they are pricing themselves right out of the business. This is a joint path. We have lots and lots of work to do — this I will guarantee you. I think we will probably need industry's help in this one also. Thank you very much.

SHIPBOARD MAINTENANCE MANAGEMENT

COMMANDER F. E. HEENAN, ROYAL NAVY

The amount of Shipboard Maintenance has increased since World War II. One cause of the increase is the demand for faster reaction times of weapon systems. The faster reaction times tend to be achieved by the replacement of human links with automatic equipment. Hence, less people are required to operate systems, but more people with high technical skills are required to maintain them. However, the number of operating personnel is not reduced too much because the ship must insist on manual back-up of many systems in the event of degradation through enemy action or defects. The net result has been a growth of ships' complement since World War II and an increasing demand for higher skill levels. At the same time, the increased maintenance commitment tends to reduce ship availability.

In the Royal Navy the problem came to a head in 1957 when the British Government imposed a reduction in force upon the armed services. If the Navy was to keep the same number of ships at sea, the complement growth had to be arrested. Fleet Work Study was introduced to conduct studies ashore and afloat to see if the utilization of men and equipment could be improved.

The first Fleet Work Study Team went to sea in 1957 to find ways of "Doing better with what we had." The team applied their new techniques and gained acceptance of many proposals by studying enlisted men physically doing work. In those early days not one study resulted in any reduction in ships force. It only resulted in less man hours being expended on individual jobs. Tackling studies in this way the team realized that it would take a very long time to achieve sound results affecting the whole ship. Their next move was to tackle a department as a whole. Through this study the need for better maintenance management was revealed. The problem was analyzed and the causes of shortcomings in management were uncovered. A method of overcoming the problem was devised and tested at sea and a Management Improvement Program established. The problem will be further discussed under four main headings:

I The Need for Improved Maintenance Management
II What is Meant by Maintenance Management
III How to Overcome the Problem
IV Effect on Reliability

I. *The Need for Improved Maintenance Management.*

In one early study done in the Electrical Department aboard a fleet carrier, it was necessary to find out which condition of operating the ship governed the complement. Was it General Quarters; was it war time cruising; or was it self-maintenance in port? Study of the General Quarters' condition showed that 110 of the 145 men in the department could do all the work needed. This indicated that the General Quarters' condition was not complement governing. Further study of the remaining two conditions was therefore necessary. In the other two conditions the work of the men was primarily maintenance. The study consisted of Activity Sampling at sea and in port. The results of the sampling are shown in Fig. 1. Only one bar chart is shown because the difference between the sea and port conditions was within the $\pm 2\%$ absolute error of the sampling. The height of the bar represents

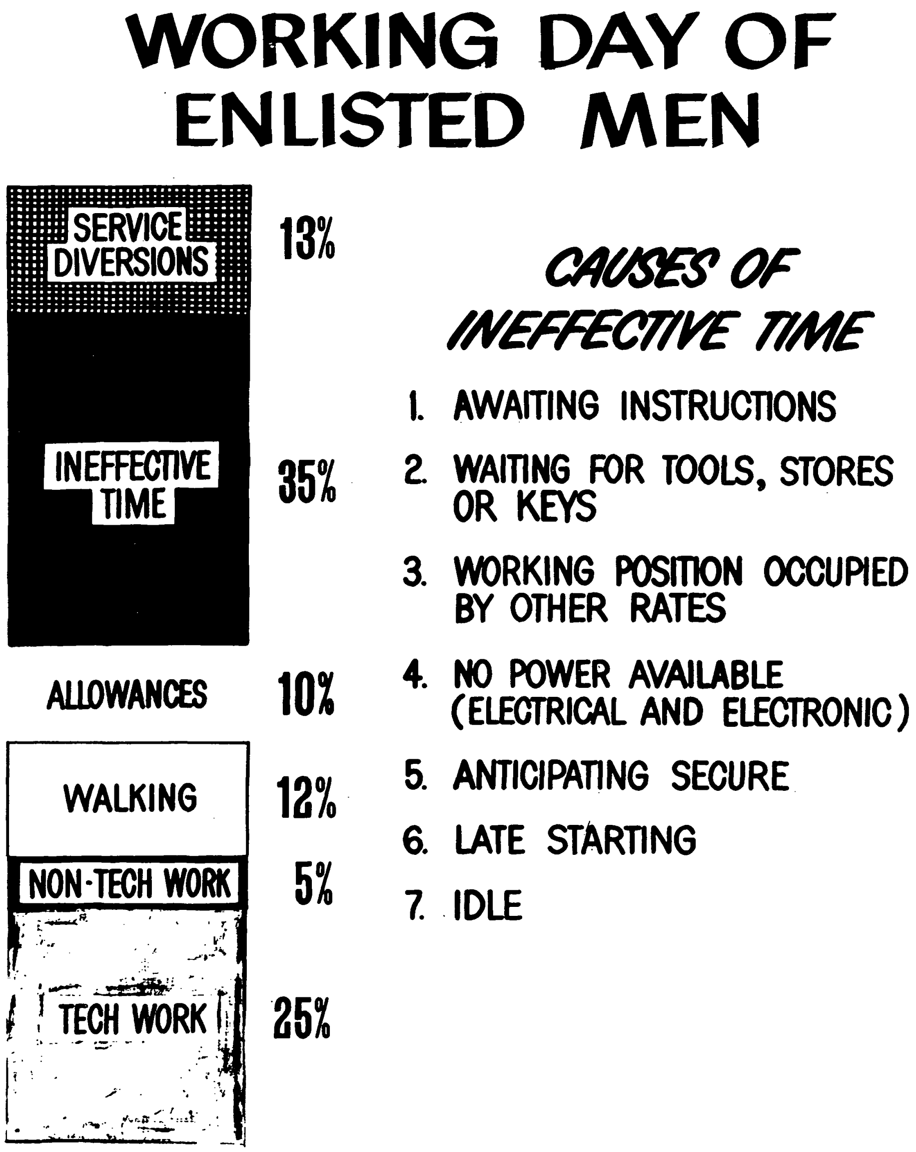


Figure 1.

145 men working for a seven-hour day. It does not include meal times or any authorized breaks. The term "Service Diversions" means the many things that sailors are allowed to do during working hours — sick call, haircut, school, visits to division officer, etc. Until this sampling was done, Work Study had been applied to work which came under the heading of "Technical Work." It now became clear that greater gains could be achieved by trying to reduce the ineffective time. Some of the causes of the ineffective time are listed in Fig. 1. It is wrong to use the "big stick" of discipline to rectify the situation as causes 1 to 4 are the fault of the people who enforce discipline. Reductions in the ineffective time can be achieved by improving the arrangements for getting maintenance work done — Maintenance Management. It is not suggested that the 35% ineffective time can be entirely eliminated; if it can be reduced to 20%, or 25%, then either more maintenance can be done by the same people or the same work can be done by less people. Other studies made in other departments in many ships of the Royal Navy and the United States Navy have produced an almost identical picture.

II. *What is Meant by Maintenance Management?*

In this part of the paper there is nothing new. It is a review of well-known principles highlighting those areas in which the Navy can make improvements. It is convenient to start with one of the many industrial definitions of Management:

> "The organization and control of human activity directed towards specific ends."

As with most definitions a little further analysis is needed in order to see exactly what is meant. Management can be broken down to four basic elements:

a) Define what is to be done
b) Provide the facilities
c) Provide the incentive
d) Make sure that it is done

In a Shipboard Maintenance Management context, these four headings can be expanded as shown in Fig. 2. Although the incentives: Pay, Leadership and Human Relations are *vital* to the success of good management, it is suggested that the Navy has concentrated on these things for officers and rated men in the Navy and has done an extremely fine job. Perhaps the pay is inadequate, but then it always is; but in two World Wars and many minor conflicts, the Navy has proved itself second to none in Leadership and Human Relations. Having acknowledged the absolute necessity of the incentive elements of Management, they will not be discussed anymore in this paper.

Each level of Management must answer every one of the questions in Column 1 of Fig. 2, and provide the facilities at their own level. Some of these questions are often answered in the overall objective stated by top management. At each level of management the questions must be repeated in the appropriate scale. Where a piece of data cannot be provided at one level it must be ascertained that the next level down has the facilities, time and knowledge to provide it. In a very simple example a father may tell his son: "You are to be at school by 9:00 a.m. tomorrow." This instruction assumes that the son knows where the school is, how to get there, how long it will take him to get there by that method so that he can determine at what time he should leave home. The boy also has to determine what he should

ELEMENTS of MANAGEMENT

The organization and control of human activity directed toward specific ends.

Define what is to be done	Provide the facilities to do it	Provide the incentive to do it	Make sure that it is done
Who	Equipment		Check:
Where	Tools	Pay	Quality and Quantity of work done
When	Material	Leadership	
How		Human Relations	
How long it takes			
<u>Work Program</u>		<u>Incentives</u>	<u>Control The Program</u>

Figure 2.

wear and what books, pencils, etc. he should take. The father should satisfy himself that the son knows or is capable of finding out the answers to these questions and many more; such as, arranging to get up on time.

All this appears simple and obvious — and it is — but this simple fact is not satisfied for maintenance in the Navy which will be explained later. It is then the Programming and Control of maintenance work that needs to be discussed further.

The easiest and most logical part of maintenance to start on is Preventive Maintenance. Fig. 3 compares the procedure for planning maintenance work in industry with the procedure for planning operations in all the armed services of Western Countries. Although the words are different the procedures are the same. The Navy applies this procedure — and has done so for many years — in operational planning. It is a formal procedure taught in War Colleges. No such formal procedure has ever been taught as applying to maintenance and housekeeping chores which are the things we do for the majority of the ship's life.

PLANNING STEPS

WORK STUDY	*INDUSTRY*	*OPERATIONAL PLANNING*
SELECT	1. ESTABLISH THE OBJECTIVE	ASSIGNED MISSION
RECORD	2. ASSEMBLE DATA	ESTIMATE SITUATION
EXAMINE DEVELOP }	3. DEVISE A PROGRAM	DEVELOP THE PLAN
INSTALL	4. ISSUE INFORMATION AND INSTRUCTIONS	THE DIRECTIVE
MAINTAIN	5. CONTROL PROGRESS TOWARD OBJECTIVE	SUPERVISION AND EVALUATION OF PLANNED ACTION

Figure 3.

Admittedly some officers intuitively have used the procedure which they have learned through experience as a subordinate officer. However, today many department heads in smaller ships are young and have had insufficient service to learn from more experienced officers. We should, therefore, teach them what Work Programming and Control is and how to use it to help them get maximum utilization out of their maintenance manpower. The procedure shown in Fig. 3 can be used to establish an effective Maintenance Management System in a ship — that is the objective. Step 2 requires the "Assembly of Data." The data required is that shown in Columns 1 and 2 of Fig. 2 for each job of preventive maintenance on each item of equipment in the ship.

A review of Preventive Maintenance data at present available to ships revealed such vague requirements as: — Inspect dummy log daily. Check refrigerator pump weekly. It is clear that in most cases all of the required data is not provided to the ship; this implies that higher management assumed that the ship's staff could and should provide it. Is the assumption correct? If the ship does provide the remaining data, then wide variations of interpretation of the broad requirements must be accepted from ship to ship. In the ship, the responsibilities of interpretation are often delegated to the man who does the job. This means that any central feedback data from ships for reliability purposes is meaningless because the maintenance standards are variable and unknown. If the Preventive Maintenance requirements are to be standardized, the data should be provided by the Technical Bureaus.

The apparently simple questions of What, When, Where, etc. have some hidden meanings. For example: "Who," in this scale, really means: The degree of skill required; and "How" is the method description written for that skill level. "When" has three aspects: Frequency (how often) — occasion (hot, cold, power on, etc.) and sequence (when each job should be done in relation to others).

Devise a program is the next step which means: Working out an organization for using the data and getting the work done in the ship. Previously the program has been devised ashore by Technical Bureaus with the main object of getting maintenance feedback. The object should be to get the work done in the ship. Therefore, the program should be devised at sea in the environment in which it is to be used. In general, there are two simple but important deficiencies in existing Maintenance Systems. They are blind and historical. Blind means that the planning data are hidden in filing cabinets or books. There is no way for the supervisor to be able to see his whole problem and give him the flexibility that he needs. Because of this, charts are invariably made which are historical — they show what was done yesterday rather than presenting the work yet to be done. The work to be done should be out in the open and the history filed in a book or drawer until ready for analysis or reference.

The headings 4 and 5 in Fig. 3 will be discussed in the next part of the paper.

III. *How to Overcome the Problem*

The maintenance problem aboard ships has been appreciated for some years. Several attempts have been made to overcome the problem by introducing Preventive Maintenance Systems. The Royal Navy introduced the "Standard Documentation System;" the United States Navy have such acronyms as PRISM, POMSEE, and WHAM — all relating to Preventive Maintenance Systems. Not one of these systems is universally used or accepted throughout the Fleet. In order to overcome the maintenance problem it is desirable to uncover the reasons for non-acceptance. There are two major reasons:

a) Not one of these systems provided *all* the maintenance data.
b) There exists a lack of education in maintenance management.

The shortage of data and description of the data required has already been described in Section II above. It remains, therefore, to explain the lack of education in Management. Investigations revealed that there is often, among Naval people, a confusion between "Leadership" and "Management." It is often felt that the two are synonymous and yet all the things discussed in Section II above are not taught in leadership training. Management is taught at P.G. schools, but all officers and rated men do not take these courses. Many officers learn the procedures as applied to operational planning in War Colleges; but these courses are not normally taken before the age of about 30. There is a need to teach more practical Management at the Naval Academy and other Schools.

Through the lack of education in Management stems a prevalent belief that there are only two types of planning: The long term plan with "off the cuff" control such as, the Watch Station and Quarter Bill; and the minute detail planning of the production line in some industries. The need is for neither of these two extremes. We need the ability to give a man a half-day's work and know with reasonable precision that there is a half-day's work content in the job.

Because of the inflexibility of past maintenance systems designed primarily to provide "feedback," the impression is rife that all forms of planning are inflexible. It has been shown that the procedure to be used is the same as that which is used for operational planning — What needs to be more flexible than that? It must be admitted that the Navy has only had to face this problem since World War II. Before that time there was little doubt that General Quarters governed the complement of ships. In all other conditions the officers were more concerned with finding work for men to do to keep them occupied, than with getting high utilization of manpower. Now the complexity of ships has led to demands for more highly skilled maintenance manpower. Unless we become concerned with efficient manpower utilization, the costs will become exorbitant and there will be a demand for a ratio of highly skilled to semi-skilled men which is disproportionate to the national proportions available.

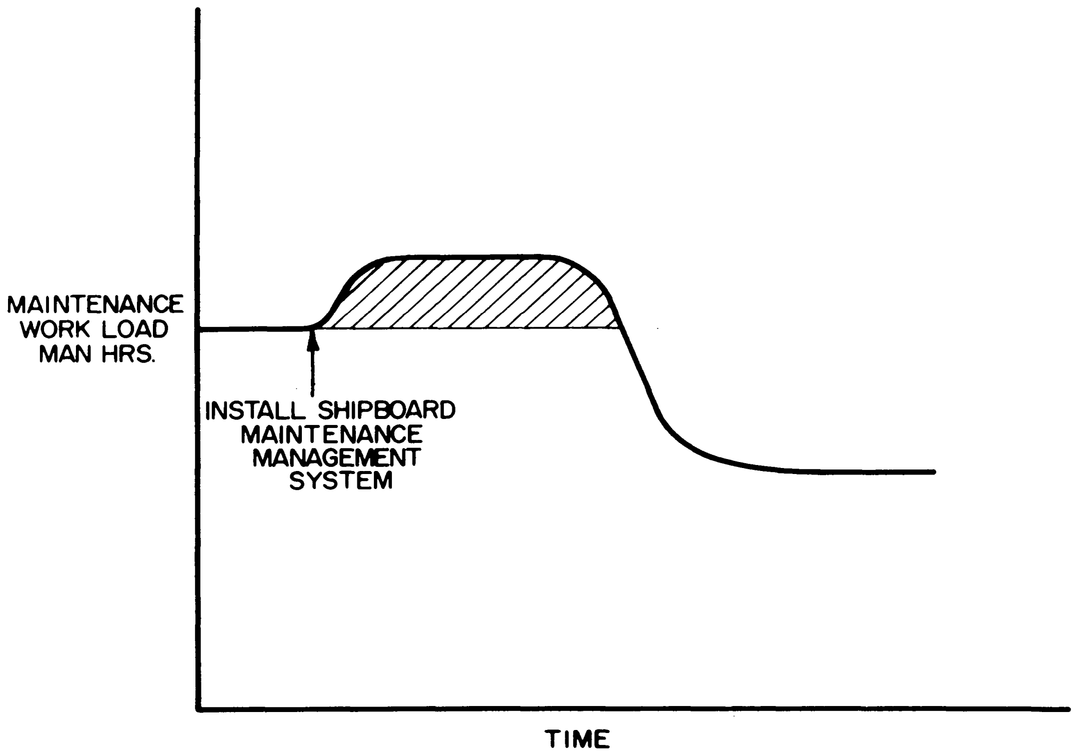
The Navy must embark upon a large scale program to improve Shipboard Maintenance Management. The Program must be conducted with great care and thorougnness if it is to be successful. The Program is as follows:

a) Provide all the Preventive Maintenance data for one ship.

b) Send a Fleet Work Study Team to sea in the one ship to devise a workable system as explained in Section II of this paper.

The System should be neither blind nor historical.

c) The Team should install the system, operate it, and prove that it will work. The ship's personnel must learn to operate it themselves during this period. The length of time required for installation is a function of the material condition of the ship at the time of installation. When the system is first installed, there will be an increase in workload as portrayed in Fig. 4. More Preventive Maintenance will be imposed on the ship in addition to the existing preventive and breakdown maintenance. In all probability, the need for more corrective maintenance will also be



EFFECT OF INSTALLING MAINTENANCE MANAGEMENT SYSTEM
ON MAINTENANCE WORK LOAD

Figure 4.

revealed. The workload will ultimately reduce; under ideal circumstances the reduction will occur about two months after installation. If the ship is left without assistance during this initial period, there is a serious risk that they will abandon the use of the system because of the increased workload. It is only after the ship's force have experienced the benefits of the system in terms of decreased workload that it will gain enthusiastic acceptance.

d) The next step is to train some installation teams and gradually to spread the system to more ships. At the same time the system must be accepted as The Standard Navy System. It can then be taught at selected officer and enlisted men training schools throughout the Navy. To improve Shipboard Maintenance throughout the Fleet is going to take at least five years. Attempts to accelerate the Program too quickly may lead to failure as past attempts have done.

e) Finally, to control progress towards the objective, all levels of Management, from the Departmental Officer up to CNO himself, must take an active interest in the Program. The existing types of inspection must continue but should be directed only to: "How well is the maintenance being done?" and not to "How well is the paper work being kept?" The control aspects of the system within the ship have already been developed. The control outside the ship and up the chain of command needs further investigation which is a part of the Navy's Shipboard Maintenance Management Program.

The procedure described above has already been applied in one U. S. Navy Destroyer with very gratifying results. In some instances where this procedure has been applied, the number of corrective maintenance items has been reduced by 50% or more. At present a Shipboard Maintenance Management improvement program is well under way in the Navy. Much remains to be done, but a good start has been made.

IV. *Effect of Maintenance Management on Reliability.*

It must be agreed that reliability should improve if all equipment is properly maintained in accordance with manufacturers' recommendations modified by user experience. Reliability can be further improved if user experience can be fed back to design. The need for some feedback data is apparent. Feedback data is needed for other reasons too. Because of the Navy's vital mobility, it is necessary to divide feedback into two quite distinct categories, i.e., the feedback needed by:

a) Line Management for:

Material Readiness
Operational Planning
Scheduling In Port Maintenance, etc.

Work *NOT* done needs to be reported to provide information on these things.

b) Technical Management for:

Rationalization of Maintenance Requirements
Repair Part Usage Data
Defect Analysis and Reliability

History or work done needs to be reported to provide information on these things.

Before the Shipboard Maintenance problem can be completely solved, it is necessary for a further study to be done to determine what feedback data is needed under each heading. A study of the best way of providing it from ships should follow.

Conclusions.

The Navy has a problem in Shipboard Maintenance. The solution in existing ships is through improved Management; first, through providing the necessary facilities; and second, through a program of education in Maintenance Management Procedures. The Navy should not be criticized for any shortcomings in their past

DISCUSSION

LAWRENCE (U.S. Army): My question is in regard to the seven step program. As I understand it, our navy is approximately in the third step. They have developed a program on one ship, and are making it operate. Is this correct sir?

HEENAN: That is true, and it has gone a little further.

LAWRENCE: The next step was to train teams and institute this program on other vessels. My question is, what calibre of people, what type of people are we talking about in these teams? Are these skilled enlisted men? Or are we talking about officer personnel?

HEENAN: Let me answer the first bit first: We have gone a little further than just divising a system and installing it in one ship. It has been expanded to nine ships on the West Coast, and four additional ships on the East Coast.

The installation in those ships has been done by the Fleet Work Study Groups. A Fleet Work Study Group consists of four officers, and eight Chief Petty officers. The officers are line or engineering officers who have done a full course in work study, and have worked on developing the Maintenance system. The Chief Petty officers are of various specialties and are trained in Work Study.

It is intended that the type commanders, Atlantic and Pacific, will provide skilled Petty Officers to be trained by the Fleet Work Study Groups in the Maintenance system. They can then install the system in their own flotilla or squadron. They will be supervised by an officer who again will either be a line officer or an engineer officer trained in work study.

HAUSCHILDT (Bureau of Ships): We've talked about work study and design of ships and also about the maintenance of ships. In which area do you feel we've had the greatest gain to date?

HEENAN: I wouldn't like to assign any measurement in answering that question. I think we have achieved tremendous gains in both areas. I think the greatest good soonest will come from the installation of preventive maintenance in the Navy as a whole. I'm sure that Work Study in Design will also produce great improvements in the United States Navy; but it is a long term gain compared with the preventive maintenance program.

THAYER (Lykes Bros.): How does a team on a ship decide what the proper interval is as an example to open a pump whether it is 6, 12, or 18 months. How do they come up with this time interval?

HEENAN: First of all the team look up the Maintenance requirements in the Bureau of Ships and other technical manuals: These are sometimes found to be somewhat unrealistic.

For example: It was found that if all of the required preventive maintenance was done on electric motors aboard USS LOWRY they would have to employ 10,000 man days of effort every two years on motors alone.

Upon questioning this with the people responsible for electric motors, it was found that much of it could be eliminated.

Preventive Maintenance requirements are established from technical manual and various manufacturers technical manuals. The team get the opinion of the Senior Chief Petty Officer aboard responsible for this equipment, and they form a judgment. This is the best that can be done in the first instance. Future refinements will have to be made through a feedback process.

THAYER: Do you lose much equipment this way?

HEENAN: No, we don't lose any equipment that way. The tendency is usually to err on the side of doing these things too often; rather than not doing them often enough. Technical people have a lot of common sense, and they will tend, usually, to err on the safe side.

LARSON (Pratt and Whitney): Isn't it part of your program to cause this feedback to come back to the original equipment suppliers so that they too can add their two cents worth in trying to develop this program of preventive maintenance as it relates to your concepts of this?

HEENAN: Yes indeed. When we eventually get the feedback, it will affect reliability we feel in two ways: The feedback will help us to make sure that the periodicity on the requirements of preventive maintenance are more realistic than they are at the moment.

Secondly, an analysis of the defects which have occurred in equipment can be fedback to the designer which will be good information for him to determine what were the causes of defects in his equipment, and help him to design those causes out.

We have a lot of work to do before we can get to that very happy state of being able to tell the manufacturer exactly what the problems are in his component.

LARSON: Following through on that same point. Are you at this time requesting him to feedback to you his experience in fields that might be allied or related to the problems you are concerned with, so that you will get the benefit of his service accumulation.

HEENAN: Yes, but not in any formal way. You will remember, however, that we are also getting the advice of the contractor through his technical manual or through consultation on what they think the preventive maintenance requirements should be. In this way we do get some of the benefits of the manufacturers knowledge from associated applications of their equipment.

MC INTOSH (USCG 9th Dist.): I would like to ask a question about personnel: how does this program affect our shipboard training system. What do we do for the "striker"? It appears to me that you put just enough bodies aboard ship to do the maintenance work, no extra bodies. There isn't any room for this fellow who wants to train, who wants to become a Petty Officer. How do we cope with that?

HEENAN: We cope very well with that aspect in two ways. The Bureau of Ships is providing the Bureau of Personnel with the minimum manpower requirements to operate and maintain the ship, through the Design Work Study Program. The Bureau of Naval Personnel know their training requirements and they add training and growth allowances to the complement recommended by the Bureau.

Another way in which we are helping considerably is that the Maintenance Management Program is providing job cards which are an excellent training aid. The job cards give a step by step procedure for each Preventive Maintenance job. Upon installation of the Maintenance system one hears Chief Petty officers saying such things as: "That striker is now doing work which I had to do myself before the system came." Previously the Chief Petty Officer had to stand over the striker and explain to him step by step how to do a job. It was usually quicker for the Chief Petty Officer to do the work himself. There can be no doubt that training for advancement is much easier.

SULLIVAN (Maritime): I was interested in knowing in what time period you could expect the payoff in reduced maintenance to show up from use of this system?

HEENAN: The time before the payoff is seen may vary. The time is a function of the material state of the ship at the time of installation. It may be anything from 2 months in a ship in good material state to 2 years in an old ship in poor material state. I hope we don't have any of the latter; but if we do, it may be better to hold up installation until after an overhaul.

OLSON (Westinghouse): Part of your program is to gather reliability data. What form do you anticipate this data will assume?

HEENAN: I don't know. The first thing we should do is to form a team. I don't mean a committee, I mean a working team which will investigate what the Technical Bureaus need to know for reliability. Having determined what they need to know, the requirements should be questioned to insure that they are necessary. The method of collecting data and its format should then be studied in ships.

OLSON: What I was searching for is the availability type thing or mathematical probability thing.

HEENAN: I will answer that sir, by saying I don't know. We've got to study what we require, but I think this is the only way we can do it. Maybe it will have to be mathematical; maybe it will be a simple sampling study which will give us the data we need. We don't know.

FRANKEL (MIT): In the criteria or objectives in scheduling preventive maintenance, what balance of performance to availability, or performance to reliability do you actually assume for this past criterion?

HEENAN: What performance to availability, is that right?

FRANKEL: The past performance of a particular machinery item or system is very much related to its availability or reliability. We all design our systems to certain margins. We assume a certain minimum performance level, and this will affect the preventive maintenance schedule. Do we try to get a proper balance there or do we just base the preventive maintenance on reliability or availability of the system?

HEENAN: We of course would like to have 100% availability of the system when it is needed. We would like, in the ultimate, to phase our preventive maintenance so that we would have, say; 90 days available; 10 days for maintenance in which we do all the maintenance and never have any failure during the next 90 days. Utopia!

This we have not achieved of course. We have no means of measuring the performance to availability ratio right now. Let us maintain it as best we can, measure what performance to availability we have, through feedback, and then we will set about improving it.

KECECIOGLU (U. of Arizona): I have several questions. Question # 1: how much has the science of reliability spread presently in Great Britain. Question # 2: How can one plan the system of effective management if you still don't know what data you need, reliability wise, maintainability wise and the sequel of these two, availability wise. It seems to me that present systems of data gathering are highly ineffective. We conducted a study which we will report on tomorrow where very little additional data is required to be entered on forms, and this simple data isn't somehow transmitted to the people who ought to be getting and putting it on forms. Certainly I think the question here is what data do we need?

Essentially what we need is early failure distribution, random failure rates; we need wearout distributions, and the combination of the three, we need bathtub curves. Little additional information, such as putting hours of operation at the time the maintenance was made, what preceeded the maintenance action, whether this was repair or preventive maintenance would be what we essentially need.

Presently, the time the maintenance was performed is recorded in only about 20% of the forms we looked into.

There is failure report in system. But only 10 to 20% of failures are reported on these forms. This means that we still haven't convinced people that this is a very vital reliability data. That from this data we can specify and optimize preventive maintenance schedules and then provide the proper spare parts to cover both random and wear out failures.

We should be grateful to ONR for sponsoring the two studies; one of Igor Bazovsky and the second our own. Now we can transmit this information to you so that you now can start gathering this data properly and use it properly.

HEENAN: Firstly: I would say that Reliability Engineering in England is probably in about the same stage as it is here. I have been in America for about 20 months so my information may be out of date.

In answer to the second part of your question: I agree there is data available in this country at the moment in the Navy. As you say it is only about 20 to 25% of the failures which occur. Even if you had 100%, it wouldn't do you any good, because you do not know to what standard the equipment has been maintained.

Let me give you a very simple example. Two similar automobiles; one owner does his preventive maintenance; he changes the oil regularly and keeps his car beautifully for 10 years. The other one does nothing apart from putting in water and gas. If you get feedback data from those two automobiles after 10 years, you will get conflicting failure rate data which will tell you nothing. You have got to maintain those two automobiles to a standard, then and only then does data become meaningful.

Even if we had 100% valid data, at present, reliability predictions would be useless as we do not know the standard to which ships or machinery are maintained. A part of the program for improving preventive maintenance is an educational program. This will help to persuade people to give us good feedback data. One major cause of invalid feedback data is the many different reporting systems imposed upon ships. A part of the Maintenance Management program is to substitute one standard system to replace the many diversified systems.

KECECIOGLU: Does this go back to the level of skills the maintenance crew should have and to the level and completeness of instructions given them?

HEENAN: Yes. The job cards provided are an excellent training aid and they reduce the level of skill required to perform many tasks. Eventually, the system may influence the demarcation between specialties.

FRANCIS (Boston Naval Shipyd): Who actually prepares these instructions that tell the people what to do and how to do it and when to do it under this system?

HEENAN: Up to now this has been done by the Fleet Work Study Group in conjunction with the BuShips technical manual and all the appropriate technical manuals plus the advice of the officers and chief petty officers aboard the ship. On completion, all of the information has been submitted to the Bureau of Ships, 600 Codes, for their approval. They have approved or changed certain aspects of the job cards.

FRANCIS: When you people go aboard a ship do you find that you have an effect of improving the efficiency by your very presence and if that happens, how do you account for it?

HEENAN: In the first two weeks or so the efficiency is improved by the team's presence. After that even the sailor can't keep it up any longer. The reason for the improvement is well known. It is simply that more interest is being shown in the work being done by the individual.

HIRSCHKOWITZ (U.S. Merchant Marine Academy): Number one, I would like to comment on your wonderful presentation. I think it was very excellent from my view point and I'm sure that the group here will share that view. The other thing that came to mind and perhaps this was said here in a different manner, but you comment again and again that the importance of our data is dependent upon the reliability of the people aboard the vessel to feed it back.

Now after your training crew comes aboard a ship, and shall we say spends several weeks, I don't know what the time duration is: is there any system devised in your Navy so that these men after having been trained will remain on that ship or as customarily on many merchant ships and possibly our own navy, be rotated around to different ships so that the whole team that has been trained will disintegrate and will have to go all over again, and I think that this is one of the problems that I would see in front of us in the Merchant Marine particularly.

HEENAN: This is a very interesting problem. The Royal Navy has a slightly different procedure of rotating crews. In fact ours is probably more detrimental to the perpetuation of the maintenance system. Where the United States Navy rotates 10% of the crew the whole time, the Royal Navy rotates 90% of the crew at the end of each 2 year commission.

The E-1 system which is what the engineering system is called in the Royal Navy has gone over a re-commissioning and has still continued to operate and be successful for a number of months after that, about a year I believe. There can be no doubt that we have an educational problem, hence, the need to teach the system Navy wide.

FIXMAN (Maritime): I would like to add a little to your planned presentation about the merchant ship and the difficulty with this data business. We devised a very simple form, a one sheet form, to try to obtain data on the maintenance man hours in deck departments and engineering departments of merchant ships, and here is the type of results you get.

This covers a total of 86,000 man hours. We first found that 40% of the time was for watch standing, and we found that 30% of the remaining time some 52,000 man hours was spent in painting and chipping. 19% in cargo work, and this is what is interesting, 12% in miscellaneous, and everything from there on down is less than 12% and then we find other miscellaneous items, 5%, which means that 17% of the man hours was in miscellaneous work which no one can account for. So obviously the 2, and 3, and 4% items have no value at all from the standpoint of statistics.

Now looking at the machinery department of the ships and we found very similar things, over 13,000 man hours of work, 12.2% of the time is spent cleaning; 5.6% of the time in other things, and finally you get 8% in electrical work, and 57% in mechanical work, and for example refractories, .5% of the man hours. So obviously this kind of data is of little value. This is what you get if you devise a simple form, give it to an operating ship and say fill it out. So this is no good.

In the system that Commander Heenan has explained; you bring in a team of people who know what they are trying to do; they know what kind of data they want and get this data and get off the ship, I hope in less than two years. Perhaps then we will then have some maintenance man hours information which can be used to try and determine reliability rates.

HEENAN: May I say in reply to that we have proved, beyond all doubt, that self recording is not a very reliable method of getting information. We are not gathering reliability data while installing the Planned Maintenance System. We are educating the crew in Planned Preventive Maintenance and demonstrating the benefits of it in terms of reduced unscheduled maintenance. The problem of getting valid Reliability data fed back still remains unsolved though it is being tackled by the Navy.

RELIABILITY MATHEMATICS AS APPLIED TO SHIPBOARD MACHINERY

IGOR BAZOVSKY
Raytheon Company

1. INTRODUCTION

The long-term behavior of shipboard machinery in service is determined by a) the machine design, b) the operational environment, and c) the applied maintenance policy. These three factors must all be considered in the machine design stage. Design must suit a specified operational environment to achieve reliability and longevity in operation and it must also provide for easy maintenance. Reliability, with longevity, and maintainability are the two basic parameters of a design which determine the usefulness of shipboard machinery in long-term operations.

Reliability is defined as the probability of operation without failure for a required period of time, and the capability of maintaining a specified level of reliability over long periods of time is longevity. Maintainability is defined as the probability of accomplishing maintenance actions in a predictable maintenance time. A measure of reliability is the mean time between failures while maintainability is measured in terms of the mean time to accomplish maintenance actions.

There are three general types of maintenance actions:

a) *Repair or corrective maintenance* takes place when the machinery fails to perform its function in a satisfactory way so that its effective operation is stopped. Under repair maintenance, the failed part or parts are usually replaced by spares. The failure may have been caused by chance, or by creeping wear or fatigue. Repair maintenance cannot be prescheduled because failures occur at random times.

b) *Preventive maintenance* takes place at scheduled intervals. Under preventive maintenance, parts nearing wearout or fatigue are replaced or necessary adjustments are made to forestall normal wearout and fatigue failures during equipment operation.

c) *Servicing* is performed regularly during operation or at suitable intervals to keep moving parts lubricated and to make adjustments to prevent premature wear.

Reliability furnishes a measure of the frequency of chance and wearout failures, and thus determines the required frequency of maintenance actions. Maintainability is a measure of the time required to accomplish repair and preventive maintenance actions and provides a forecast of anticipated equipment down-time. The frequency of preventive maintenance actions depends on the equipment's wearout characteristics.

A machine has an inherent reliability which depends on the design and the variation of materials and processes in production. The inherent or design reliability determines the rate of increase of the probability of wearout failures, and this rate of increase governs decisions about the frequency of preventive maintenance actions. Preventive maintenance should be scheduled at optimum intervals to keep the probability of wearout failures down to an acceptable and/or most economical level. Wearout failures will then occur only at the frequency corresponding to the probability level set by the preventive maintenance schedules. Therefore, in long-term operations, a machine will adopt an operational reliability which is the resultant of its inherent design reliability and the scheduled frequency of preventive maintenance actions. Machine design for inherent reliability is of prime importance since it affects not only operational reliability between maintenance actions, but also the economy of maintenance by reducing the frequency of maintenance actions and by reducing the need for spares.

The economy of maintenance is also affected by the maintainability designed into a machine and through savings in maintenance manhours. Good design for maintainability may also save expensive checkout-and-test equipment and may save on maintenance facilities. Thus it can be seen that reliability and maintainability interact to have a direct effect on maintenance logistics.

Another parameter determined jointly by reliability and maintainability is the maximum possible utilization factor, or availability, of machines in long-term operations. The frequency of repair and preventive maintenance actions determines how often a machine is out of commission. The speed at which the maintenance actions can be performed determines how long a machine is out of operation when the need for maintenance arises. Availability is defined as the probability that a machine is in an operable condition, or is actually operating, at any given instant. Of particular interest in long-term operations is the steady state availability of a machine, often called the up-time ratio and defined as the ratio of the operational mean time between failures to the sum of the operational mean time between between failures and the mean maintenance down time. We use the term "operational mean time between failures" to emphasize that we refer here to the operational reliability which a machine exhibits under a specified preventive maintenance policy.

2. DESIGN RELIABILITY

a) Mechanical Parts

The reliability of a mechanical part is given by its capability to resist chance failures and wearout or fatigue failures in the operating time domain. The general mathematical expression for reliability is given by:

$$R(t) = e^{-\int_{o}^{t} \lambda(t)\, dt} \tag{1}$$

which is the probability that no failure will occur in the time period from t=0 (when the part first enters operation) to an operating age t. We refer to this probability as the cumulative reliability of a part. The term $\lambda(t)$ is called the instantaneous failure rate of the part and, in general, is a continuously increasing function of the operating time. It is defined by:

$$\lambda(t) = \frac{f(t)}{R(t)} \tag{2}$$

where f(t) is the underlying failure density distribution of a new part and R(t) is the cumulative reliability of Equation (1), which can also be written as:

$$R(t) = \int_{t}^{\infty} f(t)\, dt \tag{3}$$

When the part has reached an operating age T, its reliability to operate without failure for a subsequent t hours, that is from a time T to a time T + t, is:

$$R(t, T) = e^{-\int_T^{T+t} \lambda(t)\, dt} \tag{4}$$

which can be written as:

$$R(t, T) = \frac{e^{-\int_o^{T+t} \lambda(t)\, dt}}{e^{-\int_o^{T} \lambda(t)\, dt}} = \frac{R(T+t)}{R(T)} \tag{5}$$

and which is the conditional probability of surviving from T=0 to T + t, given that the part has survived to T. Further, R(t, T) can by definition also be expressed in terms of a single integral of the failure density distribution of a part at the operating age T:

$$R(t, T) = \int_t^\infty f(t, T)\, dt \tag{6}$$

where f (t, T) is connected with the underlying density of a new part f(t) by:

$$f(t, T) = \frac{f(T + t)}{R(T)} \tag{7}$$

A characteristic design parameter of a part is the mean time to failure. It is given for new parts by:

$$m = \int_o^\infty R(t)\, dt \tag{8}$$

where R(t) is the cumulative reliability of Equation (1). In that equation, the instantaneous failure rate $\lambda(t)$ can be thought of as being composed of two terms, one representing the rate of chance failure occurrence λ_c which can be considered as essentially constant with time, and the other representing the rate of wearout or fatigue failure occurrence $\lambda_w(t)$ which increases with operating age t:

$$\lambda(t) = \lambda_c + \lambda_w(t) \tag{9}$$

Equation (1) can then be written as:

$$R(t) = e^{-\lambda_c t} \times e^{-\int_o^t \lambda_w(t)\, dt} = R_c(t) \times R_w(t) \tag{10}$$

which is the product of the exponential probability of surviving chance $R_c(t)$ and the nonexponential probability of surviving wearout $R_w(t)$. It is here assumed that so called early failures caused by deficient materials or poor workmanship are preventively eliminated or weeded out.

Experience seems to prove that wearout failures (in which we include also fatigue failures) of well designed long-life mechanical parts follow a normal gaussian distribution, with a very good approximation. The normal density function is given by:

$$f(t) = \frac{1}{\sigma\sqrt{2\pi}} e^{-\frac{(t-M)^2}{2\sigma^2}} \tag{11}$$

where M is the mean wearout life, σ is the standard deviation of times to wearout and t is the operating age from T=0 when the part is new. Estimates of the parameters M and σ can be obtained from a wearout test of a large enough sample of N parts which yields N different times to wearout t:

$$M = \frac{\sum^{N} t_i}{N} \tag{12}$$

$$\sigma = \sqrt{\frac{\sum^{N} (t_i - M)^2}{N-1}} \tag{13}$$

Using the normal distribution for wearout, Equation (10) becomes:

$$R(t) = e^{-\lambda_c t} \int_t^{\infty} \frac{1}{\sigma\sqrt{2\pi}} e^{-\frac{(t-M)^2}{2\sigma^2}} dt \tag{14}$$

The numerical value of the integral can be obtained from normal tables when estimates of M and σ are available.

Equation (14) says that to design a part to a required numerical reliability for an operating time (age) t, attention must be paid to three parameters: the mean wearout life M, the standard deviation (spread) of wearout life around the mean, and the chance failure rate λ_c.

The normal probability density curve of wearout failures is shown in Fig. 1.

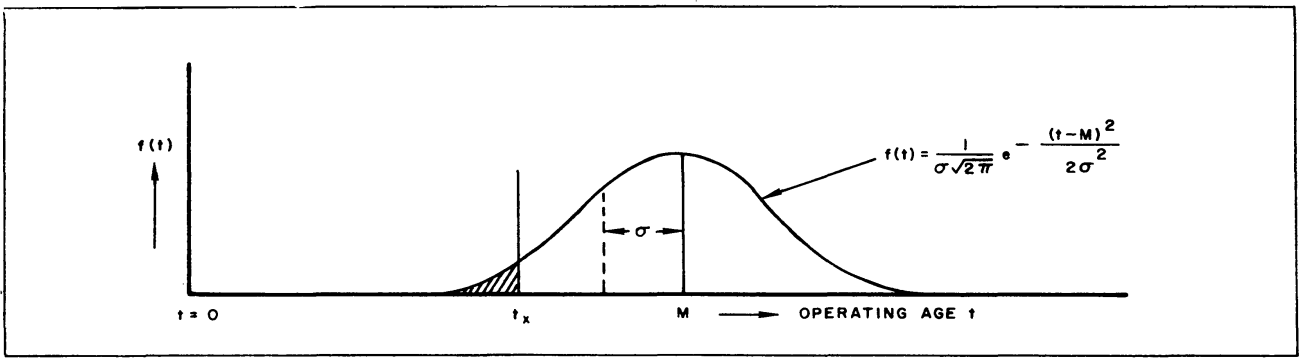


Figure 1.

The probability of surviving wearout from t=0 (when the part is new) to an operating age t_x is equal to the blank area under the f(t) curve to the right from t_x. The probability of wearout failure in the same period from t=0 to t_x is equal to the dashed tail area to the left from t_x. If t_x is the required operating age of the part and a numerical reliability requirement for surviving wear $R_w(t)$ is made for that part, the requirement can be met by various combinations of the numerical design goal for mean life M and σ, the probable standard deviation, which will depend largely on the variability of the quality of materials used and a close control over the production process to insure uniformity of quality from part to part. But once M and σ are given, the shape of the f(t) curve is determined. Another way of determining f(t) is to state two percentage points on the t axis. For instance, the time which fifty percent of the parts will survive (which is the mean wearout life M), and a shorter time which 90 percent of the parts will survive. The wear-out reliability for any time T can then be predicted.

As to the chance failure rate λ_c, this is a function of the parts design stress, that is, a function of part strength versus expected overstresses, and therefore a matter of designing with adequate safety factors.

Fig. 2 shows a generalized relationship between the chance failure rate λ_c and the design versus applied stresses.

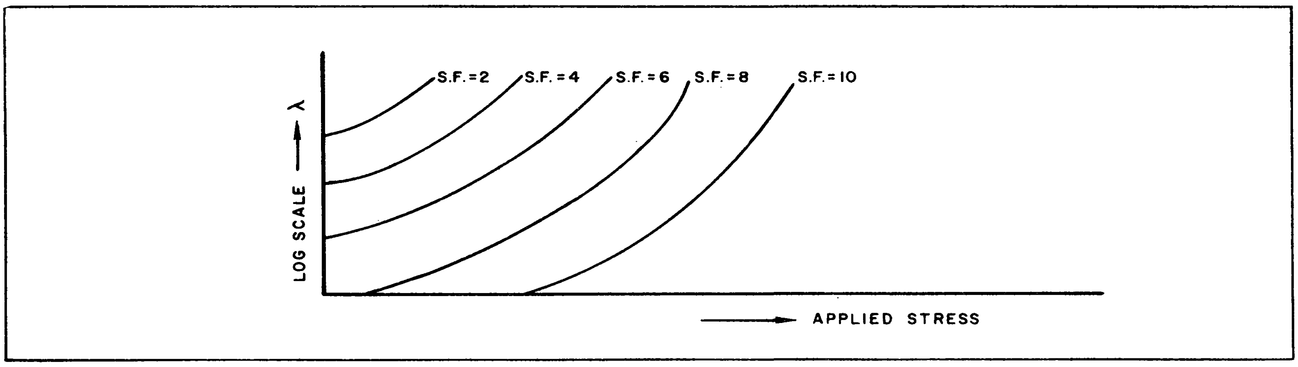


Figure 2.

b) Machines

From a reliability viewpoint machines are usually so called "series" systems of n parts. That is, if any part fails, the machine fails to perform to specification or stops completely. The machine can be restored by replacing the failed part or parts. Parts replacements constitute a partial renewal process of the machine.

Originally, when a machine is new, its reliability R_M is given by the product of the cumulative reliabilities of its n parts:

$$R_M(t) = \prod^{n} R_i(t) \tag{15}$$

where R_i of the i^{th} part is given by Equation (14). Therefore, in the case of combined chance and wearout failures:

$$R_M(t) = e^{-\sum^{n} \lambda_{ci} t} \times \prod^{n} \int_t^{\infty} \frac{1}{\sigma_i \sqrt{2\pi}} e^{-\frac{(t-M_i)^2}{2\sigma_i^2}} dt \tag{16}$$

and since the instantaneous failure rate of the machine is but the sum of the instantaneous failure rates of its n parts, i.e.,

$$\lambda_M(t) = \sum^{n} \left[\lambda_{ci} + \lambda_{wi}(t) \right] \tag{17}$$

Equation (16) can be written as:

$$R_M(t) = e^{-\int_o^t \sum^{n} \left[\lambda_{ci} + \lambda_{wi}(t) \right] dt} \tag{18}$$

where the instantaneous failure rate λ_{wi} of the i^{th} part is given by:

$$\lambda_{wi}(t) = \frac{\dfrac{1}{\sigma_i\sqrt{2\pi}}\, e^{-\frac{(t-M_i)^2}{2\,\sigma_i^2}}}{\dfrac{1}{\sigma_i\sqrt{2\pi}} \displaystyle\int_t^{\infty} e^{-\frac{(t-M_i)^2}{2\,\sigma_i^2}}\,dt} \tag{19}$$

For known M_i, and σ_i the values of λ_{wi} for various t can be readily evaluated from normal tables. Also tabulations of the standardized function $\lambda_{wi}(t) \times \sigma_i = r(t)$ exist.

Fig. 3 shows graphically the instantaneous failure rate λ_M of a two part machine, for illustrative purposes:

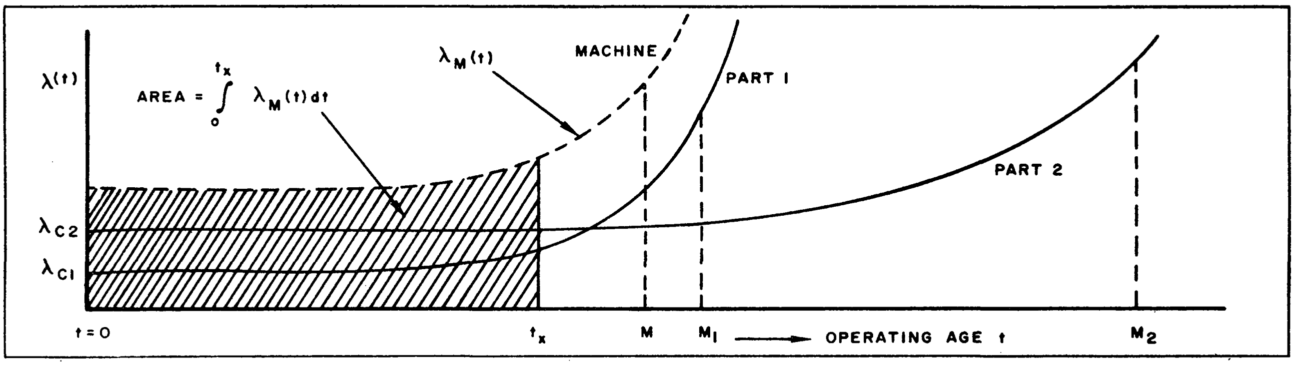


Figure 3.

Part I has a mean wearout life of M_1 and a chance failure rate λ_{c1}. The curve $\lambda_{c1} + \lambda_{w1}(t)$ has a slope which depends on M_1 and on the standard deviation σ_1. Similarly the curve $\lambda_{c2} + \lambda_{w2}(t)$ is determined by M_2 and σ_2 of part 2. The curve $\lambda_M(t)$ is the instantaneous failure rate of the machine, and $\lambda_M(t)$ is the integrand in the exponent of Equation (18), which for a two-part machine, is the sum of the instantaneous failure rates of the two parts. The numerical value of the whole exponent when integrated from t=0 to t_x is given by the shaded area in Fig. 3. It is thus the area under the $\lambda_M(t)$ curve. And the cumulative reliability of the new machine for the period t=0 to t_x is:

$$R_M(t_x) = e^{-\text{shaded area}} = e^{-\int_o^{t_x} \lambda_M(t)\,dt} \tag{20}$$

To design a machine for a numerical design reliability requirement $R_M(t_x)$ it becomes necessary to design its parts so that in the operating stress environment within the machine the product of the part's cumulative reliabilities for a period t=0 to t_x meets the specified value $R_M(t_x)$ or, which is the same, that the area under the $\lambda_M(t)$ curve does not exceed $-\ell n\ R_M(t_x)$, that is:

$$\int_o^{t_x} \lambda_{M(design)}\,dt \leqq -\ell n \left[R_{M(spec)}(t_x) \right] \tag{21}$$

This implies to design parts for acceptable values of M, σ and λ_c, which means that each part has to be designed for an adequate fatigue or wearout life M and with adequate safety factors, to meet the overall machine design reliability requirement.

Obviously, as Fig. 3 shows, the design lives of the parts must well exceed the mean time to first failure M of the machine. If, for illustrative purposes only, we assume that the mean time to first failure occurs at the 50 percent survival point (which is a good approximation), and we call this the "design life" of the machine, then according to Equation (15):

$$0.5 = \prod^{n} R_i(M) \tag{22}$$

It is easily seen, that for a machine with 100 series parts, the average reliability of each part at the M design life point of the machine must exceed 0.993 to satisfy Equation (22), because:

$$\ell n\ 0.5 = \sum^{n} \ell n\ R_i(M) \tag{23}$$

This also means that for the machine's design life of M, the design life of each part must exceed M by at least 2.5 of its standard deviation σ_i. But if the machine would have n = 1000 parts, then the average reliability of each part at the M point would have to exceed 0.9993, and the design life M_i of each part should exceed M by at least 3.5 σ_i, since allowance for chance failures must also be made as shown by Equation (14).

The exact value of the machine's mean time to first failure M is given by:

$$M = \int_0^{\infty} R_M(t)\ dt \tag{24}$$

where $R_M(t)$ is given by Equation (18). This point occurs, in general, between $R_M(t)$ values of 0.37 and 0.5. If machine failures would be of a pure chance nature, $R_M(M)$ would be 0.37. On the other hand, if machine failures would be strictly normally distributed, $R_M(M)$ would be 0.5.

Above we referred to the machine's mean time to first failure as the machine's "design life." But equally also the 90 percent survival time could be considered as design life. There exists a need for defining the term "design life" of a machine more exactly to improve communication between designers and those who specify requirements. In the meantime, however, whenever the term "design life" is used in connection with a machine, it should be made clear whether it is the machine's mean time to first failure, or the 50 percent or 90 percent survival time, or what is exactly meant by this term.

The reason why we introduced the concept of a machine's mean time to first failure M is the fact that once a machine fails, certain parts are replaced, while other parts which have already accumulated some substantial operating time are left in the machine. The machine is thus partially renewed, but not completely. Its mean time to second failure, counted from the time the partial renewal was made, will be shorter than M. This renewal process becomes of importance when considering the machine's operational reliability in long term usage.

3. OPERATIONAL RELIABILITY

a) Mechanical Parts

In long term operations a mechanical part in a machine may be subject to several types of maintenance policy. It may be subject to repair maintenance only, if it is replaced by a new part only when it fails. Or it may be subject to repair and preventive maintenance. Under preventive maintenance a partially worn or aged part is replaced preventively by a new one to forestall machine failure in operation. Failure in operation may be much more expensive than preventive replacement, since it involves interruption of operation at an undesired time, and a failing part may damage many other parts, or even destroy the machine and cause damage to other associated equipment. Thus, it is often economically advantageous to apply a policy of preventive parts replacements, if the machine's design reliability is not adequate for the intended operational life of the system in which the machine is used.

Let us first look at the long term behavior of a part in a machine when no preventive maintenance is applied. With an underlying density distribution f (t), the cumulative reliability of a new part is

$$R(t) = \int_t^\infty f(t)\, dt \tag{25}$$

and its mean time to failure is

$$M = \int_o^\infty R(t)\, dt \tag{26}$$

At an operating age T, the reliability of the part becomes

$$R(t, T) = \frac{R(T+t)}{R(T)} \tag{27}$$

and its mean remaining life, which is the mean time to failure of a part aged T, becomes

$$M_T = \int_{t=o}^\infty R(t, T) = \frac{1}{R(T)} \int_{t=o}^\infty R(T+t)\, dt \tag{28}$$

where t = o occurs at the age T. Substituting x = T + t, and dx = dt, so that x = T at t = o, Equation (28) can be written in the form

$$M_T = \frac{1}{R(T)} \int_{x=T}^\infty R(x)\, dx \tag{29}$$

where x is counted from the time the part was new.

Now if the part fails and is replaced by a new equal part the reliability, and the mean time to failure of the new part, will again be given by Equations (25) through (29). However, time in these equations must be counted from the time when the part entered service. Thus, to evaluate the exact contribution of a replaced part to the reliability of a machine at any time we must keep track of the time when the replacement part entered service and of its operating time accumulated since that time.

If no tract is kept of the times at which replacements have been made, the renewal theory permits to evaluate an average reliability of a part to operate from time T to time T + t, where T is counted in operate time from T = 0 when the machine first entered operation and when the first part was new. So we do not know exactly how often the part has failed and been replaced in time T, but we can calculate the probable or expected number of failures in time T when we know the cumulative reliability R (t) of the part.

Denoting by $E_N(t)$ the expected number of failures, i.e., replacements, of the part in an operating time t we obtain the average failure rate of the part as

$$\lambda_{AV}(t) = \frac{E_N(t)}{t} \tag{30}$$

where

$$E_N(t) = 1 - R(t) + \int_{\tau=o}^{t} f(\tau) \cdot E_N(t-\tau)\, d\tau \tag{31}$$

and this is the sum of an infinite series of probabilities: that is the probability of a single failure in t operating hours, plus two times the probability of two failures in sequence in t, plus three times the probability of three failures in sequence in t, plus etc., with the last term applying to the probability of an infinite number of failures in time t.

The average reliability of a part to operate from an arbitrary time T counted from the beginning when the machine was new, to a subsequent time T + t is then given by:

$$R_{AV}(t, T) = e^{-\int_T^{T+t} \lambda_{AV}(t)\, dt} \tag{32}$$

where $\lambda_{AV}(t)$ is given by Equation (30).

Now, the renewal theory proves that the numerical value of $\lambda_{AV}(t)$ approaches in the limit a constant value equal to the reciprocal of the part's original mean time to failure

$$\lambda_{AV} = \frac{1}{m} \tag{33}$$

regardless of the form of the original underlying density function. Equation (30) thus represents a transient condition of the part's average failure rate while Equation (33) is the part's long term stabilized average failure rate. The part's reliability in the steady-state renewal condition for an arbitrary period t averages out to:

$$R(t) = e^{-t/m} \tag{34}$$

since in a large population part failures then occur in a sequence of entirely random times and the process assumes an exponential character.

If the time when a part enters service is known, its reliability is calculated from Equation (27). The time keeping for many parts would, however, become a large operation and so also the calculations of machine and systems reliabilities for all possible combinations of parts ages. In long-term machine and system operations very good approximations are achieved by the use of Equations (33) and (34). They also permit the prediction of spare parts supplies, since over a fleet of X equal machines an average of Xt/m spares of a given part will be needed for each accumulated Xt operating unit hours if the part occurs once in each machine, and once the failure rate has stabilized.

The situation is substantially different when a policy of preventive parts replacements is applied. Two preventive maintenance policies are here considered. Firstly, the rescheduling policy under which a part is preventively replaced always after it has been in operation for T hours and is correctly replaced when it fails.

Let the reliability of a part be $R(T)$ for the replacement period T. The reliability that there will be no failure in n such periods between replacements, that is in an operating time nT, is $R(T)^n$, and for a period of $t = nT + \tau$ hours where τ is smaller than T, the cumulative reliability that there will be no failure in t is:

$$\psi(t) = \left[R(T)\right]^n \cdot R(t - nT) \tag{35}$$

because $\tau = t - nT$.

The mean time until failure occurs in such a sequence of preventive parts replacements is then:

$$\begin{aligned} M_T &= \int_o^\infty \psi(t)\,dt \\ &= \int_o^T R(t)\,dt + R(T)\int_o^T R(t)\,dt + R(T)^2\int_o^T R(t)\,dt + \ldots. \\ &= \int_o^T R(t)\,dt\left[1 + R(T) + R(T)^2 + R(T)^3 + \ldots..\right] \end{aligned} \tag{36}$$

because $\int_o^\infty \psi(t)\,dt$ is the total area under the $\psi(t)$ curve and this consists of an infinite sum of areas under $\psi(t)$ for subsequent periods of length T. Since $R(T) < 1$, the infinite series in the square brackets can be written as

$$\left[1 + \frac{1}{x} + \frac{1}{x^2} + \frac{1}{x^3} + ----\right] = \frac{x}{x-1} \text{ for } x > 1$$

and with $R(T) = \frac{1}{x}$ we obtain $\frac{x}{x-1} = \frac{1}{1-R(T)} = \frac{1}{Q(T)}$

so that the mean time to failure of a part subject to preventive maintenance every T hours is:

$$M_T = \frac{\int_o^T R(t)\,dt}{1 - R(T)} = \frac{\int_o^T R(t)\,dt}{Q(T)} \tag{37}$$

The reciprocal of M_T is the constant long-term stabilized average failure rate of a part subject to preventive replacement every T hours:

$$\lambda_{AV}(T) = \frac{Q(T)}{\int_o^T R(t)\,dt} \tag{38}$$

This corresponds to the λ_{AV} of Equation (33) without preventive maintenance. The failure density distribution obtained from the cumulative reliability $\psi(t)$ of Equation (35) is

$$f_r(t) = -\frac{1}{\psi(t)} \cdot \frac{d\psi(t)}{dt} \tag{39}$$

We could again apply the renewal process to obtain the transient average failure rate and average reliability, as done for the renewal process without preventive replacements in Equations (30) to (32). But this is not necessary since under a scheduled preventive replacement policy track must be kept of the preventive replacement times T and therefore the age of a part and its reliability will always be known within any period between two preventive replacements.

Also, knowing the stabilized constant failure rate $\lambda_{AV}(T)$, we can calculate the required number of spare supplies for long-term operations.

The term $\int_o^T R(t)\,dt$ in the denominator of Equation (38) is the mean time between both preventive and corrective replacements. The total replacement rate is therefore

$$r = \frac{1}{\int_0^T R(t)\, dt} \tag{40}$$

Since the total number of replacements is the sum of the preventive and corrective replacements, and the rate of corrective replacements equals the failure rate $\lambda_{AV}(T)$, we can write

$$\frac{1}{\int_0^T R(t)\, dt} = \frac{Q(T)}{\int_0^T R(t)\, dt} + \frac{R(T)}{\int_0^T R(t)\, dt} \tag{41}$$

where

$$\frac{R(T)}{\int_0^T R(t)\, dt} = r_P \tag{42}$$

is the preventive replacement rate. Then, over a long period t the average number of preventive replacements will be

$$E_{PN}(t) = \frac{t\, R(T)}{\int_0^T R(t)\, dt} \tag{43}$$

and the average number of corrective replacements (number of failures) will be

$$E_{CN}(t) = \frac{t\, Q(T)}{\int_0^T R(t)\, dt} \tag{44}$$

so that the total average number of spares required in t becomes

$$E_N(t) = E_{PN}(t) + E_{CN}(t) = \frac{t}{\int_0^T R(t)\, dt} \tag{45}$$

Applying X^2 confidence limits to Equation (44), the number of spares to be carried on a mission, during which no new spare supplies are obtainable, can be calculated for any required percent confidence that sufficient spares will be available to restore equipment operation in case of parts failures.

Also the minimum cost preventive maintenance period T can be calculated. Obviously, preventive maintenance can yield financial savings if the cost of an in-service failure C_F exceeds the cost of a preventive replacement C_P. Equations (43) and (44) give the expected number of preventive and corrective replacements in time t. The total cost of replacements in t is then

$$C(t) = C_P t \frac{R(T)}{\int_0^T R(t)\, dt} + C_F t \frac{Q(T)}{\int_0^T R(t)\, dt} \tag{46}$$

The cost rate per operating hour is $c = \frac{C(t)}{t}$:

$$c = C_P \frac{R(T)}{\int_0^T R(t)\, dt} + C_F \frac{Q(T)}{\int_0^T R(t)\, dt} = \frac{1}{\int_0^T R(t)\, dt} \left[C_P R(T) + C_F Q(T) \right] \tag{47}$$

Differentiating c with respect to T, and solving $\frac{dc}{dt} = 0$ for T one obtains the minimum cost preventive replacement interval T. Graphical solutions can be readily obtained by plotting Equation (47) as a curve and reading T_{MIN} where the curve shows a minimum. But a minimum will exist only if $C_P < C_F$. If $C_P > C_F$ then the minimum cost policy is obtained by not making preventive replacements. However, this may be unsatisfactory from the reliability viewpoint, if the part does not have a high enough inherent design reliability. When it is necessary to increase the operational reliability of a part above its design reliability, Equation (37) is used to determine the preventive replacement period T which will yield a required M_T, and the effect of such improvement on the maintenance cost can be found from Equation (46) or (47). Logically, an improvement of part reliability by preventive replacements can be achieved only if the part has an increasing instantaneous failure rate λ (t).

The other preventive maintenance policy considered in this paper is such that preventive replacements are made strictly every T operating hours regardless of whether or not the part has failed and been correctively replaced in the period since its last preventive replacement. The preventive replacement rate is this:

$$r_P = \frac{1}{T} \tag{48}$$

and the corrective replacement rate, which equals the stabilized long-term average failure rate becomes

$$\frac{\int_0^T Q(t)\,dt}{T \int_0^T R(t)\,dt} \leqq \lambda_{AV}(T) < \frac{Q(T)}{\int_0^T R(t)\,dt} \tag{49}$$

That means there is a lower and an upper bound to λ_{AV} (T) when the preventive replacement rate is $\frac{1}{T}$. Using $r_P = \frac{1}{T}$ and $\lambda_{AV}(T) = \frac{Q(T)}{\int_0^T R(t)\,dt}$ calculations for spares supplies and optimization of the preventive maintenance period T can again be made:

$$E_N(t) = t \left[\frac{1}{T} + \frac{Q(T)}{\int_0^T R(t)\,dt} \right] \tag{50}$$

$$C(t) = t \left[\frac{C_P}{T} + C_F \frac{Q(T)}{\int_0^T R(t)\,dt} \right] \tag{51}$$

But the lower bound for λ_{AV} (T) can also be used. In most practical situations the numerical values of the two bounds are very close to each other. This type of parts replacement policy is used when maintenance schedules are based on machine operating time rather than on parts times. It is obviously cheaper to replace several parts at the same time T, although this may not be the optimum time for each part taken separately. Also the availability of a machine increases by such bulk replacement policy.

It follows thus that the preventive replacement times of parts should be optimized from the viewpoint of their effects on the maintenance cost and availability of the overall system in which they operate.

b) Machines

When a machine has a design reliability $R_M(t)$ given by Equation (16), or more generally (15), its operational reliability will be given at any time by the age of the parts in the machine. Upon entering service when new, let the cumulative reliability be

$$R_M(t) = \prod^{n} R_i(t) \tag{52}$$

If no parts replacements are made up to a time T_o, the reliability of the machine for an operating period from T_o to $T_o + t$ is

$$R_M(t, T_o) = \frac{\prod^{n} R_i(T_o + t)}{\prod^{n} R_i(T_o)} \tag{53}$$

where each part has an accumulated operating age T_o at $t = 0$. But if the machine fails after having operated for T_x hours, and k of the n parts are replaced by new parts, so that n-k old parts with an age T_x each are left in the machine, its conditional reliability at some subsequent time $T_o = T_x + t_x$ to operate for t hours becomes

$$R_M(t, T_o) = \prod^{n-k} \frac{R_i(T_o + t)}{R_i(T_o)} \cdot \prod^{k} \frac{R_j(T_o - T_x + t)}{R_j(T_o - T_x)} \tag{54}$$

where $T_o - T_x = t_x$ is the operating age of the k parts replaced at T_x.

In general, when an i^{th} part was last time replaced at T_{xi} counted from the time when the machine was new:

$$R_M(t, T_o) = \prod^{n} \frac{R_i(T_o - T_{xi} + t)}{R_i(T_o - T_{xi})} \tag{55}$$

Of importance in this equation is simply the operating age of each of the n parts, that is $t_{xi} = T_o - T_{xi}$ of the i^{th} part. Therefore, Equation (55) can be written as:

$$R_M(t) = \prod^{n} \frac{R_i(t_{xi} + t)}{R_i(t_{xi})} = \prod^{n} R_i(t, t_{xi}) \tag{56}$$

for any operating period t. The t_{xi} values are simply the ages of the parts at any arbitrary time $t = 0$. Equation (56) is the general form of the operational reliability of a machine at any age. It applies equally to repair maintenance only, as well as to a maintenance policy with preventive replacements. Of course the numerical values of $R_M(t)$ will be different in the two cases since under preventive maintenance the maximum possible age t_{xi} of certain parts in the machine is limited by the preventive maintenance periods T.

Let us assume machine maintenance policy under which s parts are replaced preventively every T_1 hours, k parts every T_2 hours and n-s-k are replaced only correctively if and when they fail. A workable model for numerical reliability evaluation can be based on the long-term average failure rates of the parts. The stabilized average failure rate of the machine will be approximately:

$$\lambda_M = \sum^{n-k-s} \frac{1}{M_h} + \sum^{k} \frac{1}{M_i(T_i)} + \sum^{s} \frac{1}{M_j(T_2)} \tag{57}$$

where $M_h = \int_o^\infty R_h(t)\,dt$ for the h^{th} part of the n-k-s parts, while the terms 1/M(T) are obtained from Equation (49) using one of the bounds. An estimate of the long-term stabilized reliability is then given by

$$R_M(t) = e^{-\lambda_M(t)} \tag{58}$$

Obviously, the choice of T_1 and T_2 affects both reliability and the overall machine maintenance cost, specifically the cost of maintenance of the k + s parts which are subject to preventive maintenance. If the requirement for minimizing maintenance costs of these parts is made we can write a cost equation for the machine corresponding to Equation (47) for a single part:

$$C_M = \sum^{m-k-s} \frac{C_{hF}}{M_h} + \sum^{k} \left(\frac{C_{iP}}{T_1^1} + \frac{C_{iF}}{M_i(T_1)}\right) + \sum^{s} \left(\frac{C_{jP}}{T_2} + \frac{C_{jF}}{M_j(T_2)}\right) \tag{59}$$

The terms $\sum^k$ and $\sum^s$ can be plotted graphically each as a curve, $\sum^k$ as a function of a variable T_1, and $\sum^s$ as a function of a variable T_2. According to where the two curves show their minima, the values T_{1MIN} and T_{2MIN} are chosen to give minimum overall maintenance cost of the machine. However, it is usually possible to achieve a further saving by making T_2 an integer of T_1, so that T_2 becomes a major preventive overhaul, and T_1 a minor preventive overhaul. Again one must realize that a minimum cost preventive maintenance policy does not necessarily give a satisfactory operational reliability.

4. MAINTAINABILITY AND AVAILABILITY

Maintainability has been defined as the probability of completing maintenance in a given period of maintenance time. Being a probability, maintainability has an underlying density distribution of maintenance times, with a mean called mean time to maintain or mean down time. This mean down time is a most useful parameter for the evaluation of maintainability designed into a machine or equipment, prediction of maintenance manhours and estimation of machine availability over long service periods.

To perform these calculations two basic analyses must be performed on the machine design, a reliability analysis and a maintenance tasks time analysis. Each part in the machine has some mean time to failure of m_i hours and if it fails corrective maintenance must be performed on the machine, which on the average will take D_i hours. Over a long time t the part will fail $\frac{t}{m_i}$ times, on the average, and will thus cause a machine down time of tD_i/m_i. This is the part's average contribution to the machine's down time for every t operating hours. With n parts in the machine, the contribution of parts failures to the machine's down time becomes

$$t \sum^{n} \frac{D_i}{m_i} \tag{60}$$

or, on a long term average, the machine will be "down" for an average time of

$$D_h = \sum^{n} \lambda_i D_i \tag{61}$$

for each operating hour, on account of part failures. We have substituted here $1/m_i$ for λ_i, the long-term average failure rate.

If we now define availability as the maximum possible steady-state utilization factor of a machine, which is the ratio of up-time to the sum of up-time plus down-time, and consider first the case without preventive maintenance, availability becomes

$$A = \frac{1}{1 + \sum^{n} \lambda_i D_i} = \frac{1}{1 + D_h} \tag{62}$$

since for an up-time of $t = 1$ the machine will be down for at least $D_h = \sum^{n} \lambda_i D_i$, on the average. This sum D_h is the "per hour down time" of the machine in the sense that in a time $1 + D_h$ the machine is capable, on the average, to be up for 1 hour and will be down for a time D_h. Thus over a long calendar period T, the machine would be capable of being up for a time of approximately

$$t_{up} = \frac{T}{1 + D_h} \tag{63}$$

and will be down for a time of approximately

$$t_{down} = \frac{TD_h}{1 + D_h} \tag{64}$$

If we multiply in the availability equation (62) the numerator and the denominator by $M = 1/\sum^{n} \lambda_i$, the steady-state mean time between failures of the machine, we obtain

$$A = \frac{M}{M + M \sum^{n} \lambda_i D_i} = \frac{M}{M + MD_h} \tag{65}$$

where

$$D = M \sum^{n} \lambda_i D_i = MD_h \tag{66}$$

is called the mean down time of a machine without preventive maintenance. This is a very useful parameter in design evaluation for inherent long-term reliability, maintainability and availability, since in the form given by Equation (66) it contains a wealth of information from which all the other calculations can be handily performed.

When preventive maintenance is applied, the machine will behave differently in operational use. Let us assume a model such that of the machines $n = k + s$ parts, k parts are preventively maintained (replaced) every T_1 hours and s parts every T_2 hours. The i^{th} part of k category then assumes a steady-state average failure rate $\lambda_i = \lambda_i(T_1)$ and the j^{th} part of s category $\lambda_j = \lambda_j(T_2)$ where λ_i and λ_j are given by Equation (57) for the respective times T_1 and T_2.

The rate at which the machine will enter a down condition will be

$$r = \sum^{k} \lambda_i + \sum^{s} \lambda_j + \frac{1}{T_1} + \frac{1}{T_2} = \sum^{n} \lambda + \frac{1}{T_1} + \frac{1}{T_2} \tag{67}$$

if T_1 and T_2 never coincide. If coincidence occurs so that $T_2 = nT_1$, where n is an integer:

$$r = \sum^{n} \lambda + \frac{1}{T_1} \tag{68}$$

The reciprocal of r is the mean operate time of the machine

$$M_o = \frac{1}{r} \tag{69}$$

The per hour down time D_h is

$$D_h = \sum^{n} \lambda_i D_i + \frac{(n-1)\,D_m + D_M}{T_2} \tag{70}$$

where D_m is the average down time for a minor overhaul, D_M the average down time for a major overhaul and $T_2 = nT_1$.

Availability is then

$$A = \frac{1}{1 + D_h} \tag{71}$$

and machine mean down time becomes

$$D = M_o D_h \tag{72}$$

The minimum average total down time for a calendar time T is then obtained from Equation (64) using D_h from Equation (70).

The equations for r and D_h can be easily extended to more than two categories of overhauls.

5. CONCLUSIONS

The numerical evaluation of the reliability and maintainability parameters discussed in this paper will permit realistic estimation of maintenance manhours, spare parts supplies, total maintenance cost for long-term operations, and optimization of the preventive maintenance periods for minimum cost, predetermined operational reliability or operational availability. Above all, these techniques allow quantitative specification of reliability and maintenance requirements and analysis of designs.

It must be realized that basic information on failure rates and longevity of mechanical parts is not easily available, certainly not to the extent existing in the case of electronic parts. Therefore, much more weight will have to be placed on reliability demonstration tests to verify design reliability predictions. On the other hand, the technique of maintenance task diagrams showing computation of maintenance task times is much more readily applicable. However, this would not be the full answer, since these times must be weighted by the frequencies of their occurrence. When this is done, the complete story will emerge and optimum designs and optimum maintenance policies will be achieved.

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DISCUSSION

KECECIOGLU (U. of Arizona): I certainly appreciate all these thoughts. I hope some of the audience could form a picture of what was said. The report that Igor Bazovsky was the prime investigator on and entitled is certainly worth reading. I hope this will not pour in requests for copies of the report, but believe me gentlemen once you read it and penetrate into it, there is so much valuable information for every designer to use.

The important thing, however, is how to develop the data and this is the big problem and I'm sure Bazovsky agrees with us. However, we cannot hide our heads in the sand, and say the acquisition of this data is a terrific task. Therefore, to heck with all this theory, I'll do it as I've been doing it. If we do this, I think right there we could be dead as a nation. We can't afford that. We have to put every effort and emphasis to keep selling this idea. So we have to gather this data even though it might cost a little bit in the beginning. Once we acquired nationwide particularly in industry, you people in DOD push this, and press this idea on DOD, this data has got to be gotten.

You will be surprised gentlemen how it will accelerate the whole process of optimizing design at minimum cost at optimum reliability and availability.

Now the point that I do want to make is that most people talk of reliability as useful life reliability. I hope Mr. Bazovsky has allayed this thought, and that the totality of reliabilities is important, which is a product of the early, useful life and wearout reliabilities.

The electronics people have mislead us. The misleading is this, that every reliability analysis is of useful life reliability. You plug your failure rate (λ) and mission time (t) in R z $e^{-\lambda t}$ and you've got it.

Now this gentlemen, for mechanical components won't work, it couldn't work, and here is the value of Mr. Bazovsky's work in his paper today. When we talk of mechanical component systems reliability, we're talking about the product of primarily the useful and wear-out reliabilities. The next thing is that the frequency of repair maintenance, resulting from random failures, is much lower than that of preventive maintenance frequency.

Now the preventive maintenance is what you have to do by saying I will replace a part after so many hours of operation and before it fails, and this is so much more frequent. This is what adds to equipment costs. Preventive maintenance is determined by the wear-out failure distribution of the component, and here is where we need to have more reliability information.

Now let me ask this question. How can we put these techniques into more simple language and maybe in some sort of a handbook or a checklist? Can this be done, and say look this is what you can do even before you start designing your product and this is how you can optimize reliability and maintainability. How can we do it?

BAZOVSKY: We have here two problems: one is the acquisition of data, which applies primarily to existing equipment in field usage in the Navy. The reporting systems and data collection systems mentioned by Captain Kauffman and Commander Heenan earlier today would feed the data on existing equipment and would allow to optimize the maintenance cycles on the equipment to get more reliability, more availability or spend less on maintenance cost or whatever we are after. That would be a very useful and certainly rewarding thing to do, and I think even a necessary thing to do because of the tremendous cost of maintenance which now exists everywhere.

The second problem is that of new designs and for new designs. I think one of the first things to agree upon is how to design for a certain design life. For instance how to design a shaft so that it will last 10,000 hours.

We probably know how to design a bearing to last for 10,000 hours — at least we have handbooks which will show it — I think there are definite means of designing for a certain design life, but what has to be agreed upon is, what is design life, define it, and we have to define it in statistical terms, because 100% design life does not exist; do we agree on a 90% or 99% or 50% probability level. We have to agree somewhere on a probability level for design life.

I suggested the mean life as the 50% point, because that gives us a statistical parameter with which we can easily work.

FRANKEL (MIT): I would like to second your motion on the very valuable work and presentation by Dr. Bazovsky and would certainly recommend to anybody interested in reliability analysis for this work, as well as the book by Mr. Bazovsky, which I use as a text by the way. (I'm not working for Prentice-Hall.) I would like to mention a couple of things that have apparently slipped into our argument on all the work that has been done by the electronic engineers. Although I've heard recently that apparently there is no problem in the reliability of mechanical components, being a mechanical or marine engineer myself, I think we have very big problems.

One of my main arguments is, first of all, the carry-over of constant failure data or more or less assuming, of course, some distribution or chance failure.

Some of the experimental data obtained definitely shows that mechanical components chance failure rates are definitely time dependent and normally will require a 1st or 2nd order, at least a first order polynomial to represent a chance failure rate.

Now this very greatly affects the facility of the mathematical representation, because it forces us into other functions of other things and also unfortunately takes the mean time for failure which we so usefully employ in order to get an availability out of it because we don't have an exponential function anymore for reliability.

The second point I would like to make is the representation by Dr. Bazovsky of the series system. This means systems consisting of a number of components in series whose total reliability is the product of the individual component reliabilities. Now, considering individual components of a mechanical system, we unfortunately normally have to go into detailed analysis and define failure rate as a result of not just one event but several events, some interacting, others in series or parallel, causing the failure events. Apart from this, particularly aboard ships, we definitely have tried in the past and will probably in the future continue to increase our reliability by redundancy and thereby get a much more complex system than a simple series system which complicates mathematics tremendously, particularly if we are interested in getting a result of the reliability under constraints, for example, cost.

The third point I would like to make is preventive maintenance. We've talked a lot about corrective and preventive maintenance, always implying that preventive maintenance means renewal. Now, my past experience as an operating marine engineer definitely reminds me that a vast majority of preventive maintenance actually means maintenance repair, but not by renewal, and this, in particular, applies to major components of the marine system or marine power plant.

Now, we definitely cannot, after repair, assume that we again have the same MTBF, as reliability of the system is actually reduced. The time to wear-out is still a Gaussian distribution to which I agree. Also, wear-out time will definitely be reduced.

The last point I would like to make, and this is the point I've been harping on all the time and will continue to do so, is interaction — interaction of component and failure rate. We cannot treat mechanical systems like an electronic system where the failure rate of each component and the wear-rate of each component is completely independent and will not affect the performance and failure rate of the adjacent component.

Mechanical components, as we know very well, do not have independent components. If we take a simple example of a pump, a turbine, a motor, or whatever it may be, if one bearing wears due to some defect, it will affect the wear-rate of the second bearing, and obviously the performance of the part.

Now this is a very trivial example, but there are a multitude of examples, and we definitely have a dependence of failure rates and wear rates.

I would like to make a little remark in this connection. This is a very complex problem. We've just tried at MIT to solve the reliability problem of a single-cylinder diesel in order to find the interdependence of wear rates and failure rates and its consequent effect on the reliability, and we find that all these different components in this four component system definitely affect each other.

BAZOVSKY: You mentioned four things here as well as the assumption of the exponential, the assumption that everything was in series, then preventive maintenance, and then interaction.

It was a comment and I shall answer also with a comment to that. As to the exponential assumption we've seen that the failure process stabilizes with time when we have no preventive maintenance. After stabilization, about the same number of failures will occur in periods of equal length. Originally not; originally we have a long mean time to first failure, then a shorter and shorter, but finally the process stabilizes and becomes Poisson. From there on, in long term operations, we can use the exponential function to calculate the reliability of a complex system.

KECECIOGLU: But in the meantime the failure is not that of random, but it is that of the wear out.

BAZOVSKY: No, in the mean time the frequency of failures is a composite effect of the old parts with all the mixed ages of parts in the system.

KECECIOGLU: But the wear out life influencing more the mean time to failure of a system, would make the replacements of mixed age components look like random in nature.

BAZOVSKY: That's right. So the process becomes Poisson definitely. The same applies to a maintenance policy. Under the preventive maintenance policies, the process will again settle to a steady state condition, and failures will again occur in a Poisson fashion.

Now the assumption mentioned by Professor Frankel that everything acts in series in a machine that might be true, might not be true. In a simple machine, the elements are mostly in series, but you can build redundancy into machines. Maybe, I don't know. We build redundancy into electronic equipment, for instance. If I need a diode, I take 4 diodes and put them into what is called — a quad, and I have a terrific redundancy in that particular spot in the electronic machine.

Whether one can build redundancy into a simple mechanical machine, without making it complex, I don't know, into pumps for instance.

As to the last point, interaction between components I most whole heartedly agree with you that all probabilities are conditional probabilities. Non-conditional probabilities almost do not exist or very seldom you will find them. This also applies to electronic equipment because the performance of large electronic equipment depends on parts operating within tolerance, but the combinations of the drifts may be such that the whole equipment will be out of tolerance even though all parts operate within their tolerance limits. I think the same can apply also to mechanical design where all parts can still be within tolerance or within specification limits, but the performance of the machine may be out of specification due to an unfavorable combination of tolerances.

This brings us to the question of what do we define as failure. Is only a catastrophic type failure a failure, or is a certain degredation of performance also a failure? I have found there is no definite agreement on that point among people who operate machines, but they pull out the machine when they are dissatisfied with its degraded performance. That is a failure because they just cannot in their opinion continue operation with that machine, although the machine has not completely broken down.

DUNN (Electric Boat): I hope nobody is seriously considering a 50% survival point as a design parameter just in order to fit it into a failure rate method and statistical mathematics. This is a kind of ridiculous performance I would think. Most ship designers and I think that any vendor representatives here are already well convinced that they are designing for 3 sigma. Their success, and the success of their machines are based on approximately 3 sigma deviation of the variabilities that they use in the conduct of their designs.

I submit to this group and to you people that perhaps it would be a much more favorable beginning for reliability in ship design to consider success analysis rather than failure analysis. Our failures are relatively minor, and to create the statistics for failure rate based analyses we are going to have one hell of a job. Our successes are higher than our failures I believe after the important maintainability programs are well organized and the data is returned after the program is operating, we may well see that the only thing left for us to do is to improve the design details which means considering the allowable variability of every design parameter. Then we'll find our normal distribution envelope based on maintaining the variabilities of success within that envelope rather than looking outside the envelope for those random effects of failure.

BAZOVSKY: Sir, it really makes no difference which point you call the design life point. If you design to a 99.7 percent which is three sigma point, you still have to estimate your standard deviation or your variance and the mean, because this 99.7 percent point is a function of the mean and of the variance around it.

DUNN: I will agree with you, but I think that we as ship designers, and in our perhaps naivete in the technical reliability business, tend to think of the reliability more as a confidence factor than as a specific reliability number, and the three sigma tells us that we have a 99.7% confidence in the success of the device or the machinery which we are designing and building.

BAZOVSKY: That's right sir......

DUNN: I'm talking about the initial reliability that is created into the design by the designer not the subsequent probability of its degradation by production or usage.

BAZOVSKY: If you have a 99.7 percent, or a three sigma point for a thousand hours life in a part and you consider it as your design point, and if you design the part for say two thousand hours as a 50% survival point and you take the sigma around it and calculate the effect to the three sigma point you will come back to the same point depending on sigma. So it doesn't matter really what you call design life as long as we know what we are talking about and as long as we agree at some point. If we agree that it is three sigma point then fine, but at least we should get an agreement that the three sigma point is what we call the design life of the part. Then we must estimate the sigma, since we talk of a 3 sigma point that assumes we know the sigma. Then it is only easy arithmetic to find out what the M is and with the M we can perform all probability calculations. To perform probability calculations on the normal distributions, we must have the two parameters, the mean and the sigma. But what we call design life is not so important as long as we know what it is supposed to be, and also as long as we know the methods of how to design for a certain required life.

Thank you.

KARASSIK: Mr. Bazovsky made a masterful verbal presentation of a highly technical, highly mathematical subject and gained the respect of his audience by the remarkable feat of using his mathematics strictly as the background material for a clarification of many concepts in the field of reliability.

We can all take to heart the premises brought out by Mr. Bazovsky. At least I, for one, subscribe fully to the approach he has taken in his classification of "maintenance actions" and in his distinction between design and operational reliability. For there is more than mathematical formulae to be gleaned from this paper — there is a method of analysis and a clearing-up of semantics to be used when we speak of such concepts as "availability," "reliability" or "design life."

But I think that again the mathematicians have outdistanced the engineers.

They have developed for us the mathematics necessary to deal with problems which the engineer has not yet been able to define in precise terms. I refer specifically to two areas that contribute to our "ignorance factor" in regards to failure identification.

The first of these two areas involves our inability to produce a sufficiently large "population" for our mathematics to operate upon. We cannot tabulate a sufficient number of incidents from which to develop mathematically an availability or a reliability factor that is reasonably representative. It is easy to produce such statistics when we deal with electronic components that are manufactured by the thousands and the hundreds of thousands. The same is true in the automotive field where millions of bearings, axle shafts, crankshafts, gears or valves are involved. It is far less satisfactory to refer to statistics based on the record of a dozen or two dozen pumps of a particular design.

The second area deals with the validity of statistical reporting of failures and of the identification of failure causes. I refer more specifically to the reporting of either infant mortality or random failures than to failures through wear-out, as the latter are more easily identifiable and give less rise to controversy. Infant mortality and random failures can be caused as readily by circumstances external to the equipment in question as by design or execution deficiencies — even more readily in many cases. Yet, if as another speaker reported today less than 20% of the total number of failures are reported in answer to a survey, what makes us so sure that these 20% will provide us with the same distribution of causes as would become apparent from an analysis of total reporting? Human nature being what it is, I am tempted to question the identity of the distribution based on the partial reports with that of the total actual cases. It has once been said that statistics are very much like a bikini suit: what they reveal is interesting; what they conceal is vital. This certainly applies to the situation under discussion here.

PROCESS FLUID LUBRICATION TO ACHIEVE RELIABILITY THROUGH SIMPLIFICATION

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INTRODUCTION

The term "Process Fluid Lubrication" is not new, in fact, history shows that early civilizations employed it in their equipment — water lubrication was used in water wheels, ship rudders, etc. What is surprising, however, is its limited use up to the twentieth century. Today "Process Fluid Lubrication," even though perhaps not recognized by this term, is being extensively reapplied. From the point of view of good design, this trend is long overdue and its rapid growth is to be expected for it considerably simplifies the machinery system and makes it more reliable.

Process Fluid Lubrication is the utilization of the process fluid or surrounding fluid to perform the lubrication function. The lubrication process may be hydrodynamic, hydrostatic or hybrid. Common examples of Process Fluid Lubrication are:

Water Lubrication — Feed water pumps, deep well pumps, canned motor pumps, stern tube bearings
Two Phase Flow Lubrication — Saturated steam or vapor liquid metal turbomachinery
Steam Lubrication — Steam turbines
Gas Lubrication — Gas turbomachinery, gas bearing gyros and instruments
Gasoline Lubrication — Gas station pumps
Liquid Metal Lubrication — Rankin Cycle Space Power Machinery (SNAP)
Freon Lubrication — Freon compressors

Machinery failures are caused by environmental and operating stresses. The former causes deterioration failure, vibration, shock acceleration, which in turn can cause chance failures, e.g. fatigue, fracture, etc. The latter causes both chance and wearout failures, e.g. friction and wear, pressure stress, thermal stress, vibration, etc.

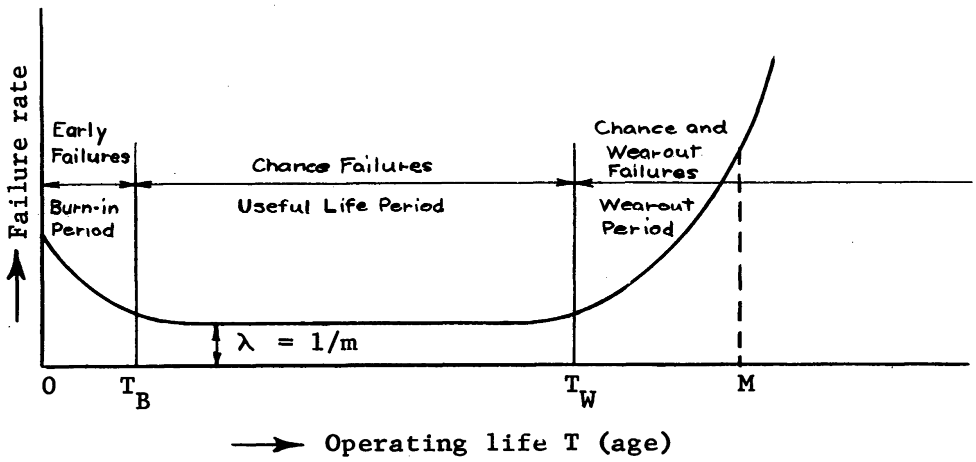
When one plots failure rate vs. time, Fig. 1*, one generally sees three distinct periods:

(1) Burn or run-in period
(2) Useful life period
(3) Wearout period

Each of these periods has a different failure rate. The first and the last have high failure rates. High reliability is obtained during the useful life period. (Reliability is the probability of a device performing its purpose adequately for the period of time intended under the operating conditions encountered.)

A review of machinery failures indicates that dirt, marginal or improper

*Redrawn from Ref. 1 - Fig. 5.1.



m = mean time between failure
T_B = burn-in period
T_W = wearout period begins
M = mean wearout life

Figure 1.

lubrication are the greatest deterrents to high reliability. Such failures as wear, high friction, excessive temperature, thermal stresses, contamination, stress corrosion, excessive vibration, noise, etc. are most common with a poor lubrication system. Seals are the next greatest contribution to machinery unreliability. The failure rate and types of failures can be considerably reduced by use of Process Fluid Lubrication.

This paper presents some of the advantages of Process Fluid Lubrication as applied to some examples of naval machinery. It then points out how the reliability of such machinery is improved through simplification and integration.

DISCUSSION

All system reliability calculations have one thing in common which is fundamental to probability calculus: the reliability of the basic components which form the system must be known. These component reliabilities must be determined from design analysis or test, giving basic probability information in terms of failure rates, mean time between failures, or mean number of cycles between failures. Since a chain is as strong as its weakest link, the reliability of each individual component must be very much higher than the required system reliability:

$$R_{system} = R_1 R_2 R_3 R_4 \ldots\ldots R_i$$

Thus, from the standpoint of high reliability and low maintenance, it is good design practice to integrate and simplify the components or subsystems and to use as few parts as possible. This rule and several others listed below should be employed to insure high reliability equipment. Through the use of Process Fluid Lubrication several of the recommendations given below can be met.

(1) Simplify the design to a minimum of parts without degrading performance
(2) Perform reliability reviews during the design stages
(3) Apply component derating techniques to the best possible advantage to reduce failure rates and to increase component life
(4) Eliminate dirt and contamination from the system
(5) Reduce the thermal gradients in order to avoid thermal stress, misalignment and distortion problems

(6) Reduce the friction losses by proper hydrodynamic or hybrid lubrication
(7) Use proper rotor balance and adequate damping to reduce the stress level at resonant frequencies
(8) Reduce space length and thus stress level by elimination of seals, labyrinths, etc.
(9) Specify component reliability and run-in period required
(10) Specify prototype tests and production equipment debugging procedure.

Recent studies of naval machinery, Ref. 2, e.g. feed water pump, etc. have focused attention on using a process fluid as lubricant. Clearly, the use of process fluid in place of oil has many advantages, among them:

(1) Elimination of the lube oil system, and hence, the maintenance and supervision needs associated with it
(2) Elimination of a fire hazard
(3) Elimination of such noise generating elements as lube oil pumps
(4) Reduction in weight and length of machinery resulting from elimination of oil seals, coolers, pumps, etc.
(5) Reduction of thermal stress problems.

Some of the process fluids available in the engine room are steam at or near saturation and water. Lubrication with either of these fluids has to be considered. Ideally, in order to minimize external piping and the problem of mixing of the fluids, steam lubrication should be used to lubricate machinery where steam is the process fluid while water lubrication should be used where water is the process fluid.

In regard to steam lubrication, two choices are available. The first of these is to super heat the steam, ahead of bearing inlet, sufficiently to insure that there will be no condensation anywhere in the bearing system. Alternately, the steam can be furnished to the bearings in its saturated condition. The second alternative avoids the use of auxiliary heating of high pressure steam. However, it introduces the additional problems of two phase flow, condensation and evaporation in the bearing and restrictor passages.

Except for the material problem, lubrication with dry steam throughout is analogous to gas lubrication of which there are numerous examples, Refs. 3 and 4. To the best of the author's knowledge, however, there is no applicable experience with two phase, steam-water, lubrication. ONR is presently sponsoring two fundamental studies, one of these studies at Rocketdyne deals with self-acting bearings while the other at Mechanical Technology Inc. deals with externally pressurized bearings, Refs. 5 and 6.

At present, several applications are available with water lubrication, Refs. 7 - 12. However, very little data is available on the effect of eccentricity and axial flow on turbulence, cavitation or boiling in the low pressure region of the bearing. Since these are generally the problems associated with low viscosity and high vapor pressure fluids, analysis and simple experiments are necessary.

Lastly, the materials used for either steam or water are different from those for oil lubrication, Ref. 13. New problems arise such as: erosion and corrosion of bearing surfaces and restrictors; dimensional stability; physical properties; compatibility of mating materials from standpoints of thermal expansion and wear; shock resistance.

This indicates that, while Process Fluid Lubrication is advantageous, there still exists a number of development problems which must be solved in order to make the application possible. The Appendix discusses these problems in some more detail. In the next section three examples are given of the application of process fluid to naval machinery.

EXAMPLES

In a recent study of low maintenance machinery for submarine power plants, Contract Nonr 3485(00), several power plants were designed which incorporate Process Fluid Lubrication. Three examples are given here.

The use of Process Fluid Lubrication brings out the following advantages which are common to all three systems.

(1) Elimination of
 (a) lube oil systems, e.g. pumps, coolers, filters, regulators, etc.
 (b) seals and labyrinth (a low reliability component)
 (c) fire hazard
 (d) oil fouling and contamination
(2) Reduction of
 (a) machine length
 (b) machine weight
 (c) noise level generated by pumps, coolers, etc.
 (d) thermal gradients (the cold bearing adjacent to hot turbine is replaced by steam bearing)
 (e) distortion and misalignment
 (f) dirt, fouling and contaminates in the system
(3) Increased overall efficiency

We will next discuss the three systems:

(1) 15,000 HP Power Plant (See Fig. 2)
The figure indicates a steam reversing turbine, nine stages forward and four stages reverse coupled to a generator. (No astern turbine would be used with a generator but it does show a package which would be applicable to drive a propeller.) The generator could also be replaced by hydrostatic transmission. The following simplification and integration of components was possible:
 (a) Steam and/or water lubrication has been shown for the shaft bearings, thereby simplifying the lubrication system, eliminating seals, and decreasing the number of bearings from four to three.
 (b) If the inboard bearing is water lubricated, the water can be used for cooling the exciter and generator rotor and stator. The stator windings can be cooled by using hollow conductors through which the coolant is circulated. By water cooling the generator, its weight can be reduced 25 - 50 percent below that of a normally air cooled generator.
 (c) The exciter and generator are of sealed or canned construction which permits their consolidation into the condenser vacuum chamber adjacent to the water-lubricated bearings. By this construction, rotor cooling is more easily assured and windage losses are reduced.
 (d) By using the boiler feed water as the coolant, the rejected heat can be reduced to the steam system to improve the cycle efficiency.
 (e) The boiler feed pump is attached to the hot well and becomes an integral part of the packaged power plant. The pump could be designed for supplying the water lubricant, generator coolant, and the fluid power for the governor servo system, as well as boiler feed water service.

Aside from the simplification and integration the following additional advantages result when compared to a conventional power plant.
 (a) Percent reduction in weight — 30 percent
 (b) Percent reduction in length — 28 percent
 (c) Percent increase in efficiency — 3/4 percent
 (d) Percent increase in mean time between failure — 63 percent

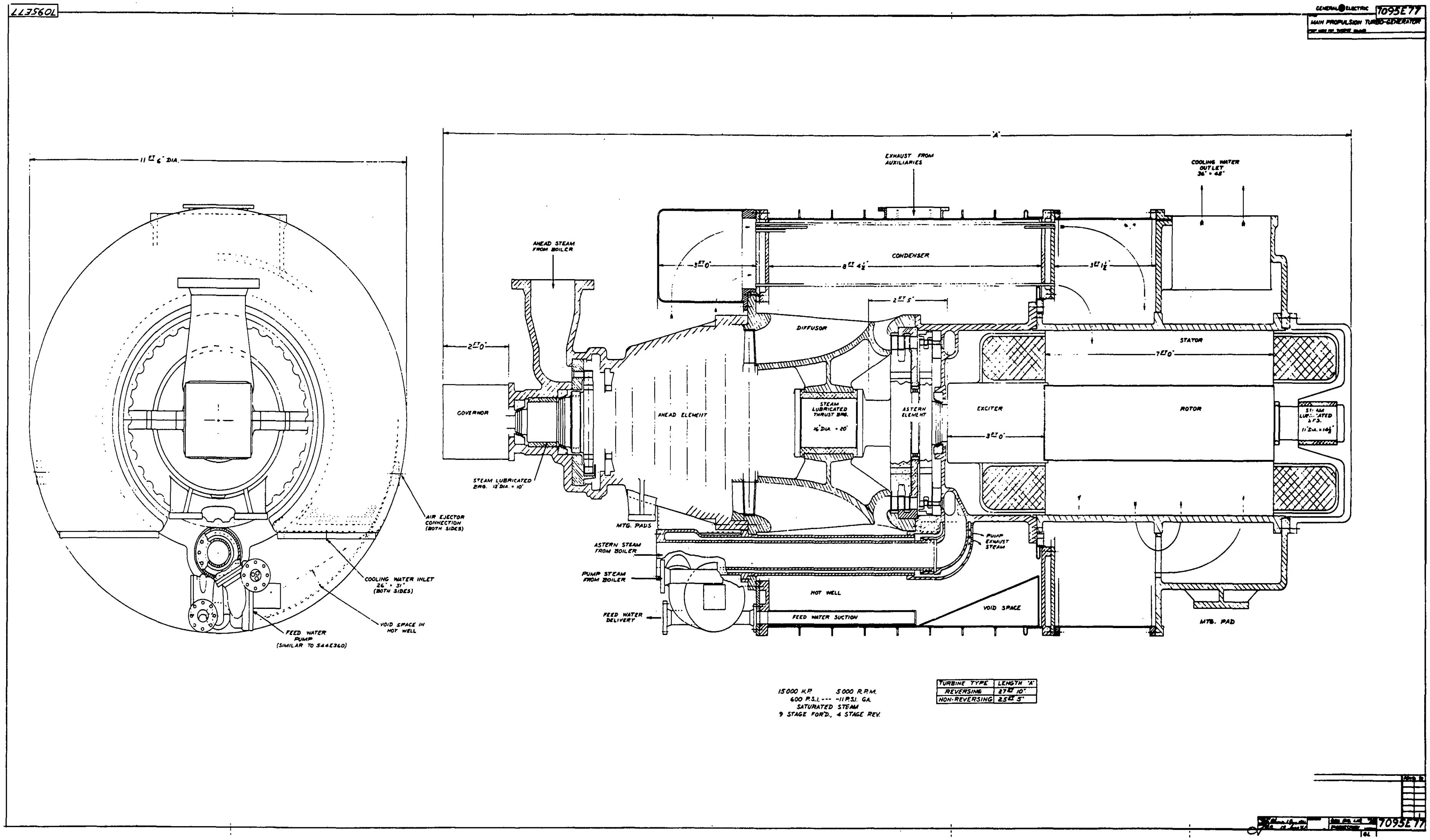

MAIN PROPULSION TURBO-GENERATOR
7095E77
11 FT 6" DIA.
'A'
EXHAUST FROM AUXILIARIES
COOLING WATER OUTLET 36" × 48"
AHEAD STEAM FROM BOILER
CONDENSER
DIFFUSOR
STATOR
GOVERNOR
AHEAD ELEMENT
STEAM LUBRICATED THRUST BRG.
ASTERN ELEMENT
EXCITER
ROTOR
STEAM LUBRICATED BRG. 12" DIA. × 10"
MTG. PADS
ASTERN STEAM FROM BOILER
PUMP STEAM FROM BOILER
FEED WATER DELIVERY
PUMP EXHAUST STEAM
HOT WELL
FEED WATER SUCTION
VOID SPACE
MTG. PAD
AIR EJECTOR CONNECTION (BOTH SIDES)
COOLING WATER INLET 26" × 31" (BOTH SIDES)
VOID SPACE IN HOT WELL
FEED WATER PUMP (SIMILAR TO 544E360)
15000 H.P. 5000 R.P.M.
600 P.S.I. --- -11 P.S.I. GA.
SATURATED STEAM
9 STAGE FOR'D., 4 STAGE REV.
TURBINE TYPE LENGTH 'A'
REVERSING 27 FT 10"
NON-REVERSING 25 FT 5"

Figure 2.

It is obvious from the above discussion that through this integration to simplification greater reliability is possible. The possible gains, therefore, justify the required development in Process Fluid Lubrication.

(2) Feed Water Pump (See Fig. 3)
The figure indicates a two stage turbine driving a three stage centrifugal boiler feed pump. This machine would have water lubricated pump bearing and steam lubricated turbine bearing. The stages are designed for high head and they supply the boiler, bearing and governor.

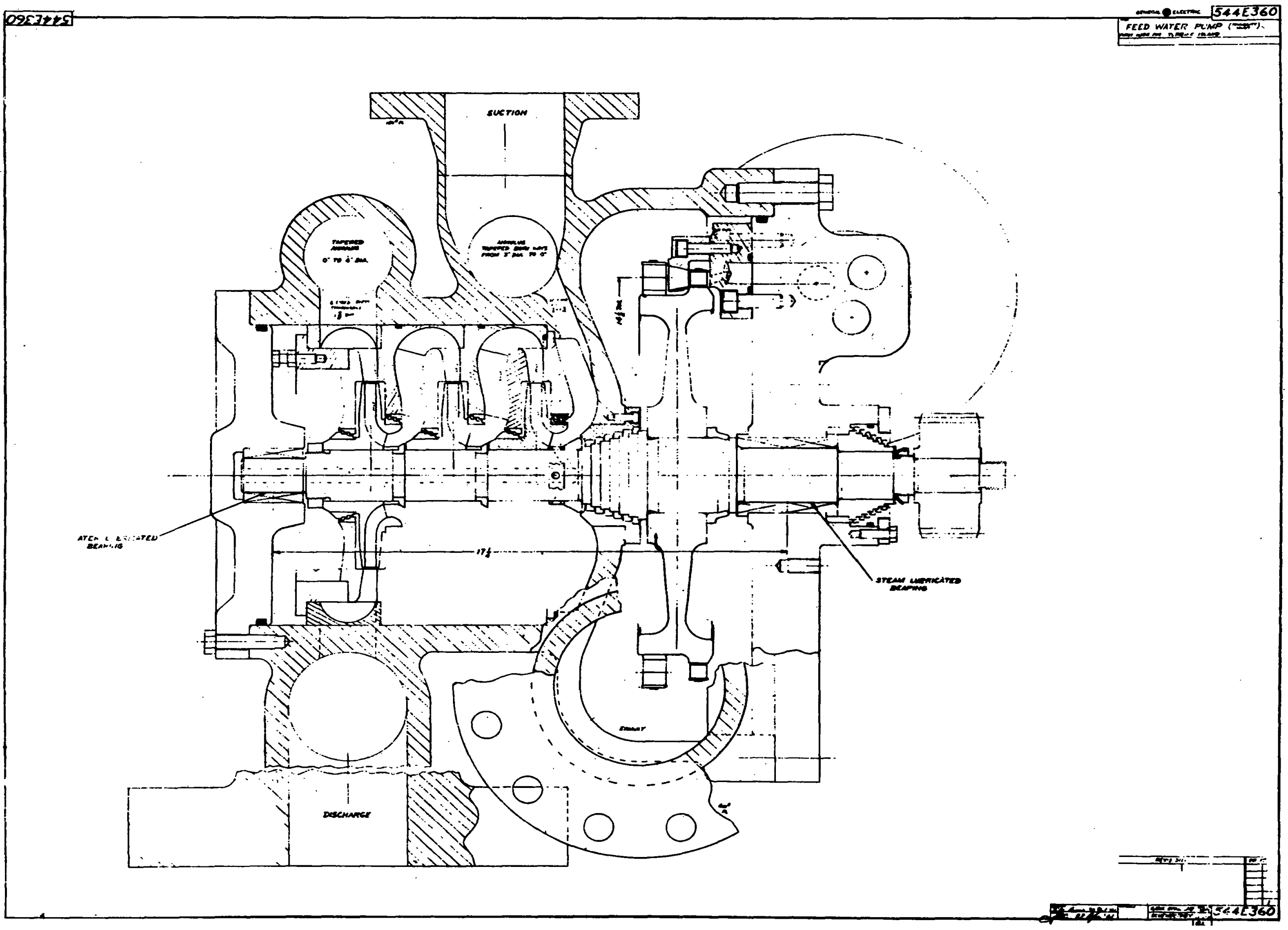


Figure 3.

In the case of a constant speed main turbine, the boiler feed pump could be driven from the main turbine with a booster pump in the hot well, whereas a variable speed turbine would use the turbo-pump design as indicated in Fig. 3.
This design has the following advantages when compared to a conventional feed water pump.
(a) Percent reduction in weight — 25 percent
(b) Percent reduction in length — 20 percent
(c) Percent increase in efficiency — 2-1/2 percent
(d) Percent increase in mean time between failures — 80 percent

(3) 2000 KW Turbo-Generator (See Fig. 4)
This unit, like the feed water pump, uses one water lubricated bearing and one steam lubricated bearing. The turbine end uses steam while the generator end uses water. Both ends of the machine can be capped off in this way minimizing chances for dirt or contaminates to get in.

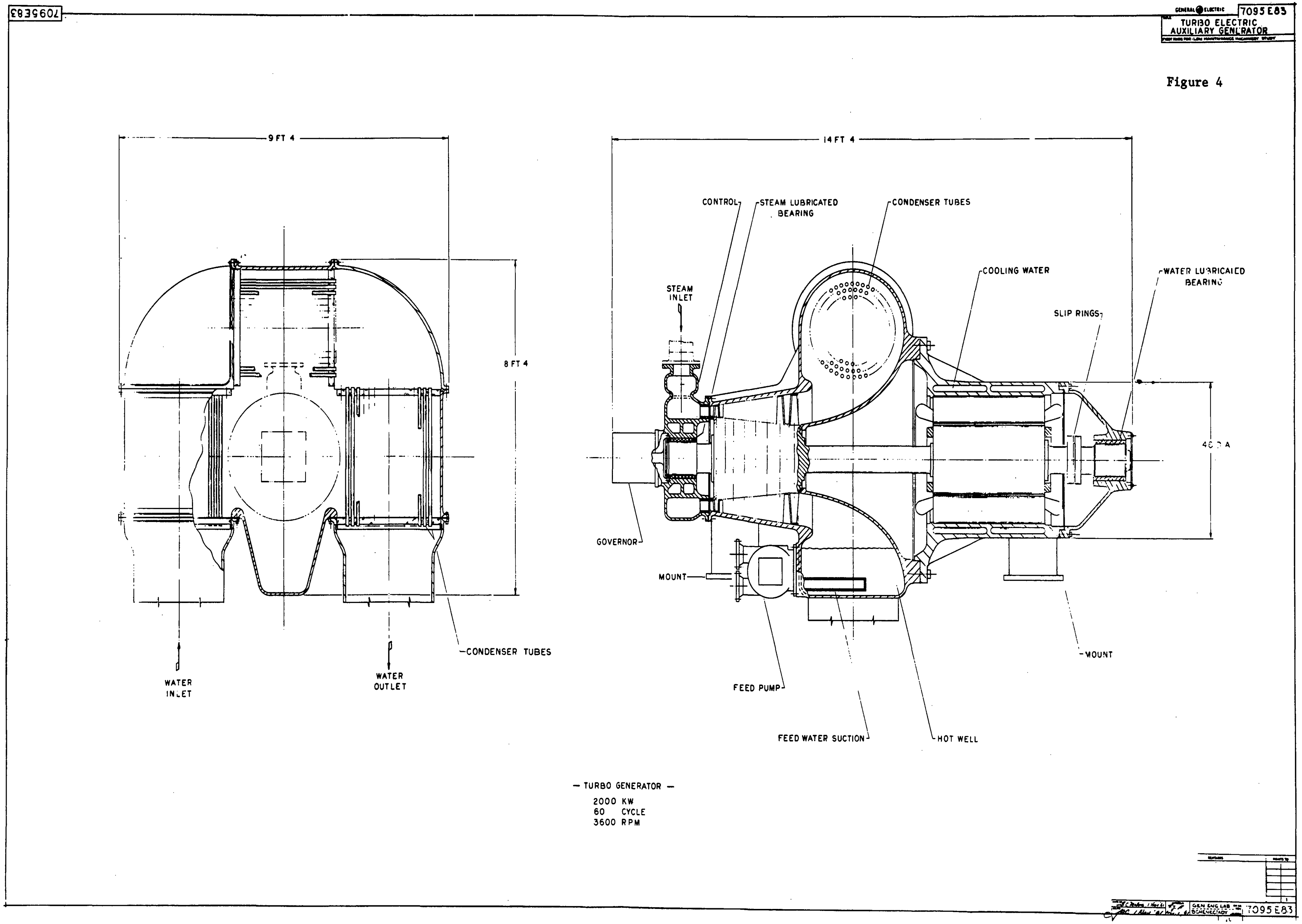

7095E83
GENERAL ELECTRIC
TURBO ELECTRIC AUXILIARY GENERATOR
Figure 4
9 FT 4
8 FT 4
14 FT 4
WATER INLET
WATER OUTLET
CONDENSER TUBES
CONTROL
STEAM LUBRICATED BEARING
CONDENSER TUBES
COOLING WATER
WATER LUBRICATED BEARING
SLIP RINGS
STEAM INLET
GOVERNOR
MOUNT
FEED PUMP
FEED WATER SUCTION
HOT WELL
MOUNT
— TURBO GENERATOR —
2000 KW
60 CYCLE
3600 RPM
GEN ENG LAB
SCHENECTADY
7095E83

Figure 4.

This design has the following advantages when compared to a conventional auxiliary turbo-generator.

(a) Percent reduction in weight — 27 percent
(b) Percent reduction in length — 23 percent
(c) Percent increase in efficiency — 2 percent
(d) Percent increase in mean time between failures — 45 percent

CONCLUSIONS AND RECOMMENDATIONS

(1) The use of Process Fluid Lubrication greatly simplifies machinery and in this way raises reliability.
(2) The reliability of individual components must be determined in order to establish system reliability. The individual component reliability must be higher than system reliability.
(3) Design and evaluation of new machinery should keep reliability aspects in mind.
(4) The possible gains with Process Fluid Lubrication justify the required developmental effort.
(5) The laws governing changes of failure rates with such parameters as temperature, static and dynamic stress, friction, surface damage, etc. should be determined.

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APPENDIX

TWO-PHASE LUBRICATION

Introduction

Vapor lubrication problems are different from those of gas lubrication primarily in the possibility for a two-phase lubricant film to exist. Circumstances related to the occurrence of a two-phase film, as well as its structure and the consequences thereof, are likely to be vastly different, depending whether the bearing is of the self-acting or the externally pressurized type; the choice of which, in turn, depends on the nature of the application. For advanced marine propulsion and space power systems, the advantages of process-fluid lubrication using vapor bearing has increasingly attracted attention. In these cases load carrying requirement usually dictates the use of externally pressurized bearings.

Discussion

The relatively well developed gas lubrication theory is directly applicable to the case of a vapor lubricant so long as it remains in the gaseous phase. The slight deviation of the thermodynamics from the perfect gas law, due to the proximity of the vapor to its saturation condition, is not likely to cause any drastic phenomena so far as lubrication is concerned. However, as soon as condensation takes place, the situation then undergoes major changes. Experimental evidence of this has already been reported in Ref. A1 for air bearings operating in an atmosphere of moderately high relative humidity.

The consequence of condensation in a vapor bearing is two fold. In the first place, due to the release of latent heat during condensation, the usual assumption of an isothermal film in gas lubrication theory probably fails. Secondly, in regions where condensate is present, various different modes of the mixed phase can exist; in the extremes, e.g. either the two phases coexist in separate layers or they could form a foam of fine structure (Ref. A2). In any event, theoretical treatment in the region where condensation has taken place would have to be different from that of gas lubrication.

For externally pressurized bearings, condensation also affects the operation of bearings in ways not normally considered as lubrication problems. For example, if a choked orifice is used as a restrictor and if the supply vapor is only slightly superheated, then along the center line of the orifice, the vapor expands isentropically and droplets of condensate will form in the stream. These droplets can cause serious erosion damages to both the orifice and the journal or runner. Thus the restrictor design is a crucial problem for externally pressurized vapor bearings.

WATER LUBRICATION

Introduction

Both hydrodynamic and hydrostatic water lubrication are applicable to Naval equipment. Selection of one or the other, or of hybrid (i.e. combined hydrostatic-hydrodynamic) lubrication, in each specific instance will depend on its load-speed relationship. The same well established relations between:

load
speed
bearing geometry
lubricant inlet conditions
lubricant viscosity

on the one hand and

film thickness
film temperature
film stiffness and damping
flow and power loss

on the other, hold equally well for water as for oil and may be used for selection of mode of lubrication and for bearing design.

The need for development work to achieve practical water lubrication for marine application arises from the differences in the properties of the two fluids. Some of these properties such as the poor boundary lubrication properties of water and its corrosive action on many common bearing materials, necessitates a material study. The low kinematic viscosity and relatively high vapor pressures may introduce turbulence and cavitation problems.

Water has a low viscosity, approximately 1/65th that of 2190T oil at 130 F. Load capacity of water lubricated bearings is, therefore, correspondingly lower. Resort to external pressurization must be made at smaller values of unit loading than is the case with oil. In addition, turbulence in the fluid film occurs at lower speeds because of the much lower kinematic viscosity. Under laminar flow conditions bearing water films have also less stiffness and damping than oil film so that the threshold of instability and rotor critical speeds occur at lower speeds. Finally squeeze film capacity and hence resistance to shock loads is lower than with oil.

Discussion

1. Turbulent flow in the bearing clearance space

Because of low kinematic viscosity, the flow in the clearance space of water lubricated bearings departs from laminar behavior at relatively low speeds. For example, G.I. Taylor's analysis indicates that a six inch diameter journal bearing with a clearance ratio of 0.0010"/" will enter the transition region between laminar and turbulent flows at about 1170 rpm (water at 130 F).

Some analytical and experimental work has been done on hydrodynamic bearing performance in the transition and turbulent regions (e.g. Refs. A3-7). It has been shown that in these regions load capacity and power loss are higher while flow is lower than in the laminar region. Most of this work, however, neglected either eccentricity or side leakage effects. Analytical extension of this work is needed in order to (a) include the effects of both eccentricity and side leakage on the point at which transition from laminar behavior starts and (b) to define the parameters which will be needed for future bearing design calculations and which can best be determined experimentally.

Turbulence is also a major consideration in hydrostatic bearing design. For example, a recently completed preliminary study of hydrostatic water lubrication of the Steamotor bearings (Ref. A9) showed that the bearing clearances had to be of the order of 0.0003"/" in order to preserve laminar flow. Presently available analysis of hydrostatic bearings (Refs. A10,11) needs to be extended to allow calculation of load capacity, supply pressure, flow and pumping power with turbulent flow in the clearance space.

2. Cavitation

Cavitation occurs in hydrodynamic bearings in regions where the pressure is less than the vapor pressure of the fluid. Since vapor pressure of water is higher than that of oil throughout the range of bearing operating temperatures, cavitation is expected to occur over larger regions of the bearings. Temperature rise in the bearings will also be extremely critical since the vapor pressure rises sharply with increased temperature. Previous work on cavitation (e.g. Refs. A12-15) has shown it to be greatly influenced by the nature of the loading (steady state or dynamic) and by the method of lubricant admission (inlet holes, grooving, etc.) The results of this previous work should be reviewed to establish the importance of cavitation in lubrication with water and to determine the analytical and experimental work required for future, more complete investigation of this problem.

3. Pumping power for hydrostatic water lubrication

High pumping power requirements are a major obstacle to hydrostatic water lubrication. This was, for example, shown in the above mentioned study of the Steamotor bearings (Ref. A9).

Presently available analysis (Refs. A10,11) is applicable only to fully compensated journal bearings (i.e. where the restrictors are of equal size and uniformly distributed around the circumference). This analysis needs to be extended to partial arc admission as one means of reducing pumping power for cases where high unit loads occur only over a part of the bearing. This is quite often the case, e.g. for bearings under unidirectional load and where the high regions of the load cycle always act on the same bearing arcs, as in many reciprocating engine bearings.

MATERIALS

Introduction

Most power plants use a dual fluid system, water being the working or heat transfer fluid and oil the lubricating system. It would be advantageous to eliminate the oil and use water in both systems. However, one of the major obstacles to such an approach is the selection of a bearing material which would operate with water or steam as a lubricant. The problems associated with the selection of a water lubricated bearing material are similar to those for conventionally lubricated bearing materials with one major exception. In conventional systems, the oil reacts with the bearing material to form a lubricating film which lowers friction and prevents metal transfer during periods when the fluid film will not support the load. Water in most instances does not have this capacity. One must, therefore, seek self-lubricating materials in the broad sense of the term. However, the selected material must also have the other properties expected of any bearing material. It must resist the water or steam environment as to static and dynamic corrosion, swell and solubility, it must compensate for misalignments and tolerate dirt, and, above all, in the event of failure it must not damage the rotor. In addition to this, a number of other problems must be solved.

Past experience has shown that the properties essential to a good bearing material include the following:

1. Good compatibility and wear resistance when the bearing is sliding in contact with the shaft.
2. Satisfactory conformability and embeddability.
3. Corrosion and erosion resistance.
4. Good dimensional stability over the required temperature range.
5. Good compressive and fatigue strength.
6. Matched thermal expansion rates between the shaft and the bearings.
7. Good thermal conductivity.
8. Ease of fabrication and methods of retention.
9. Cost.

Even for conventional oil-lubricated bearings, this list imposes stringent requirements on potential materials. For example, many aluminum alloys and bearing bronzes are far superior to babbitt in strength and thermal properties. In spite of this, neither of these materials has been found to be a suitable universal replacement for babbitt. Although cost is certainly a factor, wider successful application has mainly been handicapped by one or more of the following problems: poor compatibility and inability to conform readily under misaligned conditions or to tolerate debris. For this particular application, where the bearing must operate in a water or steam environment, corrosion and erosion resistance will also be a major factor.

This background indicates that the primary effort should be directed toward the solution of these three problem areas. This is not meant to imply that other properties are not important, but rather that these are the problems which must be faced initially. Once these have been reconciled, then consideration can be given to means whereby the materials can be modified or the designs can be directed toward compensating for the other factors.

Discussion

Background on Material Selection

In the introduction, three properties were selected as being of primary importance in choosing bearing and shaft combinations for water and steam lubrication. These included:

1. Compatibility and wear resistance.
2. Satisfactory conformability and embeddability.
3. Corrosion and erosion resistance.

The background on these problem areas is discussed briefly in the following sections:

Compatibility and Wear Resistance

Even in a well designed hydrostatic bearing, there will be many periods when the shaft may slide in contact with the bearing surface. During periods of dynamic loading or instability, contact can occur under a variety of load and speed conditions. If the bearing and shaft are a compatible combination, no surface damage should occur at these times and hydrodynamic lubrication will be restored as soon as conditions become favorable. Friction and wear are also important considerations since the friction will be a major factor in determining the rate of heat generation and wear will affect the bearing clearances. Nevertheless, surface damage is still the most important aspect.

Several empirical criteria have been used for the selection of compatible materials. These include:

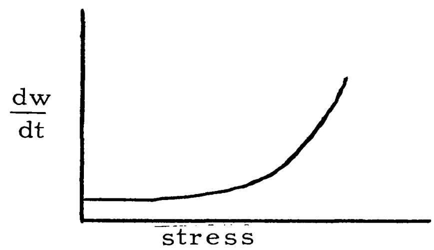
1. Materials which form a soft, low shear strength film on the bearing surfaces. This film may be the result of a reaction between the bearing material and the environment, or it may be due to the extrusion of a soft component from the bearing structure, or it may be formed by a reaction at the contacting asperities of the bearing and shaft.
2. Non-soluble material combinations or combinations which form brittle intermetallic compounds.
3. Hard materials which minimize the area of contact. These could be solid members or surface coatings.
4. Materials of particular crystal structure. For example, it has been observed that many alloys with a hexagonal structure are good sliding combinations. Cobalt sliding on cobalt is a good illustration of this. However, there are also many exceptions to this rule.

All of these criteria are based on one premise: if the adhesive junctions formed between two contacting surfaces are weaker than the cohesive strength of either one of the materials, shear will take place at the interface with little or no change in surface geometry. This will determine a compatible combination.

Although these criteria are useful in selecting potential bearing materials, it is important to note that the operating conditions also play an important role in determining material effectiveness. A given material may be suitable for use in a particular load and speed range, but any conditions which cause this range to be exceeded will result in a rapid temperature rise and subsequent failure. At high sliding velocities, the surface temperature will increase linearly with friction and as the (velocity)$^{1/2}$ (load)$^{1/4}$. Extremely high temperatures can be reached in a very short time. This is the physical basis for the widely used PV factor.

Satisfactory Conformability and Embeddability

One of the most important properties of a bearing material is the ability to tolerate a certain amount of misalignment. This could be the result of a number of factors such as manufacturing tolerances, thermal distortion, sudden changes in the load vector, etc. Were it not for the fact that misalignment is encountered in almost every application, then very hard, rigid bearing materials would be used almost universally. A soft material with a low modulus of elasticity, such as babbitt, will deform to an optimum profile readily. Wear can also compensate for misalignment to some extent. Ideally, a material should exhibit a wear rate behavior as shown below:



Above a critical stress, the wear rate should become rapid enough to reduce the pressure and prevent a sudden temperature rise. Resin bonded carbon-graphites usually conform by this mechanism. It is also possible to select material combinations which will react with steam when the energy supplied by the sliding process becomes excessively high. The soft reaction product will then wear away until suitable conditions have been restored.

Embeddability is also a strong factor in the selection of materials. In almost any practical machinery, allowance must be made for wear debris or foreign dirt particles which can damage the geometry of the bearing surfaces. Babbitt has demonstrated a high degree of effectiveness in meeting this criterion. Elastomers are even more effective in tolerating the presence of abrasive particles and are generally used for applications where it is necessary to handle slurries containing excessive amounts of solids. For an application such as a steam lubricated bearing, it would probably be necessary to use a two phase or two component bearing system to maintain high temperature strength and still accommodate a certain amount of debris.

The requirements of embeddability and conformability point up the need for using materials having a low modulus of elasticity wherever possible.

Corrosion and Erosion Resistance

Of these two properties, it is anticipated that erosion, especially in steam lubricated hydrostatic bearings, will be the worst problem. There are many materials capable of withstanding static exposure to water or steam environments. However, in this application the water or steam must flow through very small orifices at velocities on the order of 100 to 300 feet per second. In addition, this high velocity stream will be impinging on the shaft surface. This will require that both the orifices and the shaft be made of very hard, corrosion resistant materials. These may be solid members or suitable materials with surface coatings.

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DISCUSSION

BAZOVSKY (Raytheon): Dr. Sternlicht, you mentioned this double bearing gas and cold bearing combined, and you referred to it as a redundant bearing. In my opinion this is not what we understand under redundancy. In reliability redundancy means that if one redundant part fails then the other is capable of continuing operation on its own. Is this the case in this bearing that if either the gas bearing or the ball bearing would fail, the operation would not have to be discontinued?

STERNLICHT: That is correct and this is the reason why I called it redundant. This is a case where if either one fails; the other one can still operate. Of course, this depends on the mode of failure. For instance, let's assume that the friction in the ball bearing is too high, and the bearing locks in. The machine can still operate on the gas bearing.

On the other hand let us assume that the gas bearing deteriorates and its friction increases due to wear. Then eventually the friction in the ball bearing will be lower than that of the gas bearing and the equipment will operate on the ball bearing. This is a redundant system in which the system reliability is greater than individual component reliability. This is an excellent example of an exception to a rule on reliability.

SALING (Pratt & Whitney): Unless I misunderstood what you said, a comment was made that I have heard a number of times and feel requires clarification. The statement was made that reliability analysis in design should be considered a design tool and nothing more. This would seem to me to be putting our heads in the sand as far as an overall systems integrator is concerned.

For instance, each component of a missile system plays a very definite part in the overall system's reliability. Therefore, those responsible for this overall reliability must, of necessity, be able to determine and control the reliability of these missile components; they must be able to know exactly what they are receiving. In this respect, reliability analysis is a very valuable tool.

STERNLICHT: There is no question that reliability is a very powerful tool in system analysis and design, and I believe that all designers and purchasers of equipment should understand the principles of reliability analysis. On the other hand the thing I was trying to stress is that the reliability of a product depends on the designer who has to make sure that he understands the factors which influence the modes of failure in his specific design. Such factors as: stress level; maximum temperature and how it affects the performance of the component or the part; stress reversals and how they influence such things as fatigue failures; wear mechanism and how it takes place. I believe that it is extremely important for the design engineer to use all of the available tools and without any question reliability is one of them. On the other hand it should be recognized that an understanding of reliability per se does not insure reliable design. A good designer must know engineering fundamentals of heat transfer, stress, analysis, vibration and lubrication. He must be creative, must have good knowledge of materials and their behavior as influenced by external and internal body forces, etc.

COUTINHO (Grumman Aircraft): Mr. Fixman mentioned this morning that there are three ways of looking at reliability: economic, safety of life, and from the national security point of view. In the aircraft business, we have in the past used a slightly different term for reliability, namely: airworthiness. I like this word because it emphasizes the safety-of-life aspects.

Mr. Dunn from Electric Boat referred to success analysis, but I think that first you have to recognize what area you are in: whether it is economic, or safety of life, or national security. In the safety of life area, there can be no compromise with reliability or airworthiness. Before anything else is considered, the equipment must be airworthy. If it is not airworthy, there is no point in discussing it, it is not going to be built.

People will not fly in an airplane if there is any admitted probability of failure. Reliability control methods must be adapted to the problem at hand. Mr. Bazovsky said that chance failures depend upon safety factors. This is true only to a degree. We must find other ways of eliminating failures, other than by adding weight. Large turbines for power companies run for 20 years with little support and no breakdowns, but they are great big machines that would fill this room.

As soon as problems of space and weight must be considered, such as on a ship or an airplane, it becomes necessary to make some trade-offs, and this is where reliability control techniques can help. These methods are not absolute. I like to say that in the past we have not been able to design an airplane or a ship without algebra and without descriptive geometry, and now we can't design these systems without statistical analysis; but it's not the algebra and the descriptive geometry that does the work. It's the designer who is using these techniques as tools; and he has to use them to express his ideas. This seems to be the thing that we are learning today. The way to achieve greater reliability is through the ingenuity, the ideas, and the inventions of the designer. Statistical analysis and reliability control techniques can help us to do a better job, but the techniques by themselves have no power to increase reliability.

KECECIOGLU (U. of Arizona): Question number one: you put on the board, for various applications of this system lubrication, values of improvement in reliability in percent. The question is how were these figures arrived at? Question number two: I'm a mechanical engineer and I've seen system fluid lubricated bearings. I know the problems that are involved, particularly sustaining loads. In some cases you have a gas as the support fluid. Would you care to comment about the design problems in incorporating fluid support in bearings, the stability of these bearings and the relative size of these bearings to conventional?

STERNLICHT: In answer to your first question. The percent that I showed for the three examples: the main propulsion turbine, the feed water pump, and the auxiliary turbine, show the percentage increase in mean time between failures.

The calculations used a similar method, but not exactly the same factors, as were used by Mr. Bazovsky in his study of feed water pumps for ONR. It compared present existing system...

KECECIOGLU: Is this experimental or prediction?

STERNLICHT: This is theoretical prediction. However, the analysis was based on existing performance data of the various components that make up the system and evaluation of engineering parameters which effect failures of new components.

Now with reference to your second question. One would have to know the actual application in order to determine the load carrying capacity and bearing size. But as an example with steam which is available aboard a ship, the load carrying capacity, pounds per square inch, will in general be greater than with presently used hydrodynamic oil, lubricated bearings. This is because high pressure steam is available.

With reference to stability problems there are several R&D studies presently underway. Some of these are sponsored by the Office of Naval Research on stability of rotors supported in bearings which employ low kinematic viscosity liquids, vapor and gases. The problems are definitely surmountable. A typical example of this was illustrated by the development of a gas bearing compressor. The reliability of this compressor had to be very high for it goes into a highly radioactive environment. I do not wish to minimize the problems which exist with the use of Process Fluid Lubrication. There are many and some of them are enumerated in the appendix to my paper. However, the payoff is so large that it warrants the development effort. For instance, with the use of gas bearings there are problems of materials, material compatability, etc. With the use of water or liquid metals at high speed we run into the problem of turbulence. Under this condition considerably more heat is generated within the fluid film, and boiling within the bearing may result. An understanding of the conditions under which this occurs must be well understood and avoided. This illustrates the need for research in order to provide the necessary tools for the designer. He then must understand the interaction between the various parameters before he can design a reliable system. In the case of Naval machinery the payoff in using process fluids lubrication is major. It will provide in many cases simplicity, compactness, elimination of subsystems (e.g., oil lubrication system) unreliable components (e.g. seals), etc. In many cases it will enable design of hermetic modularized machinery. Hermetically sealed and modularized machinery can be more easily monitored. I've seen numerous failure reports where it was impossible to determine what initiated the failure; or even which components failed. This is the one advantage of hermetically sealed systems which can be replaced and the failed system can be sent to the manufacturer or Naval laboratory where modes of failure can be established and future designs improved.

HIRSCHKOWITZ (U.S. Merchant Marine Academy): You made a comment that you felt that the fluid lubrication eliminates dirt and fouling. I have in mind perhaps a steam system, and I visualize the problems of carry over of boiler. I had ample situations in reducing valves, things of this sort were marvelously designed, beautiful things, the sole problem was carry over and the failures of this nature which don't seem like much are almost complete. You can't just have a bearing run a little hotter, but I've had oil bearings where steam was blowing on them. Well, you could put a fan on it. You could run the cooler a little heavier. You could swab a little oil by hand if you had to. You had these alternatives, and I seem a little frustrated at the point, not that I'm disagreeing with the approach of trying to find new answers, but I'm just a little bit at the question of whether some of these details of what a man can possibly do should perfection be eliminated. Now this steam cleaner is a very vital thing. Even on the Savannah it is a problem. You have fancy boilers, flash boilers and everything, beautiful controls, but they plugged up. They spent months and months. O-Ring problems, hydraulic systems and aboard they were just delightful. They were going to do everything imaginable, and one little poor surface or somebody sent the wrong ring, but these little details that cripple the whole venture — I'm not concerned so much with the problem of designing the component. I agree whole heartedly that the modular system is ideal. The company that builds it has complete quality control, but when you get it all into the assembly of the ship, somebody had better set up this standard of quality to tie them all together.

STERNLICHT: I am in basic agreement with the points that you raise. What I propose is simplification and integration of functions. In a system where the process fluid is steam, e.g., steam turbine; the bearings should operate on steam. This will eliminate oil system and prevent oil fouling of boiler, heat exchanger and condenser. Use the present system rather than introduce another system.

The steam that enters the turbine is perfectly adequate for the bearings and is as clean as any good oil lubrication system. When looking at cleanliness it is important to look at the size of the dirt particle in comparison to the minimum film thickness.

HIRSCHKOWITZ: My comment is this: I think an oil bearing could tolerate a dirtier oil than in terms of the amount of dirt you would tolerate in the steam system. This I think really is the problem.

STERNLICHT: This is true today because the oil bearing materials have good embeddability and good materials have not yet been developed for vapor bearings. But let us consider another point. For the last 50 years we have been using babbitt. Babbitt as a bearing material has a low temperature limit. For deep submergence the loads are higher and in order to minimize bearing temperatures the bearing size has to increase appreciably, or better cooling must be provided or new materials must also be developed. Thus the material problem still faces us in both vapor bearings or high load capacity oil bearings.

The fouling that you mentioned. The biggest problem of fouling is the oil getting into the condensers. If oil is eliminated so is the fouling problem. If water is used as the process fluid obviously it would be preferable to use clean water. Since a purification system exists aboard a ship it may be desirable, provided it is economical to use this water for bearing lubrication.

These are sighted merely as examples of the things that can be done and I do not mean to imply that they can be incorporated without further research, development and good design.

FRANCIS: I would like to ask in a case of steam lubricated bearings; if you lost the steam to the turbine, how would you maintain your bearing before your turbine bends up?

STERNLICHT: The same question can be asked about what happens if an oil system is lost. Without considerable thought I can think of several solutions to the problem you pose. In some systems it is possible to design the bearing to operate on water and then on steam when it is available. In other cases small steam boilers may be available to provide steam for start and stop conditions. Air compressors are available aboard a ship and these may be used for start and stop conditions. All of these, of course, present design compromises.

It should be pointed out that if today a PR came out for steam lubricated main propulsion or auxiliary turbine, development would be required. There are still numerous basic problems which demand development. In addition there are various system studies which will have to be performed.

KARASSIK (Worthington Corporation): I agree with you one hundred percent on this. However, this is an example of a situation where the mechanical engineer has outdistanced the metallurgist, at least for a while. We have designed — and so have other companies — bearings that can be lubricated with water, with steam, with gases, using exotic materials. But to get acceptable life, we need characteristics which are still unavailable.

There are such things as aluminum oxides, tungsten carbides, cermets and other materials which show promise. You may wish to comment on their suitability for steam lubricated bearings, or mention specifically the materials you have used in such service.

One of these days, the metallurgist will catch up with the mechanical engineer. In my opinion, he has not done so at this moment.

STERNLICHT: In the appendix to my paper the problems of materials are discussed and the criteria for selection is established. We have used both metallic and non-metallic materials. Many of these were quite successful. One example is graphite which operates very well in both water and steam. The problem with graphite and graphitar is that it is extremely difficult to establish quality control and inspection procedure. Thus some bearings operate extremely well and others supposedly of some composition behave very poorly.

ANONYMOUS: Why cannot we hermetically seal our systems today?

STERNLICHT: The reason why you can't hermetically seal the systems at the present time is that you have all of this clap trap of equipment like the pumps, and the coolers which supply the oil, and the seals and the equipment which separates the oil now from the water and the steam.

These accessories and additional sub-systems are causing most of the trouble.

If one could eliminate these systems there is a very good chance of hermetically sealing practically every piece of equipment aboard a ship. The ship could for instance be propelled with an electric motor which is located outside of the hull with the power being provided by a generator hermetically sealed as was shown in one of my examples.

ANONYMOUS: Could the steam end of this thing — this steam generator — is the steam end hermetically sealed too?

STERNLICHT: Yes, it could be a completely closed loop. You have the reactor, the reactor is generating the steam . . . .

ANONYMOUS: This can't be with a conventional plant?

STERNLICHT: No, not with a conventional plant. A plant can be designed as is shown on this slide to do this very job, so you have a completely closed package.

FRANKEL (MIT): If we use a reactor, we can add a closed loop although we still may have trouble hermetically sealing complex large systems. You mentioned that under merchant conditions, for instance, carry-over, etc., wet steam, two-phase lubrication could be substituted for single-phase lubrication and could even be substituted by water lubrication.

Now, I know very little about the clearances and design of such bearings; but I should imagine that they are different from the hydrodynamic, aerodynamic point of view that different conditions exist for all these three states.

STERNLICHT: The surprising thing is that they don't. Hydrostatic and hydrodynamic bearings for large turbomachinery applications operate with approximately the same clearances of one to one and one-half mils per inch. The hydrostatic bearings clearance is somewhat smaller than the hydrodynamic bearings.

It should be remembered that the steam in hydrostatic bearings is recovered. The energy required to raise the water to steam and the shear energy within the bearing are somewhat lower than the total energy required for oil lubrication. In many cases, therefore, there is an overall gain in efficiency by using steam lubricated bearings.

KAUFFMAN (Bureau of Ships): Pneumatically sealing small components today and replacing instead of maintaining I think would be one thing that we certainly should look at. Pneumatically sealing a whole plant would demand a point to point ship concept which the United States Navy can't buy because we have to be able to do some repairs other than sending large components back to manufacturer.

In other words we don't operate on a point to point basis and I can't see for a long time that we are going to either.

STERNLICHT: I cannot think of a better experimental laboratory than a well instrumented ship. As I proposed, perhaps there should be two Navies: one which conducts experimental studies in order to obtain higher reliability, etc., and the other being an operational Navy. As the components and systems are improved they can be transferred from the experimental Navy to the operational Navy. This proposal will provide tools in similar manner that the work study does.

KAUFFMAN: As my colleague said here, our Navy costs enough right now! I think the place to start with your work is on smaller components and not with the big one.

STERNLICHT: Let's start somewhere, for somewhere is better than nowhere. Thank you.

THE CONTROL OF THE INTEGRATED MARINE STEAM POWER PLANT

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SYMBOLS

T_i = Time constant

T_s = Time constant turbine

M = Torque or moment

f( ) = Function of ( )

J = Moment of inertia

ω = Angular velocity

n = Speed in rpm

t = Temperature

D = Damping torque coefficient

p_i = Pressure

$p = \frac{d}{dt}$ = Differential operator

P = Proportion of torque developed in H.P. turbine

R(t) = Probability of nonfailure in time t (Reliability)

G = Mass rate of flow

A, B = Constants

v = Velocity

$f_m(C)$ = Minimum or maximum response of m stage system for an allocation C

H = Heat storage or energy storage

Q = Heat rate

η = Efficiency

V = Volume.

Introduction

Many different plant concepts may be developed that can satisfy the objective of an integrated marine power plant in having all of the components so related as to permit operation in response to a single control signal from a central location. However, to be of significant interest, an integrated plant must satisfy the basic requirement of making possible net improvements in ship operating cost which are taken to include amortization of initial plant cost as well as the day to day expenditures for such items as fuel, crew insurance, maintenance, supplies and overhead.

A primary advantage of a completely integrated plant should be its capability of being operable with a reduced number of crew. This being the case, it is therefore essential that special attention be directed toward achieving a high degree of equipment availability in addition to providing means for automatic control. This will make possible a reduction in the absolute amount of maintenance and repair work performed so that work can be deferred for shore-side work crews as well as there being a reduction in required shipboard manhours.

Properly conceived, integrated and simplified plants will reduce equipment outage time due to maloperation resulting from human error, as well as the more readily recognized savings in direct and indirect crew expenses.

Of prime importance in the design of integrated marine power plants is the matter of design of the means of control. The requirements of simplicity and reliability are especially applicable in this area. In order to achieve an optimum plant control concept, the selection of primary power elements, such as the boilers and associated systems, is directly affected. Properly designed automatic controls can permit the use of elements having decided advantages in cost, space, weight and efficiency that may not be seriously considered with the more conventional and essentially manually operated equipment.

To integrate a system means to form it into a whole. In a physical sense this may imply simplification and reduction of components parts to form an integrated whole. From a control point of view it can be translated into achievement of a system controlled by a single or a minimum of correlated functions. To attain these objectives requires a reduction in the complexity of the marine steam power plant and a unification of control functions. While the former can be achieved by reducing the number of cycle components with a consequently slight reduction in plant efficiency, the latter requires a watching of component inertia in order to achieve comparable transfer functions. It also requires a drastic reduction in the number of feedback loops, which implies fewer variables and less dependence among the performance parameters. It has long been recognized that a reduction in the number of series components in the plant and the introduction of a few redundant parts for critical components can increase plant reliability appreciably. At the same time the introduction of multiple control loops nullifies any reliability gain achieved by component reduction.

The relationship of control systems and their attendant reliabilities to overall plant design has been generally taken into account on a somewhat intuitive basis. However, since the development of computers capable of handling simultaneously large numbers of variables, it is now possible to obtain rational solutions to whole plant design problems including the interdependence of such items as cost, weight, space, performance, reliability and controllability. This is shown schematically by Fig. 1.

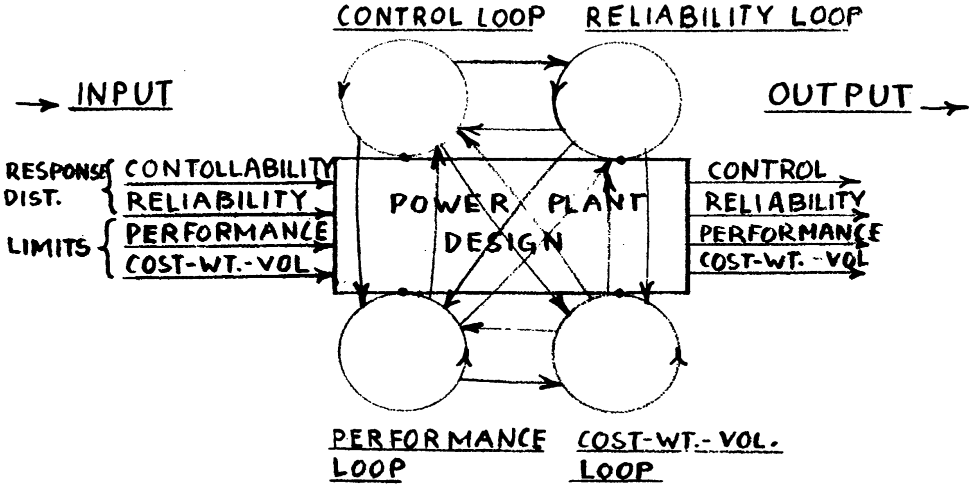


Fig. 1. Integrated design.

The significance of this approach is that it affords a perspective as to relationship of plant control design to all of the other variables considered, and by thus doing so results in the selection of solutions which may be considered to be truly optimum taken with respect to all other factors.

The Design of The Integrated Marine Power Plant

The successful design of complex power plants rests essentially on optimizing the balance of such diverse factors as performance, control, cost, reliability and time. One of the greatest obstacles to the effective application of computers and modern mathematical programming methods to such multivariable problems is the lack of reliable information on the quantitative relationship that exists between the desired end result and the parameters producing them. Much work and research

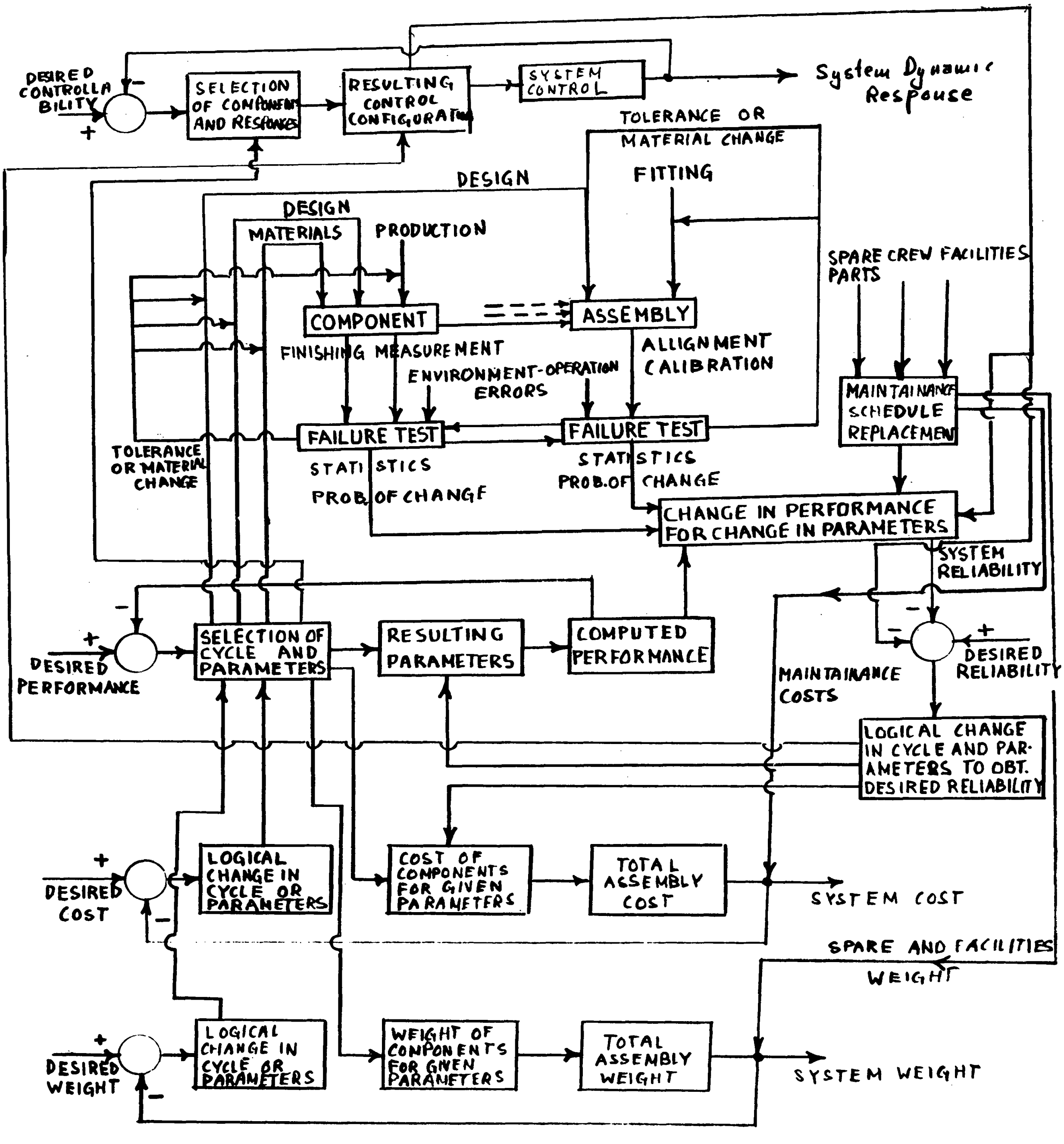


Fig. 2. Systems approach to power plant design.

must be done to extend the performance optimization studies presently resulting in supposedly optimum power plant cycles and component selection. Such studies must include other design factors such as reliability, cost, weight, dynamic response, controllability and time. A diagrammatic outline of a proposed optimization process is given in Fig. 2. This approach uses a number of feedback cycles with desired performance, control response, reliability, cost, and weight as input, and system cost and weight, for instance, as output. The input parameters are in the form of constraints and the computations are iteration processes by which an attempt is made to comply with the desired input parameters. Cost, weight and volume will normally be required to be minimum, while reliability and efficiency-maximum. Yet, constraints of "at-least" or "not more" than requirements are always in evidence. In essence, the process is one of successive approximations which will yield the sensitivity of each of the input parameters to changes in the others for a given system response.

In power plant design the problem is often a non-linear one in which effects of changes in a parameter are not the same in various particulars of its operating ranges or as a function of other parameters. Because of this it is often found advantageous for the system to work in one structural configuration rather than another.

While the input parameters in the reliability loop will be in the form of probability density distributions with their associated variance, most performance parameters will present discontinuities or discrete size change or steps. It is then required to change the stochastic output from the statistical analysis of failure tests into deterministic functions affecting plant performance and reliability, being constantly aware that the result has only a certain probability of being within said operating limits, or performing within output values. The required discrete changes in source of the input parameters will make it markedly more advantageous to operate with certain sets of values for a given parameter than with a neighboring set.

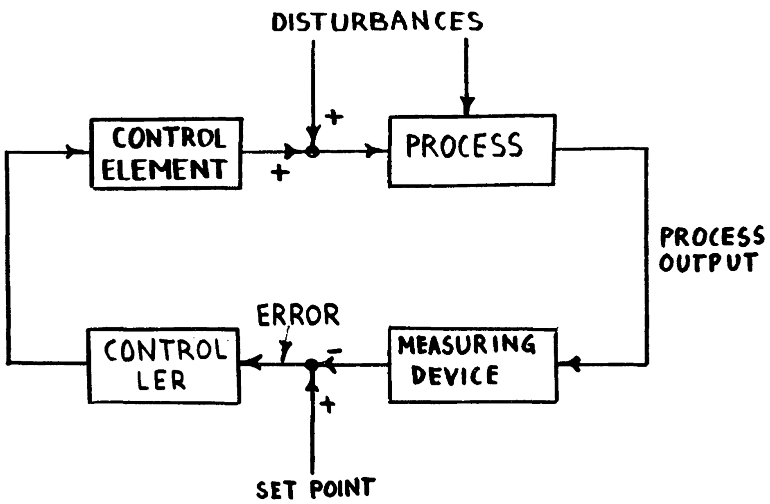


Fig. 3. Single control loop.

Considering the above-mentioned systems approach, a fuller discussion of the problems involved in introducing the dynamical and probability aspects into the preliminary design phase of the power plant follows.

Marine Power Plant Control

When control of an integrated system is discussed, it is implied that a number of coordinated actions exists based on feedback of information signals of measured variables to achieve a certain objective. This objective is normally maintenance of

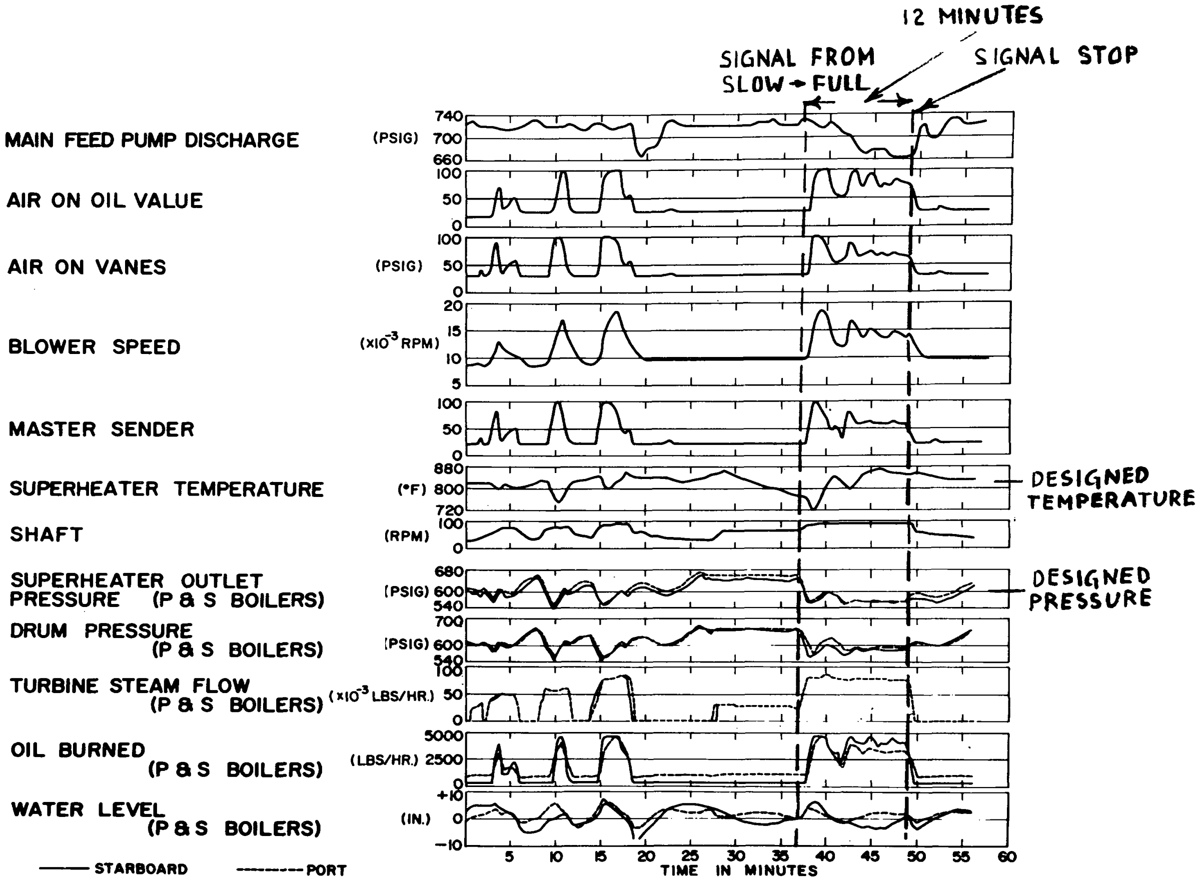


Fig. 4.

a condition such as temperature, pressure, level, or flow. The criterion of success of a particular control function is the amount by which it makes the system operation more effective or profitable. The profit is affected by the reduction of margin or safety requirements, increase in efficiency or other performance indices. As the cost of control components and instrumentation may be 5 - 20% of the capital cost of the system, evaluation of expected gains is essential. In considering power plant control it is necessary to first make a functional analysis of the system objectives in the form of performance parameters. This requires evaluation of component and system response to individual control loop configurations. This again means considering capacity, flexibility and safety built into a plant. Plants with large capacities and essentially steady state operations obviously require a minimum of control. Although the marine power plant has appreciable capacity, the kinetic response response requirements impose a need for a greater complexity of controls. In designing a control system two extreme philosophies must be considered. At one end we have the basic mixed (human and automatic) control system with controls fitted to respond to system functions; while the other extreme considers the wholly adapted control system where system components are designed to respond in a fashion resulting in easy-to-control functions. The optimum control system will normally be somewhere between these extreme alternatives. In general, plant control will consist of a multiplicity of control loops which may interact with each other. While most of the control functions are state conditions as mentioned before, continuous measurement of time rate functions are becoming of increasing importance.

In designing a control system we are concerned with controllability — which is a measure of dynamic improvement attained by control, stability — which defines success of response, and operational factors such as reliability, cost, and effects on steady state performance parameters.

While it is normally not difficult to design stable control loops for linear systems, it becomes a difficult proposition if there are marked non-linearities or strong interloop couplings. While most control functions in the marine power plant can be assumed linear within the range of small excursions from steady state, or linearized by piece-wise linearization as long as the rate of load change is reasonably small, interloop coupling is very pronounced. The latter results in difficulties of long term transient effects, local instabilities and design complexities, for example, the response of a 15000 SHP plant to load fluctuations as shown in Fig. 4. This plant is equipped with the usual modern feedwater regulator, combustion control, superheat temperature control and drum level control. It will be observed that the transient effects resulting from interaction of the different control functions take more than 15 minutes to reduce to reasonable values. Comparing this time with the usual requirement of load change from zero to full power in less than one minute, it will be recognized that the result obtained at a cost of a large and complex control system is a compromise of acceptance. Major factors in the controllability of a system are dead-time or transport lag and capacity or transfer lag. The energy storage in the process consequently plays a major part in the control response of a system. The reason for many of the difficulties and complexities in designing effective control systems for the marine power plant is the large and diverse energy storage or capacitance in the different components of the system. Another factor is the distance between the disturbances and the sensing devices which, with the designed flow rates, result in appreciable and diverse transport lags.

Much work is currently being undertaken in experimental determination of transfer functions of the different processes involved in the steam power plant. Statistical correlation of input and output data (frequency response method) (1) to obtain a measured transfer function is not yet compared with calculated transfer functions from basic design parameters as the latter is extremely difficult to compute. The marine power plant has a large number of distributed parameters which results in the formulation of extremely cumbersome partial differential equations. An added difficulty is our ignorance of the physical phenomena of two phase flow. Automatic control, in addition to feedback loops, also employs computers to regulate the logical sequence of events as a result of transmitted signals. For successful application this requires not only that measuring instrumentation be accurate, but that the control system responds quickly, and a model for operational procedure and a criteria for optional conditions be available (See Fig. 5). Apart from this, fail-safe arrangements must be provided if the computer is to take the place of a human operator. One of the difficulties in providing these essential requirements is that real systems are all incompletely deterministic, or, in other words, their response has a probability distribution.

Consequently any model provided for computer guidance is inadequate. As a result we attempt to design control computers to use collected past operating experience to improve future command response (learning). Adaptive control theory (2) has been developed to provide the tools for the design of such computers.

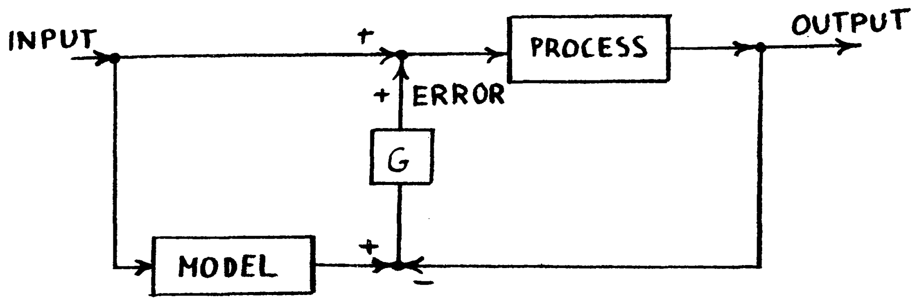


Fig. 5. Feedback system with model (transport lag and nonlinearities permissible).

In a way, the computer then acts to fill the gaps in the empirically derived design data and also takes cognizance of changes in component or system characteristics as a result of environment, wear, etc. The large amount of data collected by digital loggers permits dynamic models with some degree of accuracy to be constructed. While digital computers were mainly considered for control functions in the past, recent developments in the use of dynamic memory will permit much more use of cheaper analog computers to be made in this field.

Dynamics of The Marine Power Plant

Traditional design procedures for control systems of marine steam power plants are largely made by inspired guess-work with little or no stability or response calculations. This approach is becoming inadequate with highly loaded plants and may result in erratic response during operation, restrictions on freedom of operation, as well as performance far from optimum. Even simple calculations based on empirical data will permit more reliable estimates of control response to the complex and varied input parameters of the modern marine steam power plant. While no attempt is made in this paper to discuss details of dynamic response analysis for the multitude of marine power plant components, a few words about some specific problems may be opportune. The dynamic response of a single-cylinder steam turbine can be expressed by a transfer function of the form

$$\left[\frac{1}{T_s\, p + 1}\right]$$

A similar expression can be found for the gear, shaft and propeller assembly. If ΔM is the change in turbine torque produced by a change in steam flow rate resulting from a change in throttle position, if D is the damping torque coefficient of system, J the total moment of inertia of turbine, gear, and shaft assembly and $\Delta M'$ change in applied torque produced on gearshaft assembly, then

$$J_p^2 n + D_{pn} = \Delta M - \Delta M^1$$

from which we obtain the resulting change in speed

$$\Delta n = \frac{1}{J_p + D} (\Delta M - \Delta M^1)$$

$$\text{where } n = \text{speed in rpm}$$

$$p = \frac{d}{dt} = \text{differential operator}$$

The damping coefficient can be obtained experimentally or analytically by using the expression (torque change with speed at constant throttle position)

$$D = \frac{\partial\ (\text{Load Torque})}{\partial n} - \frac{\partial\ (\text{Turbine Torque})}{\partial n}$$

If ΔY is the change in throttle position, then $\Delta M^1 = \frac{1}{T_s\, p + 1} \Delta Y$. As explained in Appendix III the response is more complex with a two cylinder turbine arrangement or reheat plant, when an additional time lag as a function of steam pipe space is involved. The above expression then assumes a form:

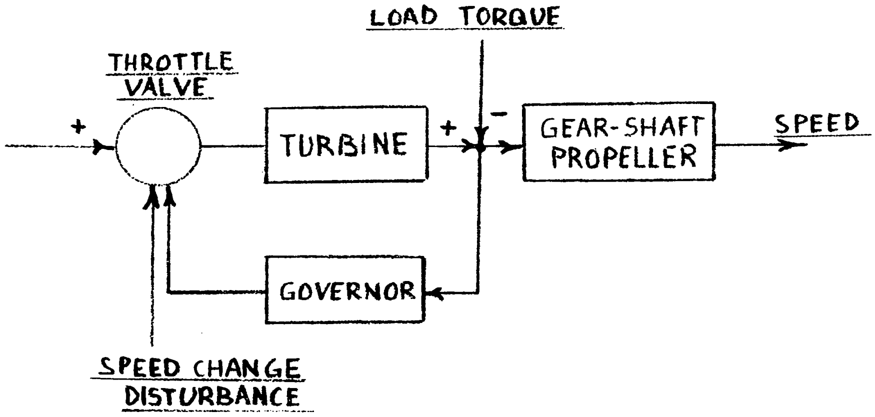


Fig. 6. Block diagram of single cyl. turbine.

$$\Delta M^1 = \frac{(P\,T_p\,p + 1)}{(T_p\,p + 1)} \left[\frac{1}{T_{s_1} p + 1}\right] \Delta Y$$

where P = proportion of torque developed in H.P. turbine

and T_p = time lag associated with the steam pipe and/or reheater

and T_{s_1} = time lag for the set of H.P. and L.P. turbines.

A considerable time lag is incurred in the steam pipe between boiler and throttle which complicates the pressure variation resulting from a change in throttle position. Much work has recently been done to analyze the dynamic response of boilers (4), (5), (6), (7), (8). However, the lack of understanding of the two phase flow phenomena requires a linearized or semi-empirical approach to the problem. Profos (9) proves from experimental results that the response of the firing system can be expressed by a formula of the form

$$\left[\frac{e^{-T_f p}}{T_f^1\, p + 1}\right]$$

where T_f = velocity or transport lag

T_f^1 = time constant or lag.

A similar expression is also applicable to feed pump response.

The major difficulty in integrating the control of a conventional (drum type) steam power plant is the dependence of the various control functions, as explained previously. As a result, a multitude of feedback signals from the different sensing devices is required for control. Steam temperature, steam pressure, water level, uptake temperature, etc., are functions of all the input variables, and consequently a change in one of the input variables requires a variation of all the others.

Power plant control philosophy rests on two diverse approaches. The boiler following configuration, Fig. 7, imposes load changes on the prime mover that utilizes the large capacitance of the steam generating equipment to follow the required load change as rapidly as its own inertia (mechanical-fluid-thermal) will permit, and assumes that the control loops of the boiler will be able to correct boiler output before the capacitance is expended. A number of conflicting control loops are imposed on the boiler with the addition of limit or safety devices, and we assume constant pressure and temperature conditions at boiler exist at all loads for the turbine computations. Practical experiments show that it takes a long time for the transients to die down, Fig. 4, and that, in fact, superheated steam

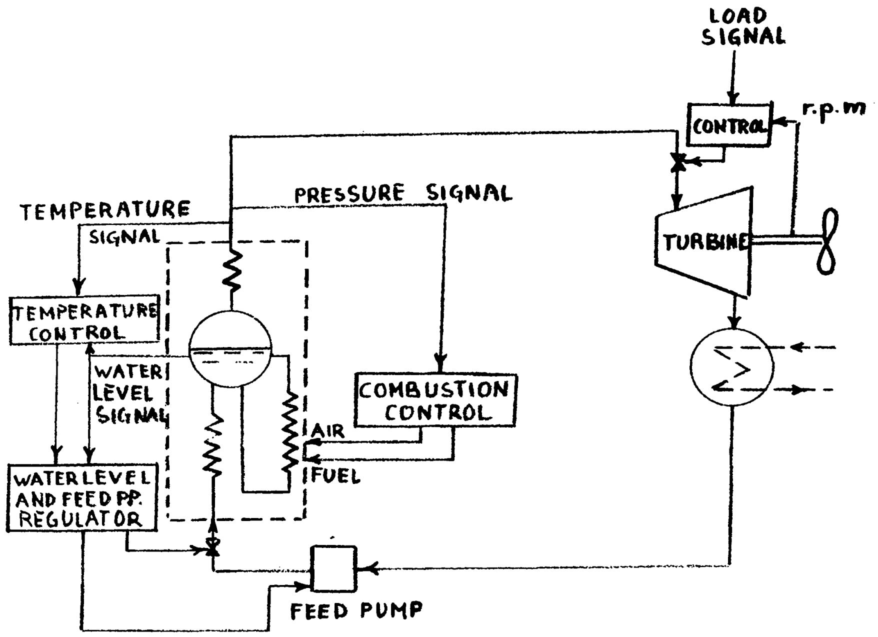


Fig. 7. Boiler following control.

temperature will vary appreciably over the load range, which will affect plant economy at other than designed output, Fig. 8. "Boiler Following" is universally employed in marine steam practice owing to the recognition that the inertia (fluid-thermal) of the drum type boiler plants is too large to respond to the required rate of load fluctuations. Feed water controls are still based on drum level with sometimes a modifying signal from steam or feedwater flow. Actually the basic requirements are to supply feed at a rate that matches the steam flow required and not to maintain a hypothetically correct drum level. The inclusion of drum internals imposes limitations on permissible drum level variations, and the diverse time lags in response to combustion and feed control signals result in long term transients. Similarly, most superheater temperature controls generate corrective action only after occurrence of a temperature deviation. As the derivative of this deviation is seldom used as a signal, over control throughout a lengthy transient period occurs. The real objective of superheat temperature control should be the prevention of temperature excursion.

Combustion control is also normally designed to control the isolated system variables of fuel and air rate to maintain or correct steam pressure, instead of supplying the correct amount of fuel to match load requirements.

The foregoing remarks are not new and attempts have been made to impose absolute control functions by feeding modified load-change signals to the different controllers programmed to predict and prevent resulting excursions. This is done by digital devices, using empirical data stored in their memory, which are superimposed on the conventionally sluggish control elements. The philosophy of this approach is that, assuming a given change in power output, all the required input parameters are computed and applied with their required lags to the system by means of their controls. Feedback is temperature, pressure and steam or feed flow and not final control element position. Automatic start up and shut down as well as casualty conditions are also programmed based on monitored conditions of the system and not on the conventional pure time basis. It will be found that analogue computers are much more amenable to perform these functions and are sensitive enough for most applications.

The second approach of power plant control philosophy rests on the assumption that load fluctuations imposed on the system will not exceed the response capabilities of the highest inertia component of the system, the boiler, and consequently

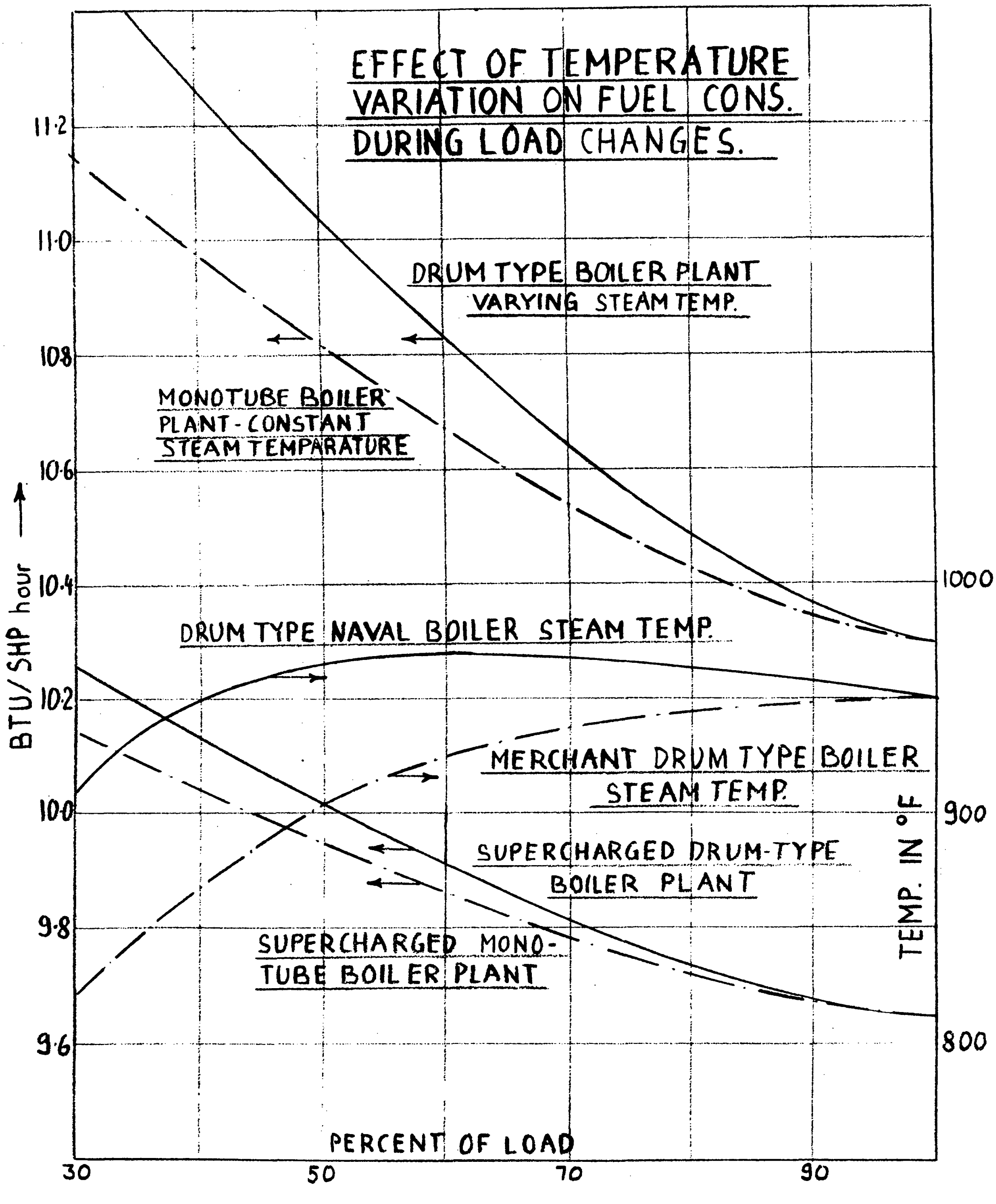


Fig. 8. Effect of temperature variation on fuel cons. during load changes.

load changes are affected by varying the input parameters to the boiler and allowing the turbine to follow the resulting steam flow rate (See Fig. 9). As constant turbine inlet pressure is advantageous, an arrangement limiting steam flow to the turbine as a function of boiler output pressure is required.

This system has been used in a small number of stationary plants with low inertia steam generators. In marine practice the required rate of load change (60 sec. from 0 to Full Power) and the high inertia of the universally applied drum type boiler have resulted in rejection of the turbine following concept. Considering the reliability of the two opposing philosophies, it will be shown subsequently how the turbine following approach permits a significant reduction in the number of control loops required and, in fact, may permit a single loop control to suffice in

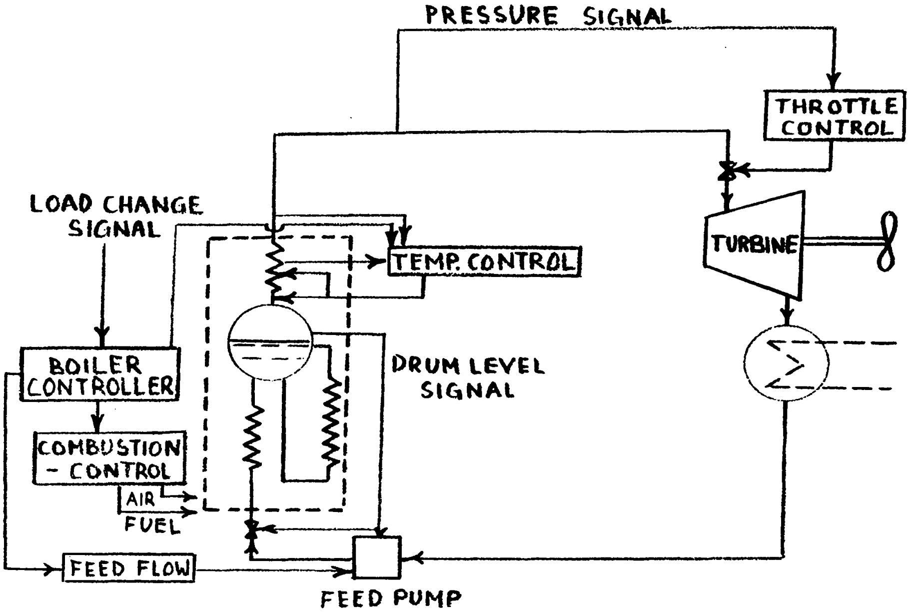


Fig. 9. Turbine following control (drum type boiler).

systems using drumless once-through boilers. Essentially the approach is one of load control by firing control with the related feedwater flow rate adjustment.

It is noted, however, that the boiler following system will require a minimum of four control loops resulting in lesser plant reliability. Failure of any of the control loops will necessarily constitute failure of the plant and, as mentioned before, in actual practice additional limiting or safety loops would have to be introduced to guard against permanent damage as a result of failure of any one of the control loops.

Reliability as a System Parameter

The introduction of additional feedback control loops can change system availability appreciably. In the past, power plant design was primarily based on computing the combination of performance variables such as state points, flow, etc., to give optimum efficiency. In recent years economic parameters and power plant weight have also been introduced. Yet, reliability which is probably the largest economic factor in overall plant performance has traditionally received only passing consideration in the design stage. Reliability, which affects practically all performance variables including efficiency, must be treated as a major parameter in plant design. In addition, it is suggested that dynamic response terms be introduced as system variables so that the cost and requirements of transient behaviour can be incorporated at an early stage.

To achieve optimum efficiency of our plants as well as to introduce control, the complexity of our systems has increased tremendously. If we plot theoretical system performance (which may imply efficiency, output, etc.) as a function of complexity, we obtain a curve approaching theoretical maximum performance. Yet, if actual performance is plotted on the same base, performance will decrease after reaching a certain amount of complexity (Fig. 10).

The reason is severe degradation brought about by unreliability. This tends to be a vicious circle. Complexity is increased to achieve high performance, but at the same time the increase in complexity actually prevents ever reaching this goal. Consequently, reliability has to be introduced into the initial design studies and not as an after-thought when the design parameters are all fixed. One of the reasons

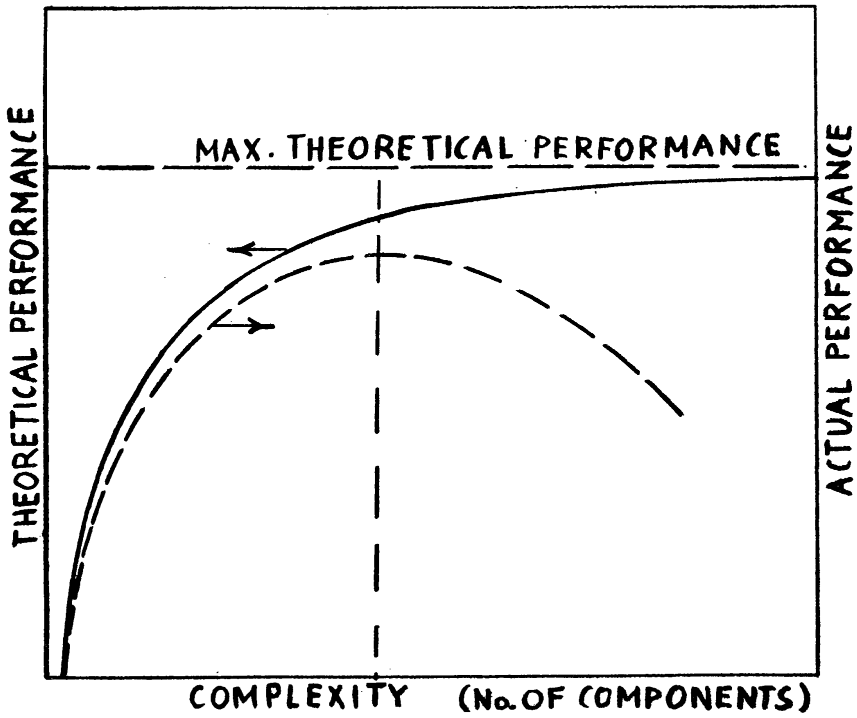


Fig. 10. Variation of performance with complexity.

that reliability has not received sufficient attention is the fact that true operational reliability is difficult to analyze. Quality, design, maintenance, environment, production methods, etc., all affect the result.

It is therefore wise to place an upper limit on the reliability a component will attain, assuming thereby, that there is a high probability that its reliability is below this value. This procedure will assure safe system reliability results. The consideration of reliability as a design parameter must be recognized. In optimizing system parameters, reliability is then compared with other variables to find the coupling effects. For a most effective or best plant a compromise between best performance and reliability is then arrived at within certain constraints, (weight, cost, volume, maximum conditions etc.). Considering the probability of successful component or system performance as a function of performance parameters, it is apparent that this curve will approach the theoretical maximum, yet at the same time plotting reliability to the same base will result in an asymptotic curve.

Another difficulty in using reliability as a design parameter is that it is not a readily deterministic quantity. Life distribution of "identical" components varies, environmental effects change and interactions of different failure events is hard to predict. Yet by considering the whole array of stochastic inputs in the form of a multivariable distribution with certain intersections, useful results are obtainable and reliability improvement can be incorporated. The indiscriminate use of redundancy for reliability improvement is another aspect for discussion. As the reliability of the failure detection and switching devices required by redundant components is finite, an upper limit is placed on reliability improvement. In general, exponential reliability expressions are used for components in a mathematical evaluation but it must be remembered that exponential component life characteristics do not essentially imply exponential system life characteristics, nor is system failure rate based on component meantime - before - failure necessarily valid.

Reliability has always been of prime importance to the Marine Engineer and the criterion by which every decision in design and operation was measured. Complex

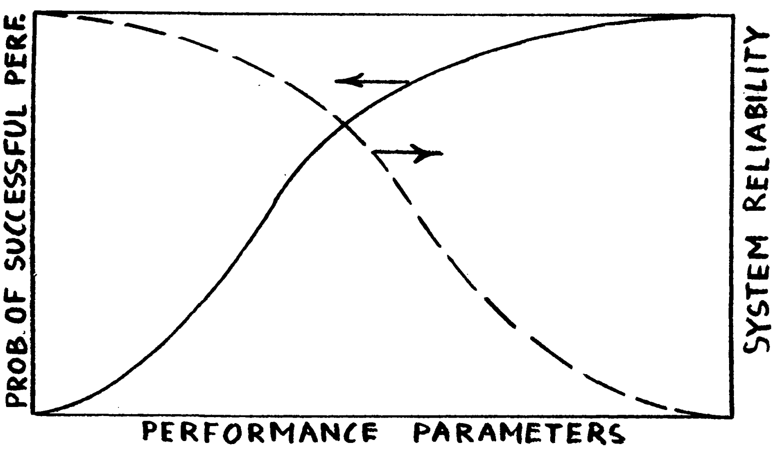


Fig. 11. Relation between variation of performance parameters (eff.) and system reliability.

equipment and intricate cycles have been installed to provide optimum efficiency after being fully aware that the effort was of doubtful benefit under actual operating conditions. How often is it discovered that the control and governing devices on these ingenious auxiliary arrangements devised to attain optimum cycle efficiency are overridden by manual control because of malfunction, unreliable operation, or resulting instability?

When reliability of components, systems, or complete plants are discussed, limited and often biased experience is used as the yardstick for decision. How many designers are able to quote actual quantitative values for the reliability of a component? The fact is that the builder or designer normally dissociates himself from the product once legal responsibility has lapsed. The reliability of a particular component is sometimes increased to cure a specific fault as a result of complaints in order to maintain good will, but often simultaneously changes the component response resulting in an overall decrease of plant reliability. Analysis of power plant reliability is an essential prerequisite to power plant control. An unmanned plant has to achieve optimum reliability and have accurately predictable maintenance schedules. On the other hand automatic control complicates a system by adding subsystems and additional components all with a finite time between failure which must necessarily decrease plant reliability.

The discussion of reliability analysis of complex systems under constraints shows this effect (Appendix IV). Consequently, an automated plant has to consist of a smaller number of series components to assure equivalent reliability as a non-automated plant. Simplicity can be said to be the key to reliability, and this applies equally to the design of individual plant components and to the whole plant design. An essential requirement for simplifying cycle design is conformity of component parts. Further, designs are only justified if they combine simplicity, effectiveness, and utility with essentiality of their function. Interest in marine steam power plant reliability is particularly timely at present. Since the emergence of the diesel as a competitor in the higher power ranges of ship propulsion, complacency in the steam field has been excused by the inherent advantage the steam concept offers with regard to maintenance, reliability, and spare parts' costs. The hitherto accepted continuous maintenance requirements of the diesel demanded large proficient crews and resulted in periods of unavailability between voyages.

Recently Japanese operators, in cooperation with large slow direct drive diesel manufacturers, have completed extensive trials in which no maintenance or inspection was performed during 16 month periods (6000-8000 operating hours). It was found that wear as well as operational reliability was equivalent to that of

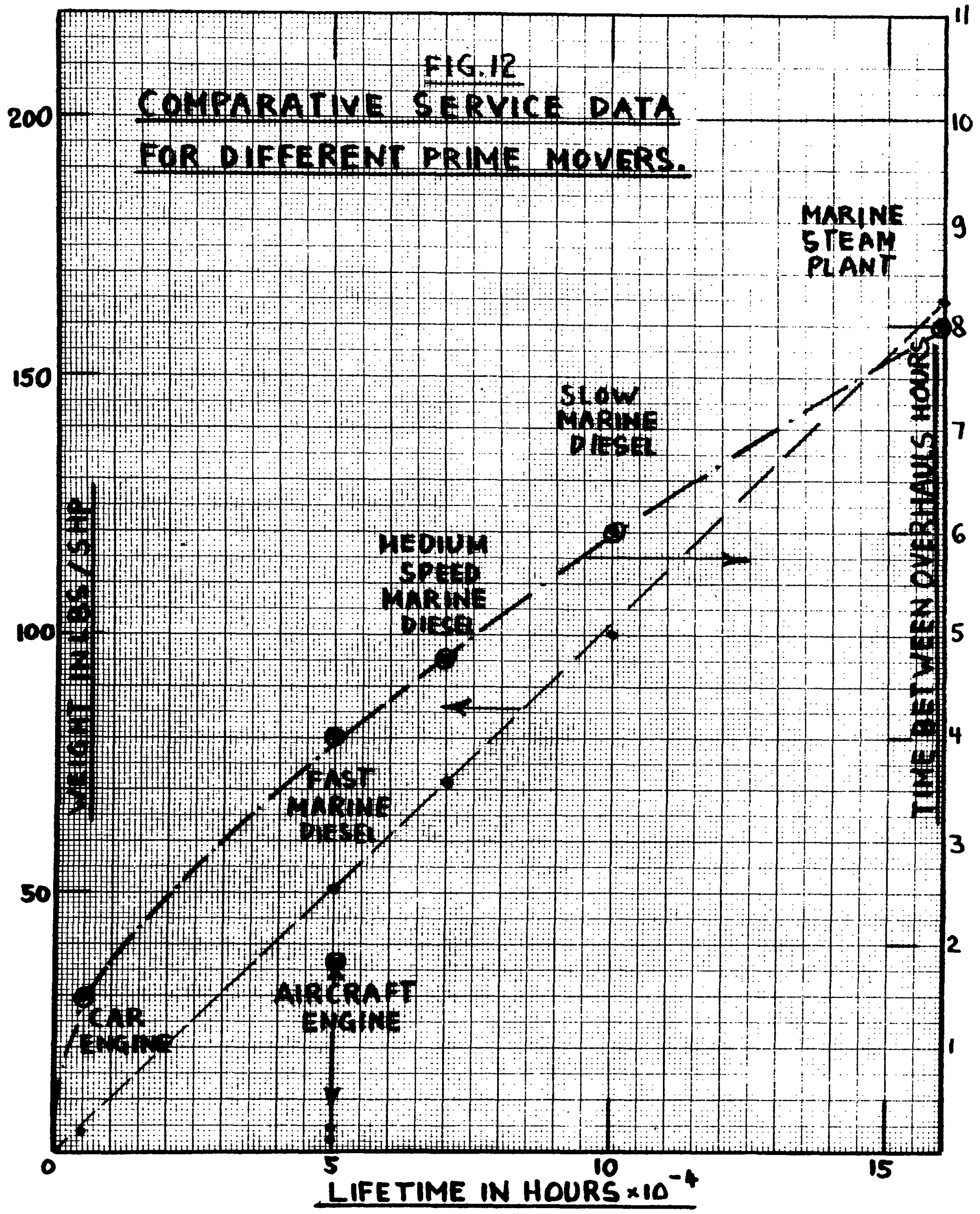


Fig. 12. Comparative service data for different prime movers.

continuously maintained plants and with an average linear wear of less than 7/ 1000″/ annum. In these trials a working life (without linear or piston replacement) of 20 years could be assumed. Pretreating the heavy fuel oil by more refined methods and chrome plating cylinder liners is expected to decrease wear even more. The obvious reason for the above experiment is to drastically reduce the crew on diesel propelled vessels, many of which today already have single console or bridge control, directed toward the eventual elimination of attendance in the engine room and widely spaced maintenance and inspection periods. The diesel engine manufacturer normally designs as well as builds all the component parts of the prime mover, and is intimately conversant with component response and, consequently, control requirements. However, the process of evolution of the marine steam power plant has resulted in a large number of different and often non-coordinated components each with its own control arrangement. The result is needlessly complex and unreliable and contains, in many cases, a multitude of redundant features (Fig. 12).

Control of the plant, or even of subsystems, is seldom integrated. Non-uniform display and recording systems often result in reading errors and misinterpretation. Each component manufacturer represents a different point of view with regards to plant control in which his primary interest is to prove the superior performance and reliability of his part without reference to the input requirements of other components in the system. The responsibility for this state of affairs is a joint one, as suppliers are normally only given required single point performance data which does not permit an evaluation of component response interaction with the control and stability of the plant cycle. Power plant design is normally concerned with the optimum definition of static state points.

Little consideration has been given to dynamic behavior in the design of steam power plants. While it is true that the stationary steam plant operates essentially in steady state conditions with random disturbances causing fluctuations in system properties, the same does not apply to the marine power plant. The mathematics of distributed non-linear systems precludes simple, direct, exact solution of dynamic response, and theories for optimizing control configurations using standard techniques are tedious even if parameters are restricted to one loop gains, etc. Therefore, little progress has been achieved with our classical methods in achieving quantitative design information concerning plant dynamics and controllability. Monitoring every operational variable and parameter is still not always possible, because of interdependencies and control actions of our high speed correlators. The so called black box treatment may achieve success; but Messarovic (15) has shown that if each dependent variable is considered a function of all other variables (dependent and independent) there are not enough independent measurements that can be made to establish dependencies empirically, and, therefore, universal success is not assured. During casualties modes of operation may also occur with which the computer or control element had no previous experience. Some extrapolative scheme must therefore be included, as there is no certainty, prior to a casualty, that a control extrapolation is valid for the case and that other system components will not fail as a consequence of the failure of the initial component. A lot of work has to be done yet in defining the coupling of fluid, thermal and mechanical inertia of the system, and the interaction of the so different mass, momentum, and heat storage. Fluid-mechanical transients are generally an order of magnitude faster than thermal transients, but coupling is important, particularly in the case of the boiler or condenser where two phase storage is significant. So far the reliability of control of a complex plant has been discussed, but some mention of the control of reliability may also be opportune. An analysis of a large sample of control components, as well as mechanical components, has shown that the reliability of a part can be expressed as a function of its cost or tolerance:

$$\text{Cost} = \frac{A \exp - B\,(1 - R(t))}{(1 - R(t))}$$

where A and B are constants

$R(t)$ = Probability of Non-failure in time t

$(1 - R(t))$ = Prob. of Failure in time t

The cost of a component is normally a function of tolerances in materials, manufacturing processes, inspection and testing procedures, design limits, erection, or installation tolerances, etc. It is obvious that for optimum plant reliability within a set of constraints, only one optimum selection of components is possible resulting in an expression of the form:

$$f_m(R(t)) = \underset{\text{system}}{\text{Min}} \left\{ \left[\left\{ n_1 \sum_{j=1}^{s_{11}} \frac{A_{1j} \exp - B_{1j} (1 - R_{ij}(t))}{1 - R(t)_{ij}} \right\} + \left\{ (n_1 - 1) \frac{A'_{1j} \exp - B'_{1j} (1 - R_{1j}(t))}{(1 - R(t)_{1j})} \right\} \right] + f_{m-1} \left(\frac{R(t)}{R(t)_1} \right) \right\}$$

as explained in Appendix IV,

where $R(t)_{ij}$ = reliability of j^{th} redundant component of the i^{th} series component

and $R(t)'_{ij}$ = reliability of the associated fault detection and switching device.

To control reliability much more work and data collection will have to be done to present reliable definition of the distribution and consequent variance of failure events with tolerances.

The intersection of the distribution of different failure events will also have to be studied in order to arrive at valid control configurations and programming during casualty conditions. Such analysis will also permit effective and reliable maintenance scheduling and spare part stocking. A brief introduction into some of the problems involved in these problems is given in Appendix IV.

It is proposed herein to investigate the possible application and expected advantages of a radically unconventional departure both from modern marine power plant design as well as control. It is recognized that none of the ideas discussed is strictly original and that many successful low inertia boilers have been built by all of the major boiler makers. Yet the reluctance of marine engineers to adopt, evaluate, and try this concept is only reminiscent of the period when ships were equipped with fire tube boilers when utilities had been using water tube boilers for decades. Then, as now, control and operating problems were quoted as the major obstacle to their acceptance. With the present state of the control art as well as the knowledge of water treatment gained in nuclear plants it appears that these reasons are bound to disappear shortly. It is felt that with all the excellent developments of improved marine steam power plants, full advantage has not yet been taken of all the prospective benefits of automatic control. Further, it may be even cheaper and more effective to design plant components to comply with control requirements, and not the other way around. Neither the proposed plant nor some of its components have been proven in the marine environment, but land applications have been so successful, that the proposed steam generator type has been almost universally accepted for new utility stations. To illustrate, a plant is proposed which reduced this functional dependence appreciably and consequently requires a minimum of control. It is furthermore hoped that this approach may result in a more stable, less costly, lighter, and eventually more reliable plant.

The Low-Inertia-Boiler Power Plant

The basic reasons for proposing a radical change in power plant control philosophy and component design is, as discussed previously, the need for simplification of control functions as well as reduction of the total number of plant and control components. The results of this initial study, which covers only some of the most fundamental aspects, shows that large gains are possible, although it is recognized that many operational and design problems are yet to be solved. It is hoped that more conclusive results may be presented at a later date.

Table 1. Capacitance of Boiler Types

	Ratio $\frac{\text{Time constant of Storage}}{\text{Time constant of Thermal Inertia}}$
Fire Tube Boiler	1/8 ⟶ 1/1
Drum Type Natural Circulation	1/4 ⟶ 1/6
Drum Type Forced Circulation	1/8 ⟶ 1/10
Once-Through Boiler	1/60 ⟶ 1/100

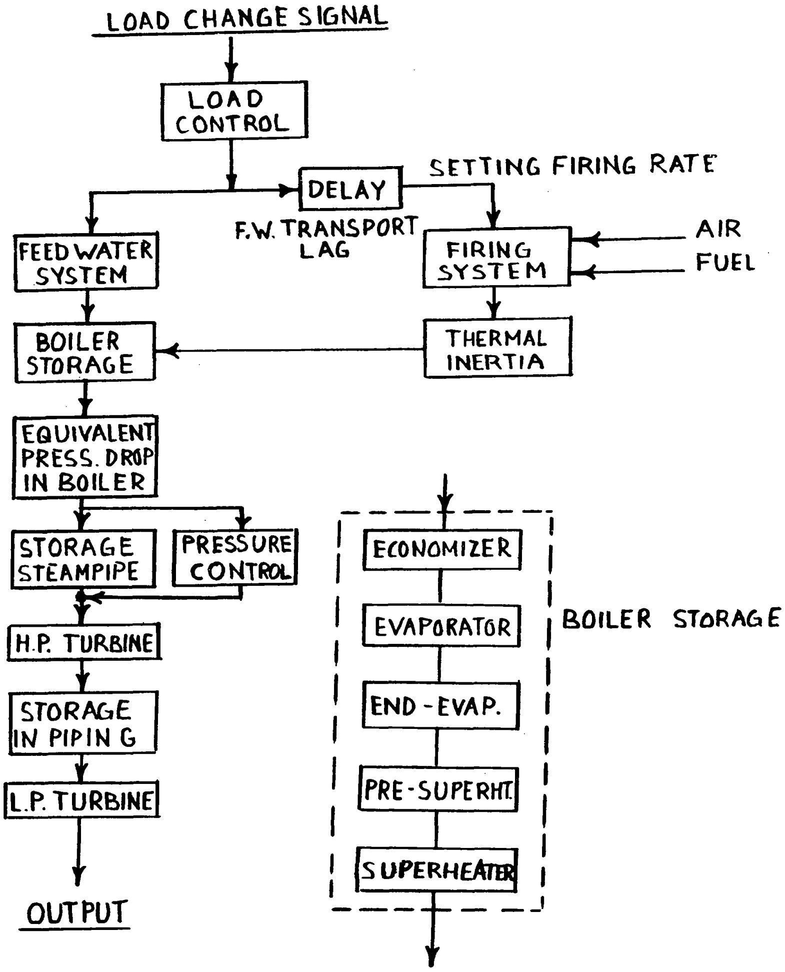


Fig. 13. Basic control scheme.

Some of the objectives sought and criteria imposed on this design are summarized as follows:

1. Power plant must respond to the usual required changes in load demands. Statistical evaluation of operational results shows that a typical plant of 22000 SHP requires 50-80 seconds to change from zero to full load throttle setting, and 30-50 seconds from full load to zero load. (The discrepancy is mainly due to time lag in combustion and feed control. After 50-80 seconds only 80-82% full power is actually developed in spite of full throttle setting. It takes 8-12 minutes to achieve 100% full power.) In a turbine following plant, therefore, a boiler response of about 50-80 seconds from zero to full load will be required, which should result in comparative turbine speed response.

2. The number of control loops and control functions is to be reduced to the absolute minimum consistent with safe and stable operation.
3. The plant should comprise the smallest number of components consistent with reasonable efficiency.
4. Plant control should be adaptable to remote or fully automatic control. Computer control of start-up, shut-down, and casualty conditions should be incorporated.
5. Reliability of the plant should correspond at least to that of present day marine steam turbine plants. Component reliability should be adjusted by design, material, quality, or redundancy to achieve optimum plant reliability, and an integrated consistent maintenance schedule to retain the reliability for the life of the plant.
6. The plant should require a minimum of in-service maintenance. Atomizer and filter cleaning operations are to be automated and restricted to once a day or once a week functions.
7. To simplify boiler firing control, burner torn-down ratio is to be extended to the full range of load. (A lower limit of 5-8% of boiler throughput is imposed by boiler requirements.)
8. Repair costs for the plant are to be minimized by reducing components and materials subject to deterioration to the bare minimum. As an example, the amount of refractory in the boiler is to be greatly reduced.
9. Although full supercharging with present day Bunker C fuel and pressure type atomizers is not yet feasible, a small amount of pressure charging may be considered and the advantages and problems of supercharging using a variety of atomizers is to be investigated.
10. Plant dynamic studies by the author have shown that the length of main steam piping between boiler and throttle is a major factor in steam exit pressure instability and load change response. Attempts are to be made to reduce the length of steam piping to absolute minimum.
11. The design is to achieve reductions in both capital cost as well as plant weight. The effect of these parameters in operation costs is to be evaluated.
12. Space studies are to be undertaken to minimize total volume requirements and also to optimize location of components for least piping lengths and ease of access (reductions in maintenance costs).
13. An investigation of auxiliary machinery, its functions, redundancy, and influence on main plant operation and stability is to be made.
14. The use of demineralizers to replace much of the chemical boiler water treatment is to be considered to assure pure feed and protection from condenser leak damages. It is also expected that greater feed purity and uniformity may appreciably reduce boiler upkeep costs.
15. The steam plant is to be able to allow quick starts and shut-downs and variable temperature operation to reduce thermal stresses in insufficiently preheated turbines.

The plant considered is of 22000 SHP with a steam pressure of 850 psi and a steam temperature of 950° F. Desuperheated steam at 850°-870° F and 550-600 psi for the turbo generator is to be obtained by desuperheating in a steam air heater, economizer or external desuperheater. It is recognized that this method is inefficient due to the large temperature differentials, but as the amount of steam involved

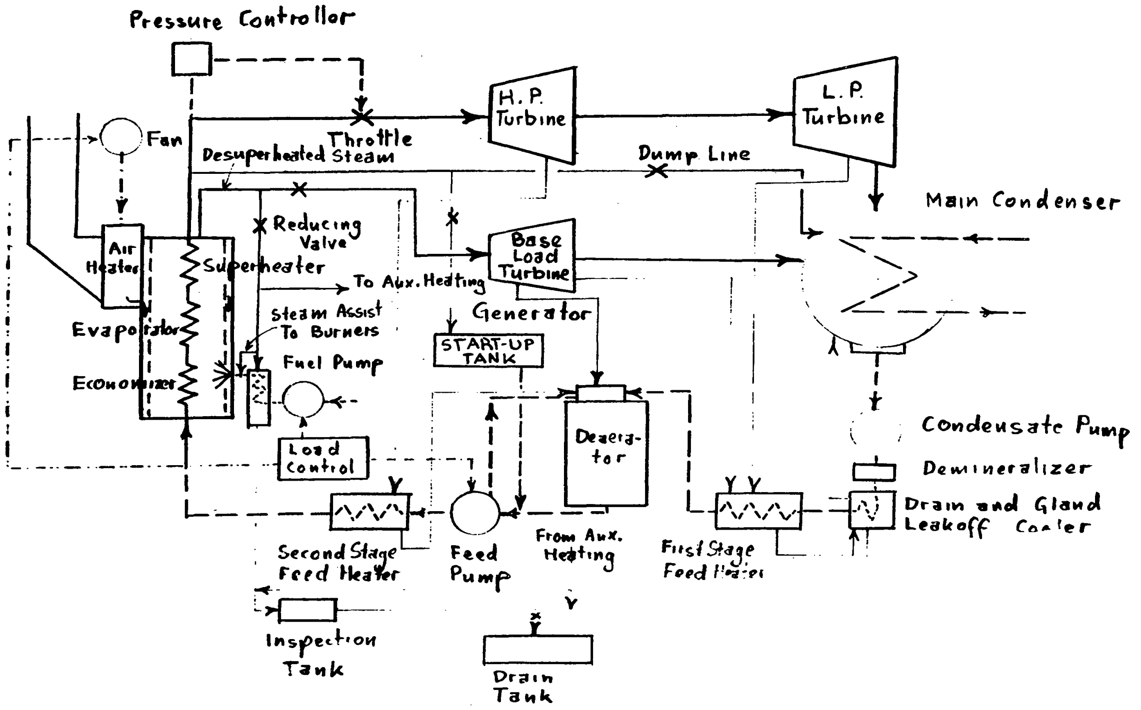


Fig. 14. Basic heat cycle diagram

is small and other methods would complicate the plant, this approach is deemed justified. The steam conditions proposed are the highest which have been proved in service up to the present. Load control is by direct firing rate control which also controls feed supply with an appropriate time or transport lag. The pressure at the boiler outlet is maintained practically constant by making the main turbine throttle opening a function of exit steam pressure variation. A single boiler is proposed for this plant. It is recognized that only a minority of shipowners will presently consent to have single (even drum type) boiler installations and it has been shown that the difference in capital cost is only about 20% which most owners are willing to pay for the ease-of-mind and facility of maintenance scheduling. Yet it is believed that the great increase in boiler availability (about equal to turbine availability), the vast reductions in refractory mass, and the provision of pure feed, will reduce maintenance requirements greatly thereby affecting this bias. From a control point of view single boiler plant operation results in great simplification. Similarly, weight and space gains are appreciable. In the case of the low inertia steam generator plant, dimensions would permit placing the boiler between or adjacent to the cylinders of the turbine, thereby, also reducing main steam pipe lengths appreciably. In the proposed plant a take home motor of 750 SHP is to be provided.

The main propulsion unit consists of high pressure and low pressure cross compounded units. Astern turbine elements are, as usual, located in the low pressure casing. The main condenser is of single pass design with scoop injection and pump circulation. An attempt was made to evaluate possible condenser weight savings by increasing water speed through the tubes to above 6 feet per second. It was found that the small reduction in weight did not warrant the increased size of the pump, the increased power requirements, as well as the increased tube and condenser cost (copper-nickel instead of aluminum-brass tubes). Using a 6 feet per second speed and 3/4 inch diameter tubes, the optimum vacuum condition (as a function of cost differential) was found to be 28.5″ Hg. A simple study of the effect of tube size again showed that, while 3/4 inch diameter tubes could result in a saving of nearly 3 tons in tube weight as compared to one inch diameter tubes, the cost

TABLE 2. SIMPLIFIED COMPARISON OF MARINE POWER PLANT CONTROL FUNCTIONS.

	Drum Type Boiler Plant			Once-Through Boiler Plant		
Control Function	Control Signal	Corrective Measure	Influencing Factor	Control Signal	Corrective Measure	Influencing Factor
Superheat Temp. Control	Steam Exit Temperature	Feed injection or Sat. Steam injection	Steam entry temp. Steam flow Firing Rate % Excess Air Position of Superheater	None or Steam Exit Temp.	Change in Fuel/Feed ratio, Feed injection Sat. Steam injection	Fuel/Feed ratio
Drum Level Control	Water Level	Adjustment of Feed Reg. Valve	Steam Drum Press. Steam Flow Firing Rate % Excess Air Pitch or Roll Movement of vessel Rate of Load damage			
Firing Rate	Steam Exit Pressure	Change in Fuel Admission	Drum level Steam Exit Temp. Steam Exit Press. Forced Air Supply	Load required	Change in Fuel flow by fuel pump speed control	Forced Air Supply
Air Rate	Fuel Flow Rate	Adjustment of Vanes (or speed control)	Firing Rate Drum level Comb. Chamber Pressure Air Temp.	Fuel Flow Rate	Adjustment of vanes and fan speed control	Firing Rate
Feed Rate	Drum Water Level	Adjustment of Feed Reg. Valve	Steam Drum Press. Steam Flow Firing Rate % Excess Air Pitch-Roll Movement Rate of Load change	Load Demand	Change in Feed flow by feed pump speed	(Fuel/Feed Ratio) i.e. Firing Rate
Propulsion Turbine Speed	Load Reg.	Adjustment of Throttle Valve and/or nozzle admission valve	Steam Press. Steam Temp. Throttle opening	Turbine speed always corresponds to amount of steam flow steam pressure before nozzle constant. Variation in boiler exit pressure	Throttle opening	Steam press. Steam Temp. Throttle Opening

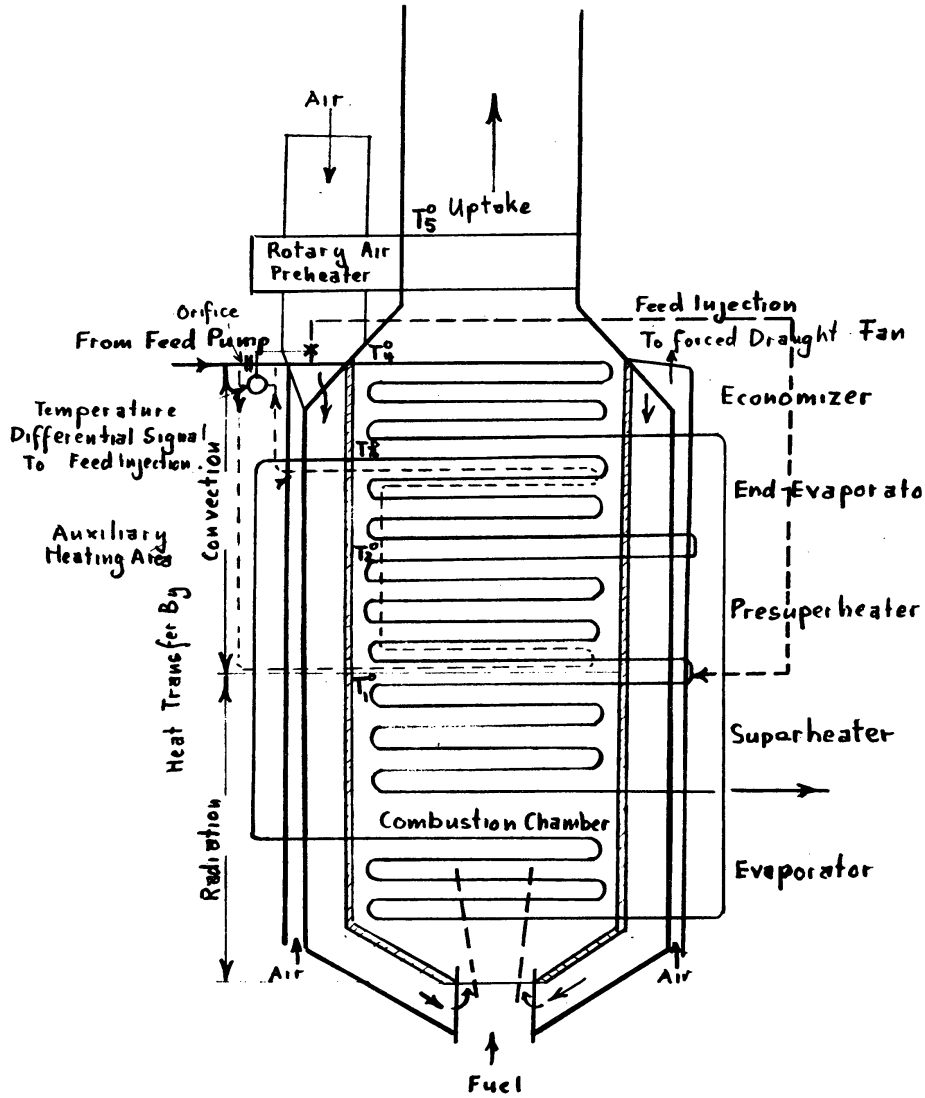
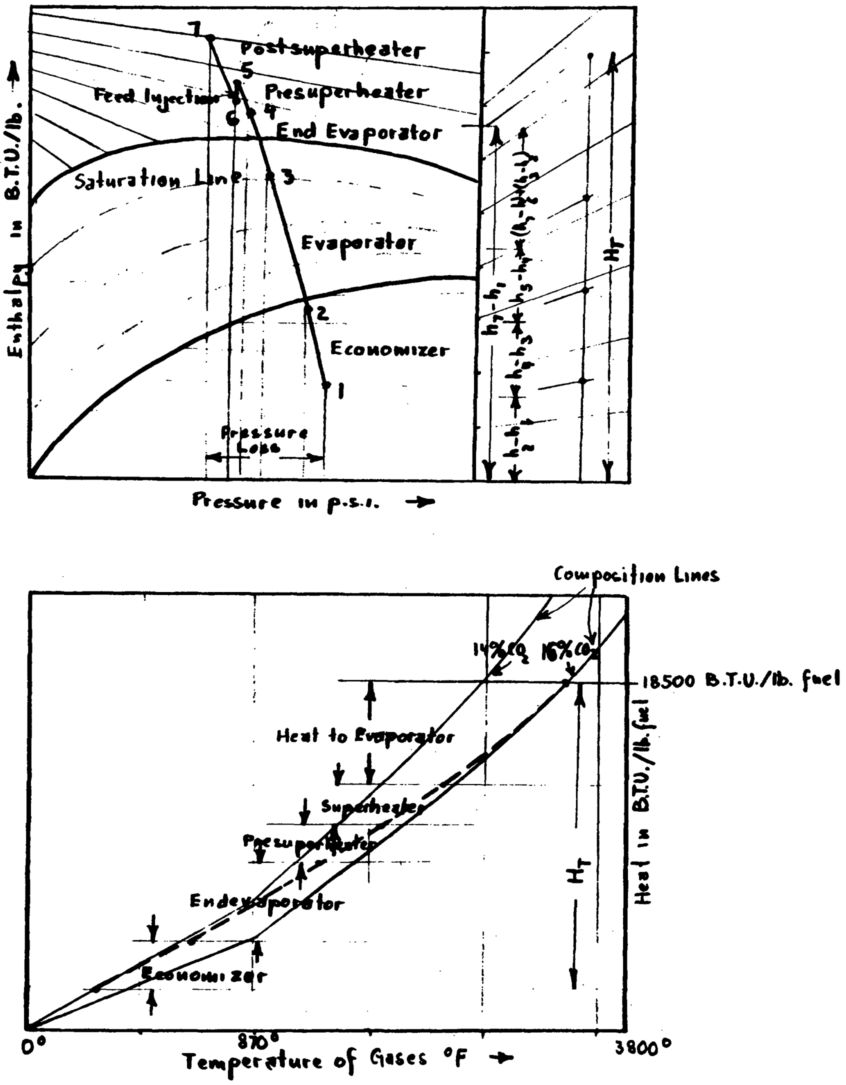


Fig. 15. Diagram of boiler regions.

of installation of the smaller tubes was vastly greater. On the other hand, an increase of 17% of circulating pump power was required for the one inch diameter tubes. It was therefore decided to use a standard condenser with 3/4″ tubes designed for vacuum of 28.5″ Hg. at a sea temperature of 72°F.

The Monotube Boiler

The dynamics and possible control configuration of power plants with high inertia (drum) boilers has been discussed previously. Any possible control scheme starts with the boiler following criterion, as the required rate of load change is far less than the high inertia of this type of boiler permits. The use of the capacitance of the boiler has been seen to permit full usage of the low rotating machinery inertia with consequent fast load changes. On the other hand, it has been shown that using this capacitance complicates the control scheme by introducing additional variables like water level and superheated steam temperature, in addition to steam pressure, steam flow rate, feed water flow rate, fuel rate, and air rate. It was furthermore shown that dependencies exist among most of these variables resulting in a multitude of feedback loops (Table 2). Considering superheated steam temperature control, for instance, it was found that although the control signal is normally provided by exit temperature, there are four factors actually influencing the variables:



h-p Medium and H-T Combustion Gas Diagram
(After Münzinger and Rosin) Not To Scale

Fig. 16

1. Temperature of steam at entry to superheater
2. Steam flow
3. Firing rate and position of superheater
4. Fuel/Air ratio

The once-through boiler in comparison can be shown to provide a low inertia boiler wherein most of the above variables are independent, and such dependencies as exist are only in the form of a ratio of two of the variables. The monotube, or once-through boiler consists essentially of a long tube into which feed water is fed at one end and superheated steam exits at the other end. The tube can be divided into an economizer, evaporator, and superheater region; but the lengths of these regions are not well defined as both the start and end point of evaporation vary with firing rate or throughput. The ratio of water in the boiler to normal designed boiler output is very small, and consequently the thermal capacity of the monotube boiler is essentially provided by the boiling water, tube material, and steam expansion in about equal parts (Table 1). In the case of the drum type boiler, the boiling

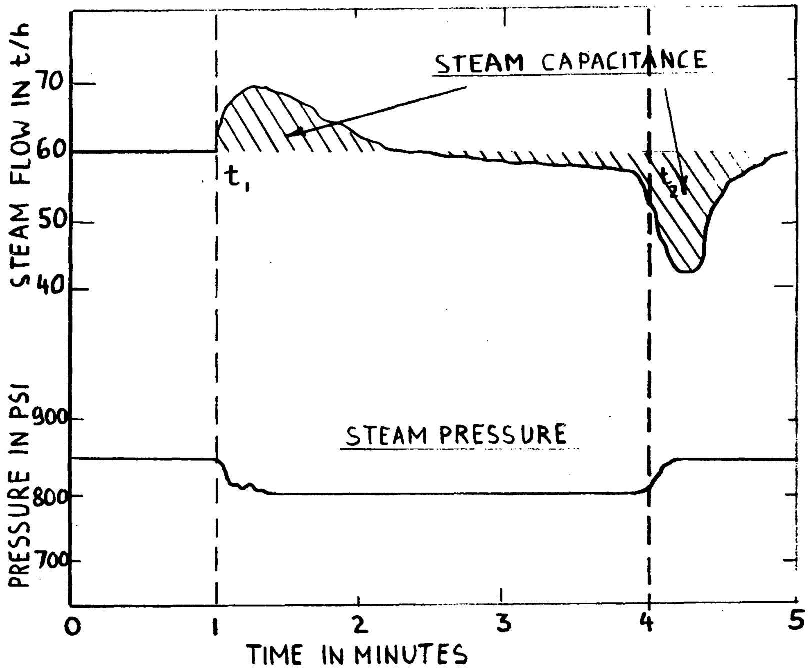


Fig. 17. Measurement of thermal inertia.

water with the refractory provide most of the thermal capacity. The thermal capacity of a monotube boiler is normally evaluated by experiment in which the firing rate is maintained constant and the exit pressure is suddenly changed (Fig. 17). Measuring the new steam flow rate, which is above or below the steady state flow rate for the new pressure at the particular constant firing rate, the thermal capacity of the boiler is obtained. This is found to be manifold smaller than that of a drum type boiler of equal capacity $\left(\frac{1}{20}\text{th to }\frac{1}{60}\text{th}\right)$.

Using the thermal capacity, the pressure of the monotube boiler reacts similarly to that of the drum type boiler with load changes; but as pressure variations are inversely proportional to the thermal capacitance, these pressure changes would therefore be much greater. If boiler following control is used with such a boiler, it is usual practice to provide regulation through a by-pass or dump regulator whereby the thermal capacitance of the boiler is not used and stream flow rate is directly regulated by firing rate.

If, on the other hand, turbine following control is adopted and steam pressure or load is regulated by firing rate, the capacitance of the boiler becomes operative and this results in temporary changes in the superheater length which has a beneficial regulatory effect on the steam temperature during transient conditions (Fig. 18).

The heat transfer rate per unit steam flow rate rises with load in convective heat transfer surfaces and falls in radiation surfaces. Inasmuch as the superheater region is arranged predominately in the convective region, the length of the superheater required falls with an increase in load. On the other hand, both the starting, as well as end point, of the evaporator region move towards the exit with an increase in load. Superheat temperature becomes practically independent of load changes. In fact, it can be shown that the superheater temperature is only a function of the ratio of flow rate to fuel, or firing rate, which can be maintained constant as long as constant superheated steam temperature is desired.

On occasion, to reduce steam turbine thermal shock, it may be desirable to change steam temperature by changing the ratio of flow rate to firing rate. Inasmuch as the whole evaporator region is moved with every load variation, there is a

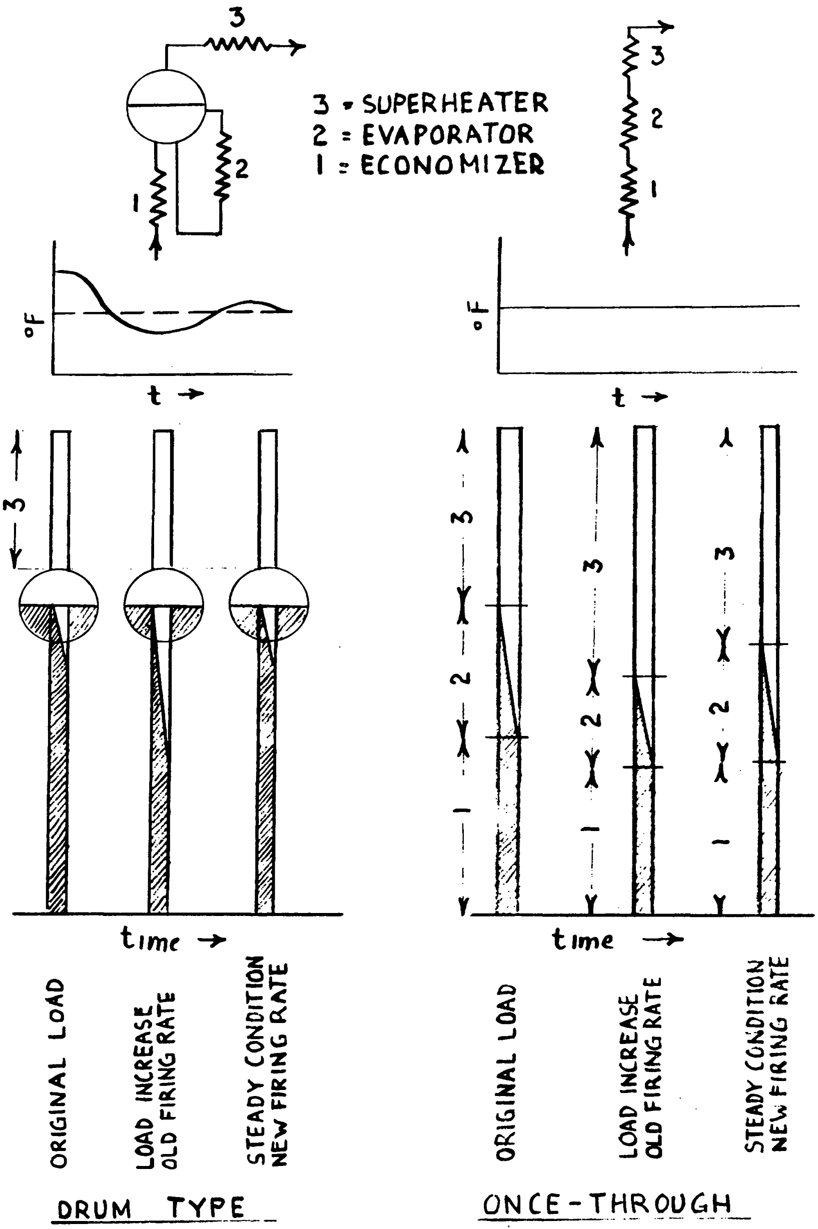


Fig. 18. Effect of load increase on boiler superheat temperature.

different mass of fluid in the boiler at every load. This fact has a profound influence on the dynamic response of the boiler.

To guard against overheating the end point, the evaporator region is normally placed in a cooler combustion chamber space, or, as in the Sulzer design, actually fixed by fitting a small separating drum outside the actual boiler. With the recent rapid developments in demineralizer technology, it is the author's opinion that installation of high flow rate mixed bed, full flow, demineralizers will be justified. Not only will high quality water be guaranteed, but simultaneous protection against condenser leakage will also result.

Water impurities are the main cause for boiler malfunctions, defects, and resulting unavailabilities. It should be recognized that water purity is also a prerequisite to low turbine maintenance and long term high turbine efficiency through the prevention of carryover.

Using turbine following control with a once-through boiler, firing rate becomes the independent variable, with feed flow rate and air flow rate as dependent

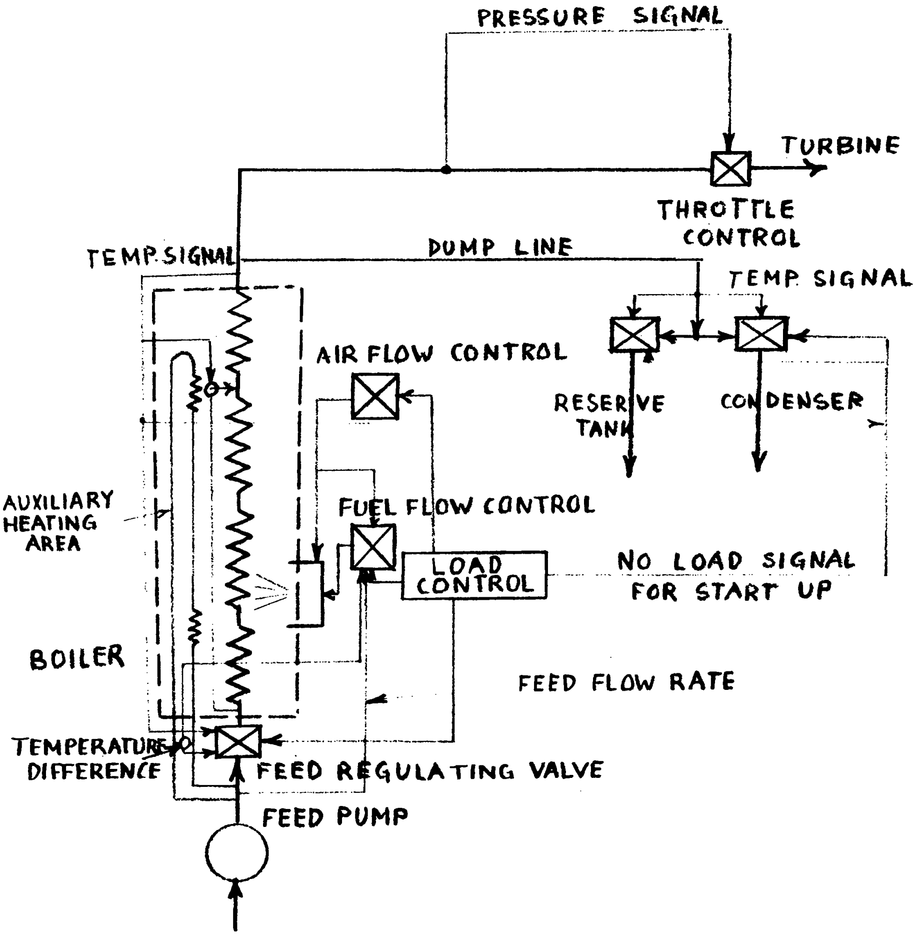


Fig. 19. Monotube boiler control system.

variables (Fig. 19). As mentioned before, it can be shown that for constant superheater temperature operation the fuel/feed flow ratio is kept constant so that feed flow becomes a linear function of firing rate. A difficulty arises during start up though, as constant temperature operation at constant fuel/feed ratio presupposes constant outlet pressure. Or in other words, only when the desired pressure is established will the outlet enthalpy, and therefore the outlet temperature, be controlled by the ratio. An additional start up problem to be considered is the maintenance of circulation. During start-up recirculation and/or dumping of heated water and low pressure steam will have to be arranged.

Several possible schemes will be discussed when component details are evaluated. As start-up is extremely rapid, energy losses will obviously be negligible and consequently the least expensive and simplest design will probably serve the purpose. Inasmuch as firing rate directly determines the amount of steam produced, supply of steam to the turbine and other elements must be regulated by a pressure control system in such a manner that line steam pressure is always maintained within set given limits.

A difficulty arising in controlling the monotube boiler by direct firing rate is the relative discrepancy in the time rate. The through-flow of the water is a time function normally one order of magnitude larger than the change in burning rate. For a typical 50t/h boiler, the firing rate can be increased from 10% to full load in about 5 seconds, while it takes about a minute to reach the new fluid flow rate through the boiler. Therefore, some regulation of the fuel/water rate will be required. An impulse relying on superheat temperature as a regulating function is probably too late. An improvement can be achieved by incorporating an auxiliary heating area consisting of a single thin tube with water through-flow. This through-flow is proportioned to the total water through-flow and is not evaporated but only heated. The difference in inlet and outlet temperature from this tube gives an indication of the fuel to water rate change without the long time delay and is therefore useful as a regulating input function. The disadvantage of such a device is the dependence of the temperature rise on load and the CO_2 content, but it can be shown that this dependence is not too sensitive and does not interfere with the effectiveness of the signal.

If accurate superheater temperature control above the fairly constant self-regulating control of the monotube is required, water injection just prior to the last superheater stage or a combination of a first impulse before the final superheater stage and a main injection after the superheater is resorted to. The first responds more to temperature variation with regard to time, while the latter adjusts the end temperature.

A major problem in defining exact control functions is limited knowledge of the two phase flow phenomena. Work on obtaining transfer functions for once-through boilers both experimentally and analytically is currently under way at M.I.T. with the experimental work being done in part at the Naval Boiler and Turbine Laboratory in Philadelphia and at the M.I.T. Projects Laboratory and the Analog Computer Center.

It is recognized that certain disadvantages are introduced by the once-through boiler. For example, it is probably more sensitive to feed impurities than any other type of boiler. Its regulation is also very sensitive. There is a large pressure loss in the boiler requiring higher feed pressures with a consequent energy loss. On the other hand, the advantages introduced by the low inertia, low weight, once-through boiler are manifold and are considered to outweigh the aforementioned considerations. A few of the advantages of the monotube boiler are as follows:

Universal pressure boiler
Fast economical starts due to low boiler thermal capacitance
Great flexibility during hot restarts
Economical low load operation
Quick cooling of boiler and availability for inspection
Wide temperature control range
Simplified controls
Quick response to load changes
High reliability and availability
Ability to match turbine temperature
Ability to maintain steam temperature with poor fuel and unusual proportions of fuel content
Reduced weight and space (Fig. 20)
Ability to use supercharging effectively
No gage glasses and required instrumentation
No water level control (feed regulator, feed check)
No steam temperature control
No blow-down system
No steam separating equipment
No distant reading drum level gage
Can be adapted to specific space requirements
Assured positive circulation at all loads
Reduced thermal stresses
Lack of large refractory masses enables safe immediate shut down
Reduction in construction time (no drums required)

The main feature of the proposed light weight plant is the once-through boiler. Two main types of monotube boilers have been used extensively. The Benson type

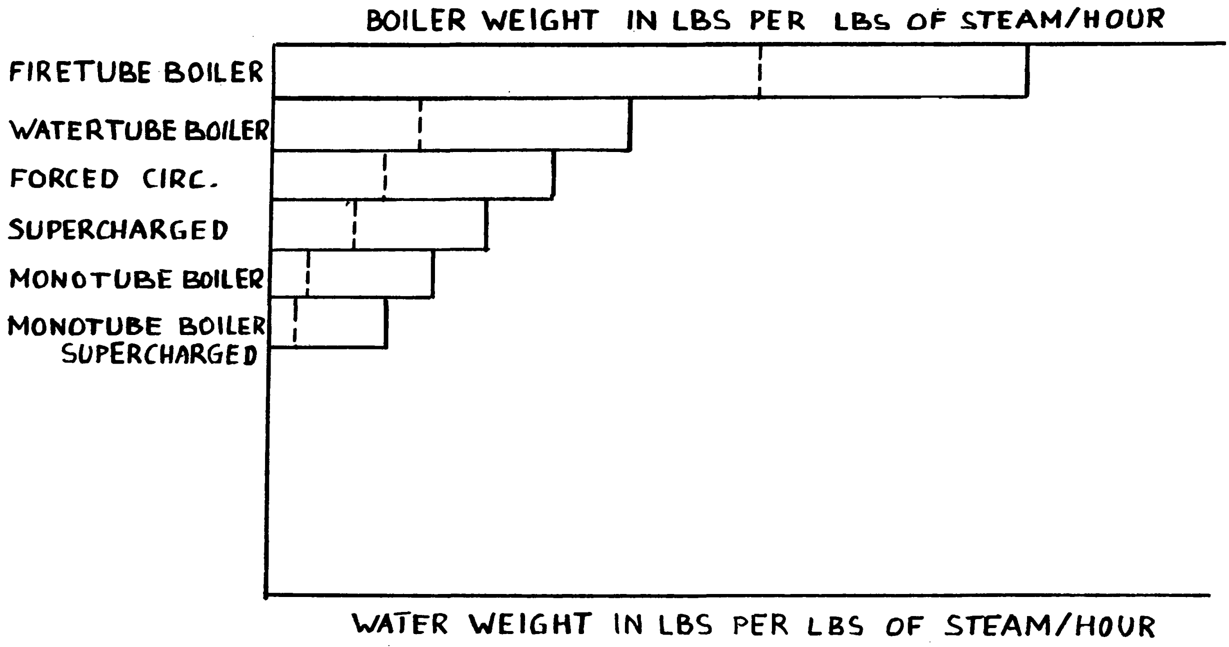


Fig. 20. Comparative boiler weight.

uses essentially a single tube divided into preheating, evaporating and superheating regions with the varying end point of evaporation arranged in a cool space of the boiler to reduce deposits of solids. The Sulzer type has a small separating drum arranged at the exterior of the boiler, at the end of the evaporating length in which it separates the moisture from the steam. A primary difference between the two approaches is that the first does not require as pure feed. Further, the Sulzer boiler has a constant superheater length which thereby requires the usual amount of superheat temperature regulation. In the straight once-through boiler, with varying superheater length, the temperature is essentially constant for constant fuel/water ratios. An important point to consider, though, is that this elementary consideration has a difficulty in that there exists a relative discrepancy in time rate between through-flow of the water and fuel flow to the burner. It is believed, though, that this transport lag can be incorporated into the respective control of the feed and fuel pumps. After consideration of the costs and operational problems involved in using demineralizers to assure the required water purity, it was decided to propose the drumless once-through boiler.

Water purity is to be maintained by full flow demineralizers. Some details of their use are discussed in Appendix II.

Fuel atomizers with full range turn down ratio 8% → 100% full load requirement could be installed. Both spill valve pressure burners and steam assisted pressure jet burners were considered. While the first system shows a slight economic advantage, the added complications introduced through cooling requirements of return oil and the rather unwieldy control functions influenced the choice of the steam assisted atomizers. The small loss of distilled water during maneuvering conditions is not serious (average consumption of distillate for 24 hours, assuming 4% maneuvering time and 50% of that spent at less than half power, is only 40 gallons of water/average day). The ease of control as well as advantageous effect on combustion deposits make it an attractive choice. It is also envisioned that in-service burner cleaning will be introduced by arranging steam flow through the fuel passages at intervals of about 10 hours. The burners operate without steam-assist through the fuel oil pressure range of 450-150 lbs/in^2 which brings the load to below half power.

Thereafter, steam at about 25 psi (reduced by passing auxiliary steam through an orifice plate) is admitted which permits fuel oil pressure to be reduced down to less than five psi with a resulting output of about 8% full load. Boiler circulation requirements also impose this lower limit. In order to prevent the need for dumping 8% of full load steam to the condenser during stop or standby conditions, electric power requirements are divided into base and variable load functions. A steam turbine generator acting as the base power plant is designed to run at practically constant load, thereby providing a lower limit of steam flow. Inasmuch as the boiler exit pressure remains constant and any variation in pressure results in a change of main turbine throttle opening, only constant pressure steam supply to the generator is provided. With a base load of 600 KW, the generator, together with the steam supplied to the atomizer and other auxiliary steam functions, will easily consume the lower limit of steam generation. With a constant minimum of auxiliary steam demand, the control problem is greatly simplified. A dump line with a pressure controlled valve will be provided to allow flexibility of operation and recirculation during start-up.

Inasmuch as the base load generator will normally be running during main power plant operation only, no auxiliary condenser is to be installed; but, instead, all exhaust is to be led into the main condenser. Condensate line, pumping and control arrangements are greatly simplified, as well as the condensate purifying problem. The electrical load requirements above the base load are to be supplied by one of a set of two diesel generators. This arrangement permits a wide range of flexibility and, in addition, assures enough starting and take-home power.

The number of feed heater stages required was analyzed and it was concluded that two stages of feed heating, using bleeder steam from the L.P. and base load turbines for the first stage heater and bleeder steam from the H.P. turbine for the second stage heater, will result in a most effective system. A very slight increase in cycle efficiency could be obtained by introducing additional stages, but consideration of the economizer size in the boiler eliminated most of these advantages. By using the constant base load generator, a bleeder steam supply to the first stage heater and deaerator is assured together with fairly constant feed temperature to the deaerator and boiler (Fig. 14).

Considering some of the problems of the once-through boiler in greater detail, it was decided to divide the heat transfer regions into five distinct sections for the purpose of this initial study:

1. Economizer region
2. Evaporator region
3. End-Evaporator region
4. Presuperheater region
5. Superheater region

(See Fig. 15)

The economizer region is placed above the end evaporator and designed to raise feed water temperature to 500° F under full load conditions. Convection gas temperature at entry to this section is then about 1120° F and at exit about 840° F. Placed just below the economizer is the end-evaporator. This position is required to reduce any deposit of solids at the end of evaporation. To assure that the end point would be within this convection heat transfer region with combustion gas temperature between 1450°F and 1120° F under all conditions of load and rate of load change, computations for extreme conditions were made. It was found that designing this heat transfer area so that the medium entering with an enthalpy of 125 BTU/lb. below the enthalpy of dry and saturated steam at 930 psi and leaving with an enthalpy 110 BTU/lb. above that of dry and saturated steam at 930 psi will result in the end point of evaporation being at all times within this cooler region.

An estimate of the expected full load pressure loss in the pre- and main superheaters gave a value of 95 psi, which implies a mean expected pressure of about 930 psi in the end evaporator. The presuperheater is the last element placed in the convection region and was designed for a medium temperature rise to 750°F.

In the preceding general discussion of monotube boilers, it was shown that the once-through boiler is essentially a constant temperature steam generator. Yet, due to the inherent possibility of outside changes to combustion conditions, such as change of quality of fuel, composition of combustion gases, fuel temperature, etc., as well as the changes due to steam assisted atomization during low level operation, a feed injection type fine adjustment superheat temperature regulator is installed between the pre- and main superheater. This device receives a signal from the temperature differential of the auxiliary heating area as shown in the diagram. As previously discussed this method assures adjustment before any large temperature excursion. This heating surface is made of 50 feet of 1/2 inch tubing interspaced between end-evaporator and superheater with areas in both the convection and the radiation regions. Both the superheater and the evaporator sections form the sides, floor and ceiling of the combustion chamber and are designed for heat transfer by radiation. The temperature in this region is expected to vary between 3800°F and just above 2000° F.

Assuming a flow velocity of 9 feet per second (referred to water at 68° F in the radiation section, evaporator and superheater), and a tube diameter of 1 1/4 inches (inside), the number of parallel tubes required for the boiler is 12. This results in dry and saturated steam velocities in the end evaporator of about 40 feet per second and mean superheated steam velocities of about 56 feet per second. Each tube therefore handles about 15000 lbs/hour. The pressure losses were computed by summing terms due to friction losses, momentum change, entry losses, exit losses, static losses, and inertia change. The resulting pressure loss for the superheaters was 95 psi and for the economizer 86 psi. Only a rough estimate of pressure loss in the evaporators could be made, and this was based in part on empirical results obtained by the "VGB" (12) during a performance trial of a monotube boiler. This estimate resulted in a value of 139 psi, which gives a total pressure drop of 320 psi. An evaluation of the heat transfer areas required in the different sections resulted in a total tube length of about 1820 feet from economizer entry to superheater exit.

It is not the purpose of this paper to present detailed design results and procedures and these values are therefore given only to show orders of magnitude. An effective tool in monotube boiler design is the use of H-P steam and H-T combustion gas diagrams (Munzinger and Rosin) as shown in Fig. 16.

Assuming a boiler efficiency of 88%, the maximum amount of fuel flow is 6000 lbs/hour and the air flow acquired for complete combustion is a maximum of 32,400 CFM (against 20″ W.G.). The resulting power required for the fuel oil pumps and blowers is 20 and 77 SHP, respectively. Very little is known about the maximum thermal loading in radiation heat transfer regions. The heat transfer rates assumed in marine boilers are vastly greater than those of comparable land installations. With a cylindrical boiler, which results in a more uniform temperature distribution in the combustion chamber, as well as with fuel injection vertically upwards, which results in a reduction of the penetration distance of the fuel particles and consequently a larger temperature gradient as a function of distance, it can be assumed that a 30% reduction in combustion chamber volume is feasible. Furthermore, mixing of the air and fuel can be arranged to be much more uniform and effective due to the symmetry of the combustion volume. It is also suggested in this design to admit part of the air at an angle inclined towards the center of the fire cones at an advanced location, or in other words, admission by stages so that the added air penetrates the cone in a spiral fashion.

The power required for the feed pump is 345 SHP assuming a delivery pressure of 1180 psi. Part of this power is regained in heating the medium. The effect of this energy transfer must be included in the design of the different regions by an iteration process. The total mass of water and steam in the boiler during steady state, full load operation is approximately 6400 pounds. Desuperheating of auxiliary steam is accomplished by feed injection after passing a pressure reduction valve. Other means such as steam-air heater desuperheating were considered but the temperature differential may result in excessive thermal stresses.

Air preheating is accomplished by a Ljungstrom type rotary air-preheater taking its supply from the outer casing space around the boiler and discharging the air via the inner casing to the burners. The walls and part of the conical floor of the combustion chamber are watercooled by the tubes of the evaporator section arranged in spiral form with a continuous upward gradient and ending in a ceiling made up of converging nested spirals. The wall spirals are backed by an up and down arrangement of the superheater tubes which also follow the nested ceiling tubes of the evaporator with a nested arrangement. The end evaporator, presuperheater, and economizer section are all arranged in nested form in the convection region. The vertical superheater tubes forming the combustion chamber walls are backed by the downcoming spirals of the tubes connecting the economizer and evaporator section. Inasmuch as the economizer is designed for a temperature rise to less than 500° F, no steam would normally be formed in this part of the evaporator and consequently a downward gradient of the tubing is justified. No estimate of the refractory thickness required on the combustion chamber walls is available yet, but it can be assumed that apart from the usual insulation around the burners and the boiler floor, very little refractory will be required. Preliminary estimates show that this type of boiler would have an outside diameter of 12 feet and an overall height of 22 feet. Considerations affecting the decision to choose a cylindrical combustion chamber were minimum structural weight combined with strength, permitting slight supercharging with little change in scantlings.

Supercharging this type of boiler even to as little as 30 psi (Velox 37 psi, Naval boiler 67 psi) would result in a saving of about 35% in boiler volume and 18% in boiler weight. It is the author's belief that additional research in combustion, particularly with regard to optimum fuel droplet size distribution and penetration, will result in the effective application of bunker C for burning in supercharged boilers operating at pressures of about 30 psi. The resulting gain in efficiency of about 13% is attractive enough to warrant sustained effort.

Due to the varying mass of medium in the boiler during load fluctuations a reservoir must be provided from which the feed pump can draw its supply. Calculations show that the maximum mass variation, during operation, amounts of 0.8 tons of water at 72° F, a volume easily incorporated into the deaerator. For start-up, additional reserve capacity for the initial filling of the boiler, including evaporator and superheaters with feed water will require 2.2 tons above the normal steaming quantity. This can easily be arranged for by the provision of a start-up reserve tank. This tank can also be used for recirculation through the dump line during start-up, thereby bypassing the condenser. By this means faster start-ups are possible. A thermostatically controlled valve arrangement switches dump line discharge to the condenser after a certain temperature has been reached as steam begins to form.

A comparison of structural cost and weight gains of the low inertia boiler plant, compared to a conventional two drum dual boiler plant of the same power, shows the following approximate results: —

Monotube Boiler			Conventional Boiler
Steaming rate	180,000 Lbs/Hr.		90,000 Lbs/Hr.
Dry Weight	34,000 Lbs/Hr.		108,000 Lbs/Hr.
Wet Weight	40,400 Lbs/Hr.		119,000 Lbs/Hr.
Box Volume	3,500 Ft.^{3}		5,800 Ft.^{3}
	Total for 2 Boilers:	Wet Weight	238,000 Lbs.
		Box Volume	11,600 Ft.^{3}

In addition, it is estimated that main steam pipe lengths can be reduced to less than one quarter, resulting in an additional weight saving of 3 - 5000 pounds.

Total estimated weight savings = 200,000 Lbs.
Total estimated volume savings = 9,100 $Ft.^3$

Sketches of proposed floor and section plans are given in Figs. 21, 22 and 23. Engine room lengths of less than 42 ft. can probably be accomplished. In addition, it should be noted that inasmuch as the proposed boiler diameter is only 12 feet, it should be feasible to fit it into the usual skylight trunk, thereby reducing the engine room height appreciably. It seems that this type of plant would be particularly useful for naval ship application.

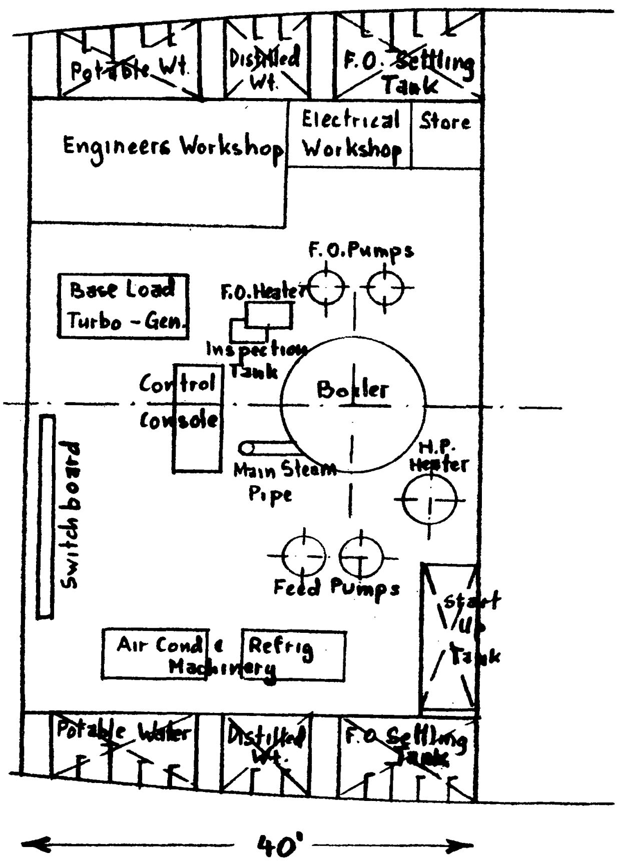


Fig. 21. Plan operating level.

Control of the Low-Inertia-Boiler Power Plant

The main function of power plant control is to satisfy load demand within physical limitations of the plant. Of prime importance in this respect is the maintenance of fairly constant steam conditions at the boiler exit. In both the boiler following, as well as the turbine following control systems, the steam pressure is kept constant.

In the first, the boiler controls attempt to rectify any steam pressure variations resulting from changes in demand of turbine output, while, in the second system, steam pressure is directly maintained by the load change imposed on the boiler, inducing a new flow rate and consequent output on the turbine. In other words while steam flow rate is the independent variable in the first case, firing rate is the independent variable in the second. Considering the proposed plant with turbine following control, a major problem is to adapt feedwater flow to the actual rate of combustion. During any change in firing intensity, fluctuations in both end points of the evaporator region occur (Fig. 18).

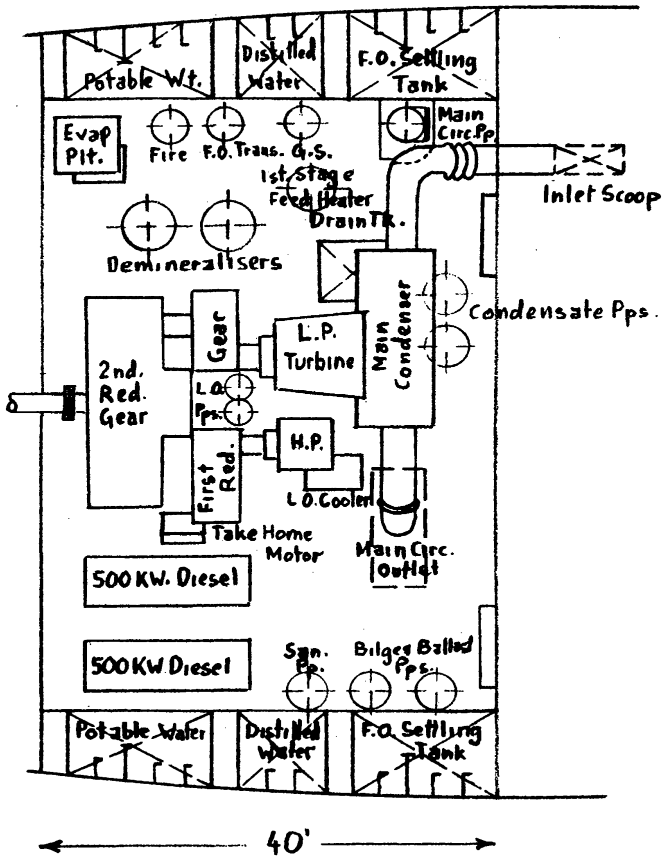


Fig. 22. Plan engine room floor level.

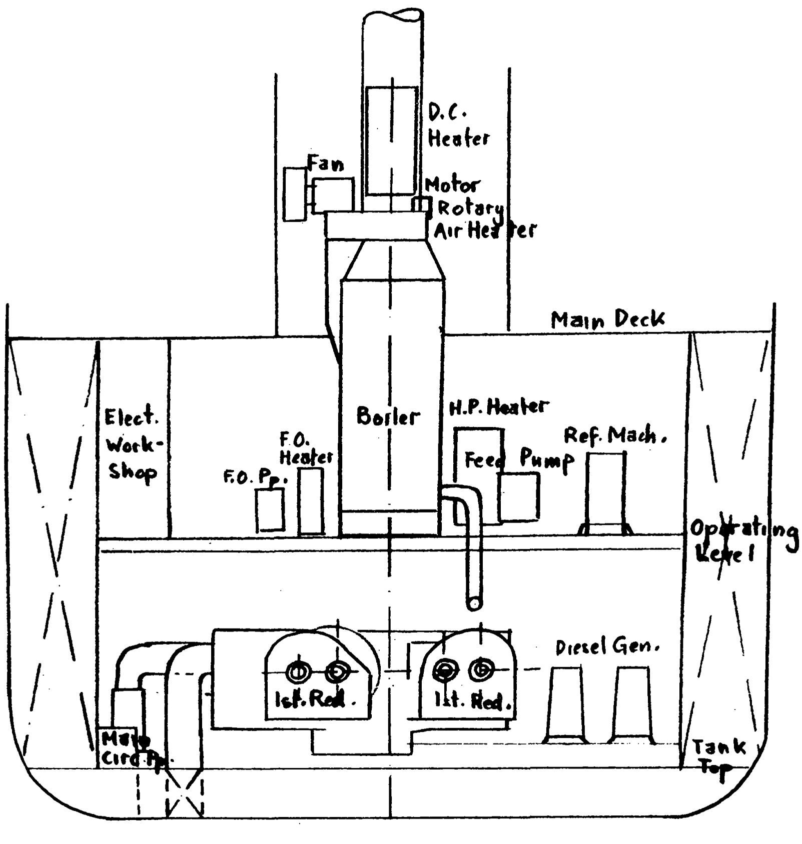


Fig. 23. Section looking forward.

When a load change is imposed, the energy content of the system must therefore be adjusted before the change in firing rate takes full effect by providing a new steam flow rate. Inasmuch as the amount of brickwork is insignificant, and its rate of temperature change is small compared to that of the other materials in the boiler, it can be considered a minor factor in thermal storage change. After any change in firing rate, a new temperature gradient in the tube material must be established. The thermal inertia of the tube material is found to be about equal to that of the fluid medium. It is convenient to obtain values of thermal inertia for a once-through boiler from experimental results giving the response of steam flow at constant firing rate to steam pressure changes imposed from outside (Fig. 17).

Much work is currently underway in the analysis of the dynamic response of two phase flow and heat exchangers in general. The author hopes to report some of the results of current work at M.I.T. in this area in the near future.

The problem of load control by firing rate is essentially the maintenance of a constant fuel/feed ratio and an appropriate fuel/air ratio. The rate of change of air flow is much slower than that of the fuel. Similarly, the time constant for increasing the inertia of the mass of fluid medium in the boiler by a change in feed flow rate is about one order of magnitude larger than that of the fuel flow rate.

Feed flow can be adjusted by either feed regulating valve lift or a variation in feed pump speed. While it is easy to obtain a rapid valve adjustment from a load change signal, regulating valve control requires a large amount of additional power as a result of the large pressure drop.

Considering variable speed feed pump control, a feed pump turbine arrangement in series with, and between, the two superheater elements was investigated (similar to the fuel pump/fan turbine series arrangement proposed by Allis-Chalmers, etc.). This type of control is feasible if the flow rate change of the fuel pump is reduced to below that of the feed pump by appropriate step delays.

Due to the fact that electric power is required for circulating feedwater during the entire start-up period, an electric motor drive is, however, recommended for the plant. Another concept would be to use an electrically driven constant speed pump preceding a turbine driven variable speed pump. The final pressure of the first stage pump is sufficient for starting the boiler at full pressure and running at about 10-20% load. The turbine driven pump, therefore, is only started when steam is available.

After analyzing speed flow rate responses of different feed pumps, it was apparent that feed regulating valve control would result in a more responsive system. The control of valve lift is affected by both a signal of load demand as well as steam outlet temperature and the auxiliary heating area differential temperature signals. The last will result in a feedback control loop relating feed flow to actual firing rate. Firing rate is directly controlled by load demand with an appropriate delay, and auxiliary signals from feed flow rate, auxiliary heating area, and air flow rate.

It should be noted that it is hoped that most of the auxiliary signals will not be required during normal operation even under rapid changes in load. They are presently proposed in order to account for the lack of reliable analysis of the dynamic response of some of the system parameters. It is expected that a majority of controls can be eliminated as more experience and insight is gained. Eventually, it is hoped to control the plant with a single load demand signal to the fuel, air, and feed controller.

During start up and shut down periods a "No Load" signal will actuate the control for the dumping line to divert the circulating flow to either the condenser or reserve tank according to system requirements. The control of the dump line valves will be arranged to open at about 40 psi below normal operating pressure in the "On Load" condition during stop commands to accommodate steam flow not used by the base generator (if the base load is below minimum boiler output). As soon as the order is changed to some higher load, the dump control is put back to its normal "On Load" condition where it will only admit steam at 25 psi above normal operating pressure. A pressure range of 810 to 875 psi is therefore provided for throttle lift adjustment to correspond to the resulting steam flow rate to the main turbine. Many specific control problems have been considered as well, yet the scope of this paper does not permit their detailed discussion.

A significant advantage of the proposed control scheme is the ease with which the plant is shut down during casualty conditions. If feed flow fails, the immediate response of decreasing fuel flow results in a safe condition. The low thermal capacity of the boiler will prevent any overheating. Similarly a failure in the combustion system will result in shut down of the feed pump and immediate actuation of the "No Load" signal to the dump line control, which in turn diverts the remaining steam and water to the condenser, while at the same time prevents carry-over into the turbines.

Conclusions

An attempt has been made herein to present the case of the low inertia/low weight power plant as a possible means of achieving a more competitive, compatible, and controllable marine steam turbine plant. The fact that many of the components and concepts have not been tried in the marine environment should not preclude further investigation of this radical departure from conventional marine practice.

It is recognized that many problems exist and many questions are as yet unanswered, but the prospective gains achievable warrant a sustained effort. These gains include low weight, fast response, small volume, ease of control, ready access for repairability on short notice, ability to take advantage of supercharging, and many more. It is hoped that this discussion will result in the stimulation of some effort to develop the turbine-following principle to the control of marine steam power plants so that the apparent advantages can be realized in practice. The suggested plant is considered a first attempt, and changes of parameters, control loops, logic, and dimensions will, no doubt, be required before a practical design is developed. Yet, it is hoped that this effort will be successful in providing a starting point with regard to an optimum design of an integrated marine steam power plant.

APPENDIX I

FEED WATER REQUIREMENTS FOR ONCE-THROUGH BOILER

Total Dissolved Solids	0.050 ppm
Dissolved Oxygen	0.007 ppm
Carbon Dioxide	0.00
Organic	zero
Silica	0.020 ppm
Iron as Fe	0.010 ppm
Copper as Cu	0.005 ppm
Hardness	zero
Free caustic	zero
pH	8.5 - 9.2

APPENDIX II

A DISCUSSION OF DEMINERALIZER USE IN STEAM TURBINE PLANTS

Demineralization is affected by ion exchange with the aid of specially prepared resins. In this process the water at first gives up cations in exchange for hydrogen ions in what is called the cation exchanger, and thereafter it gives up its anions for hydroxyl ions in the anion exchanger. The radicals of any dissolved gas thus remain in the cation exchanger, while the ions formed there combine again to form water. This process continues as long as exchangeable ions are available in the resin. When these are used up, the resin is exhausted and has to be regenerated by acids or lyes to be usable again. In some arrangements the exchange process between cations and anions is produced in separate units (two bed plan), while in others the process takes place simultaneously in a single bed by mixing both cation and anion exchange resin. Demineralization is normally assisted by preliminary salt removal (distillation, chemical, precipitation, lime treatment, etc.) and/or by deaeration to remove the CO_2 in order to assist the anion exchanger. Prior filtration to remove non-dissolved impurities should also be introduced. A comparison of mixed bed and two bed plants shows that the first will result in purer water, while the second is more economical with respect to regeneration of chemicals consumed, and also requires regeneration only at much longer intervals. However, the mixed bed will also consume more resin by erosion.

Non-dissociating substances like oxygen and organic compounds are normally not affected by the ion exchange process as such.

Considering the demineralizing effects in order, radicals and acid residues belonging to the strong bases are exchanged first, next follow the ions of the weaker bases and acids. Consequently, in boiler water the sequence of elimination is as follows:

Sulphates, chlorides, CO_2 and silica.

Silica is therefore the last to undergo exchange, which means that it is removed in a zone nearest to the outlet from the exchanger. As the resin approaches exhaustion the absorption zone of all substances is shifted towards the outlet, and consequently silica will be the first to escape exchange. The temperature limit at which exchange takes place is largely independent of the exchange process, yet demineralizers must be guarded against operation at excessive temperature by protective devices.

In this steam plant demineralization serves two distinct purposes:

1. Purification of feed water
2. Safeguarding boiler against defects from leakages.

Providing a full flow demineralizer could, therefore, increase plant reliability. A problem arises normally in full flow demineralizer capacity where make-up or auxiliary demand is superimposed on peak throughflow, which may result in an excessive demineralizer size. Arrangements have to be provided, therefore, to have continuous make-up supply.

As units have to be taken out of service for regeneration, parallel installation of at least the in-line plant is required. The time for regeneration is judged by the silica content, pH value, and water conductivity as a function of water throughput. The regeneration process as such consists of backwash, supply of regenerant, rinsing, and running in to starting point. The regeneration cycle is easily programmed and automatically controlled (hydraulic). Safety devices to guard against early exhaustion and "carryover" are also normally fitted. One possible device is placing a completely separate protective exchanger downstream which is never allowed to become exhausted.

In conclusion it may be said that demineralizers are as reliable as other major components of the steam power plant today. The cost of operation is only slightly above the cost of chemical water treatment, and the initial installation cost can easily be amortized by the savings in internal boiler cleaning. The constant high quality of the feed with the resulting reduced maintenance and optimum plant (boiler and turbine) efficiency throughout the life of the plant appear to make this approach to water purification worthy of serious consideration.

APPENDIX III

NOTES ON POWER PLANT DYNAMICS

DYNAMICS OF THE STEAM TURBINE

In control equations for the steam turbine, we normally assume infinite steam capacity of the boiler and, consequently, constant steam pressure and temperature. Even using low capacitance boilers, the arrangement of a dump valve bypass (boiler following) or pressure-controlled throttle valve (pressure control-turbine following) allows us to make the above assumption. In plants without reheating, time lags due to turbine clearance spaces play only a minor part in the transfer function. It can be shown though that the appreciable volume of the piping between the two cylinders of a turbine results in a time lag of the L.P. response behind the H.P. response, which is quite respectable. Considering the turbine then, if

M_{to} = steady-state moment applied

M_c = internal friction torque

M_{pg} = resisting torque of propeller gear and shafting

J_1 = moment of inertia of rotor of single-cylinder turbine, shaft, and pinion

J_2 and J_3 = moment of inertia of the intermediate and of the bull gear, shaft, and propeller gear.

Then, $$J = J_1 + J_2\left(\frac{\omega_2}{\omega_1}\right)^2 + J_3\left(\frac{\omega_3}{\omega_1}\right)^2 = \text{moment of inertia of system.}$$

(ω_1, ω_2, ω_3 are angular velocities of shafts.)

$$\frac{J\, d\omega_1}{d\omega_t} = \Delta M_t - \Delta M_c - \Delta M_{pg} \tag{1}$$

Now, $M_t = f(p_1, t_1, p_2, \omega_1)$

$$M_t = M_{t_0} + \frac{\partial M_t}{\partial p_1}\Delta p_1 + \frac{\partial M_t}{\partial t_1}\Delta t_1 + \frac{\partial M_t}{\partial p_2}\Delta p_2 + \frac{\partial M_t}{\partial \omega_1}\Delta\omega_1 + \text{terms of higher order}$$

and $$\Delta M_c = M_c - M_{co} = \frac{\partial M_c}{\partial \omega_1}\Delta\omega_1$$

Similarly, $$\Delta M_{pg} = M_{pg} - M_{pgo} = \frac{\partial M_{gp}}{\partial \omega_1}\Delta\omega_1$$

and $$\Delta M_t = M_t - M_{t_0} = \frac{\partial M_t}{\partial p_1}\Delta p_1 + \frac{\partial M_t}{\partial t_1}\Delta t_1 + \frac{\partial M_t}{\partial p_2}\Delta p_2 + \frac{\partial M_t}{\partial \omega_1}\Delta\omega_1$$

(1) then becomes--

$$J \frac{d\omega_1}{dt} = \frac{\partial M_t}{\partial p_1} \Delta p_1 + \frac{\partial M_t}{\partial t_1} \Delta t_1 + \frac{\partial M_t}{\partial p_2} \Delta p_2 + \frac{\partial M_t}{\partial \omega_1} \Delta\omega_1 - \frac{\partial M_c}{\partial \omega_1} \Delta\omega_1 - \frac{\partial M_{pg}}{\partial \omega_1} \Delta\omega_1$$

where t_1 = inlet temperature

t_{10} = steady-state inlet temperature

p_1 = inlet pressure

p_{10} = steady-state inlet pressure

p_2 = back pressure

p_{20} = steady-state back pressure

if $\varphi = \frac{\Delta\omega}{\omega_0} \quad \rho_1 = \frac{\Delta p_1}{p_{10}} \quad \rho_2 = \frac{\Delta p_2}{\rho_{20}} \quad \xi = \frac{\Delta t_1}{t_{10}}$

Then, by using the dynamical constants,

$$T_\varphi = \frac{J}{\frac{\partial M_{pg}}{\partial \omega} + \frac{\partial M_c}{\partial \omega} - \frac{\partial M_t}{\partial \omega}}$$

$$T_\xi = \frac{J\,\omega_0}{\frac{\partial M_t}{\partial t_1}\, t_{10}}$$

$$T_{\rho_1} = \frac{J\,\omega_0}{\frac{\partial M_t}{\partial p_1}\, p_{10}} \quad \text{and} \quad T_{\rho_2} = \frac{J\,\omega_0}{\frac{\partial M_t}{\partial p_2}\, p_{20}}$$

We obtain
$$\frac{d\varphi}{dt} = \frac{\rho_1}{T_{\rho_1}} + \frac{\xi}{T_\xi} + \frac{\rho_2}{T_{\rho_2}} - \frac{\varphi}{T_\varphi} \qquad (2)$$

as the dynamic equation of the system. The dynamic constants are obtained from the characteristics of the turbine set. The applied torque, for instance, is

$$M_t = \frac{3600\, G h\, \eta_e}{860\, \omega_1} \text{ (Kg mtr)}$$

$$= \frac{2 \cdot 95}{860} \frac{G h\, \eta_e}{\omega_1} \text{ (lbs ft)}$$

where G = steam flow in lbs/sec

n = enthalpy drop in turbine

η_e = thermal efficiency of turbine

ω_1 = angular velocity of turbine

and
$$\frac{G}{G_o} = \frac{p_1}{p_{10}} \sqrt{\frac{t_{10}}{t_1}} \sqrt{\frac{1 - (p_2/p_1)^2}{1 - (p_{20}/p_{10})^2}}$$

as $p_2 << p_1$.

$$\frac{G}{G_o} = \frac{p_1}{p_{10}}\sqrt{\frac{t_{10}}{t_1}}$$

where p_1 = new inlet pressure

p_{10} = steady-state inlet pressure

t_{10} = steady-state inlet temperature

t_1 = new inlet temperature.

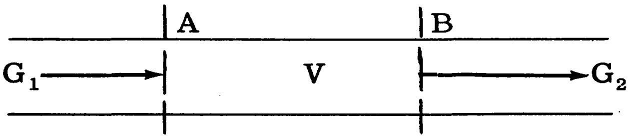
The above expressions are applicable where turbine speed is controlled by throttle setting without altering the number of nozzle openings, and steam flow is the independent variable.

If speed control is obtained by nozzle openings only, assuring constant p_1 and t_1, then

$$\Delta M_t = \frac{\partial M_t}{\partial G} \Delta G + \frac{\partial M_t}{\partial \omega_1} \Delta\omega_1$$

from which the dynamical equation for the turbine is obtained again.

Dynamics of Steam Spaces



if $G_{10} - G_{20} = O$ where G_{10} = initial steam flow past A

G_{20} = initial steam flow past B

G_1 = new flow rate into volume V

G_2 = new flow rate out of volume V

$(G_1 - G_2)\,dt = Vd\gamma$ = specific weight of steam in pipe

$\Delta G_1 = G_1 - G_{10}$ $\Delta G_2 = G_2 - G_{20}$

$p\gamma^{-n} = p_0\gamma_0^{-n}$ (assuming polytropic change)

if $W_0 = \gamma_0 V$ = mass of steam in pipe at time zero.

$$\frac{W_0}{np_0}\,dp = (\Delta G_1 - \Delta G_2)\,dt$$

but $G_1 = f(p_1, t_1, m_1)$

= f(pressure, temperature, control coordinate)

$$\Delta G_1 = \frac{\partial G_1}{\partial p} \Delta p + \frac{\partial G_1}{\partial m_1} \Delta m_1 + \frac{\partial G_1}{\partial t_1} \Delta t_1$$

and again the characteristic equation can be written as:

$$\frac{\partial \rho}{\partial t} = \frac{\mu_1}{T_1} - \frac{\mu_2}{T_2} - \frac{\rho}{T_3}$$

$$T_1 = \frac{W_0}{n \frac{\partial G_1}{\partial m_1} m_{1_{max}}} \quad \text{and} \quad \mu_1 = \frac{\Delta m_1}{m_{1_{max}}}$$

$$T_2 = \frac{W_0}{n \frac{\partial G_2}{\partial m_2} m_{2_{max}}} \qquad \mu_2 = \frac{\Delta m_2}{m_{2_{max}}}$$

$$T_3 = \frac{W_0}{n\left(\frac{\partial G_2}{\partial p} - \frac{\partial G_1}{\partial p}\right)p_0} \qquad \rho = \frac{\Delta p}{p_0}$$

where the dynamic constants T_1 , T_2, and T_3 have the dimension of time.

This type of analysis is useful in finding the time lags due to the steam space between the turbines, as well as that due to the main steam pipe between the boiler and the H.P. throttle. The functional relationship of G, the flow rate, will obviously have to admit any particular relationships consistent with system operation. The results show a definite hump in the dynamic response curve for most two-cylinder turbines.

A similar effect which is even more pronounced occurs with reheating. For instance, after an increase of load, the reheater pressure must first be raised, which results in a time lag which is a function of the storage capacity of the reheater space.

Dynamics of the Firing and Feed System

It is normally convenient to keep the dynamic behavior of the firing system separate from the dynamics of the steam generator itself. Profos (9) has shown that, in general, firing systems can be characterized by a finite distance velocity lag T_1 and a time constant Z which written in complex form essentially results in an equation of the form--

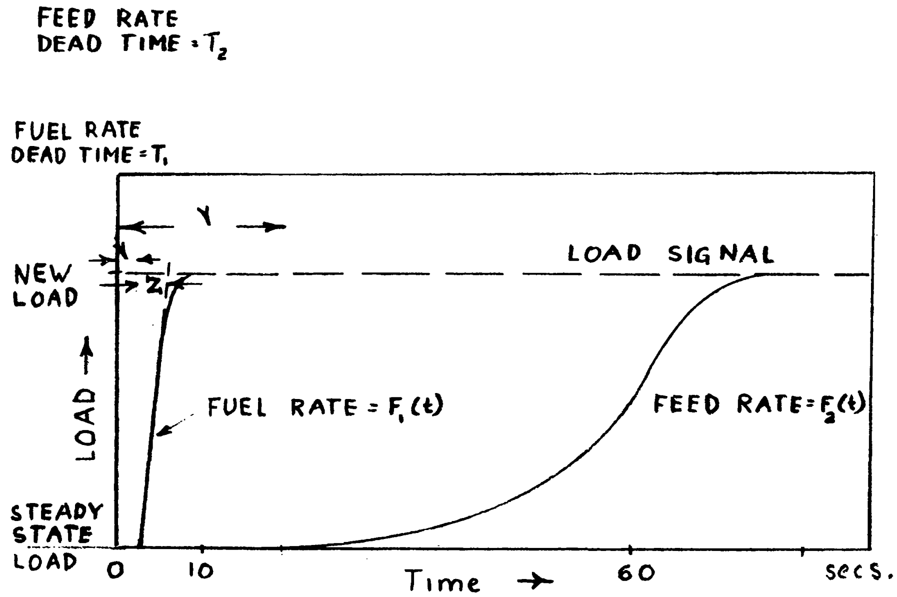
$$\text{Firing rate} = F_1\,(p) = \frac{e^{-t_1 p}}{1 + Zp}$$

where T and Z are obtained from the response curves of the system. As $T \rightarrow 0$ in most oil-fired systems, a general equation for C_F = load control signal = $F_1 + Z_1 F_1{}'$ (from which the simplified complex form is derived) can be used. The value of the time constant Z_1 has been found to vary from 2 to 4 seconds.

Much more work in studying firing system response and the resulting time constants has to be undertaken to allow a deeper understanding of the behavior during load changes.

Feed-flow response can be expressed by a similar equation, but velocity lag T_2 will be of the order of 4 - 8 seconds. Similarly, several time constants (Z_1, Z_2, etc.) will be required for the characteristic response equation, and some of these will be an order of magnitude larger than the time constants of the firing rate. Fig. 1 shows the dynamic response of the two main components of the direct load control system.

While it was originally planned to have fuel and feed pumps on a single shaft, the difference in F_1 (t) and F_2 (t) would require a complex feedback loop operating a fuel bypass as a function of feed-flow rate in order to keep the fuel/feed ratio constant. While not discarding the possibility of single shaft load control, more work is required in dynamic response analysis before such a system can be designed.



Appendix III. Fig. 1. Firing and feed system response.

Dynamics of the Steam Generator

The dynamic response equation of the boiler is made up of terms relating firing intensity with virtual steam generation, pressure drop, and storage of the boiler components. The energy content (iron plus medium) is a function of boiler load at any time. This can be obtained for the monotube boiler by plotting the heat content of the tube system per unit length over the tube length for various loads, and then integrating heat content over the total tube length. Plotting heat content (H) against load (Q) (as heat transmitted per unit time), we obtain the time constant for virtual steam production or following Profos (9) (see also Fig. 13)--

$$F_1(t) = S_v + \frac{\partial H}{\partial Q} S_v^{1}$$

where S_v = virtual steam generation

and $\frac{\partial H}{\partial Q}$ = time required to increase heat content of boiler to new steady state after load increase (unit step).

The effect of the pressure drops in the different parts of the boiler is normally introduced by considering an equivalent pressure drop at the boiler outlet. Boiler storage is included by considering its relation with the time rate of change of boiler pressure at constant load, as shown in Fig. 17. The working medium balance will at any time consist of the sum of the virtual steam generated and boiler storage.

Combining the equations for virtual steam generation, equivalent pressure drop, boiler storage, and medium balance, we obtain the differential equation of the steam generator which gives us the required relation between firing rate and actual steam generation.

APPENDIX IV

RELIABILITY ANALYSIS UNDER THE CONSTRAINTS OF AN INTEGRATED DESIGN

Introducing reliability as a parameter into the design stage of a marine power plant complicates the optimizing procedure. We can use variational approaches or dynamic programming to find optimum attainable reliability under a number of constraints such as first cost, operating cost, weight, volume, and performance parameters. Similarly, we can find the set of possible optimum performance parameters under constraints of reliability, maintenance, schedule, cost, weight, and volume. Cost, weight, volume, as well as performance parameters such as pressure, temperature, and efficiency, will normally be given as a set of limits. In the following brief notes an approach to some of these problems arising in preliminary design is given using the functional equation of dynamic programming as an approach for the solution of the optimum sets of functional relations, which are then used in the iteration procedure arising from the feedback of the reliability, control, performance, and physical constraint loops limiting the optimum design.

Complex System with Component Stand-By

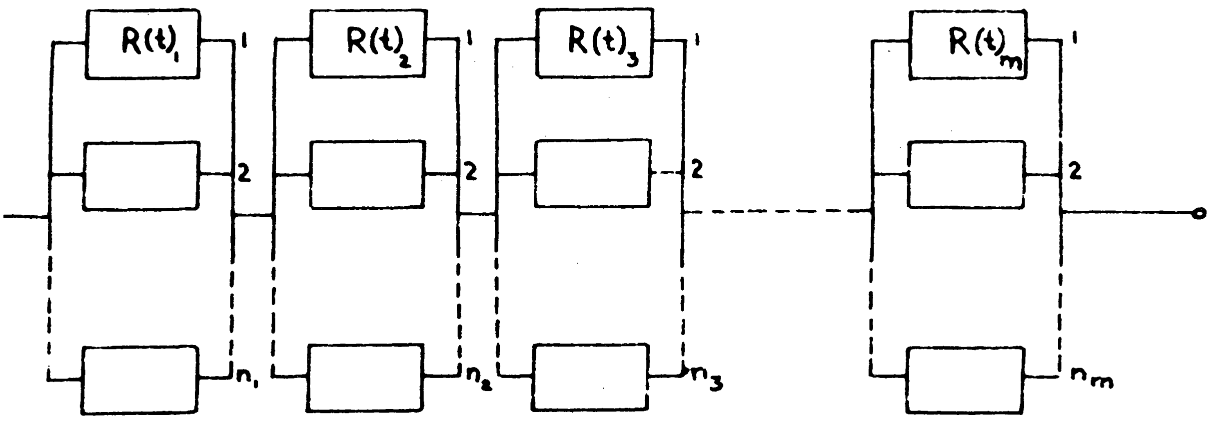
If m in dependent components is arranged in series to form a system, the reliability of the system is expressed by:

$$R(t)_{system} = \prod_{i=1}^{m} R(t)_i = \prod_{i=1}^{m} e^{-c_i t}$$

if constant failure rate $f(t) = C_i$ is assumed. Similarly, if n components are placed in parallel to form a single component system of redundancy degree n, the reliability of the system is equal to:

$$R(t)_{system} = [1 - (1 - R(t))^n] = [1 - (1 - e^{-c_i t})^n]$$

if the system requires only the operation of a single component at a time. Considering a complex system with component stand-by (see Fig. 1), we obtain by combining the two preceding equations:



$$R(t)_{system} = \prod_{i=1}^{m} \left\{1 - (1 - R(t)_i)^{n_i}\right\}$$

$R(t)_i$ = Reliability i^{th} component
m = # of components in series
n_i = Redundancy of i^{th} comp.

1.) Complex System With Component Stand-By.

Appendix IV

$$R(t)_{system} = \prod_{i=1}^{m} \{1 - (1 - R(t_i))^{n_i}\} = \prod_{i-1}^{m} \{1 - (1 - e^{-c_i t})^{n_i}\}$$

If the constraint of our system is defined as $Z = \sum_{i=1}^{m} Z_i$, where Z_i is the allocation of our constraint parameter to component i which has a degree of redundancy n_i, then using the recurrence equation of dynamic programming, optimum allocation policy or allocation for maximum system reliability within the allocation constraint Z, given individual component reliability is fixed, can be expressed in the form:

$$f_m(Z) = \text{Max.}\{[1 - (1 - e^{-c_1 t})^{n_1(z_1)}]\cdot f_{m-1}(Z - Z_1)\}$$

where n_i is a function of the allocation for the i^{th} stage Z_i.

Complex System with Switching

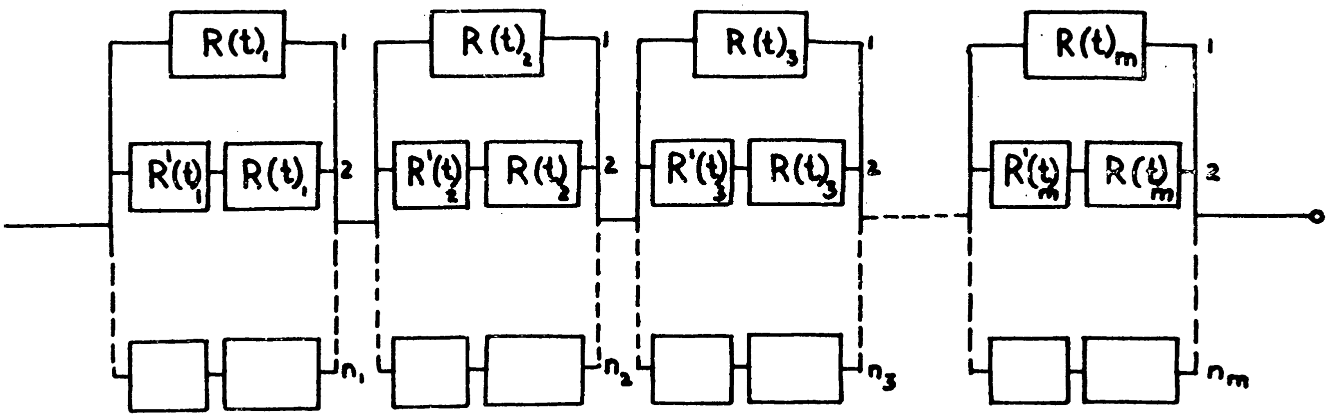
If this system is to be automatically controlled, then a failure detection and switching device must be incorporated in series with all redundant components (Fig. 2). The reliability of these devices will affect the overall system reliability. If the probability that these devices will detect faults in the operating component and switch is equal to their reliability $R^1(t)_i = e^{-c_i^1 t}$, then

$$R(t)_{system} = \prod_{i=1}^{m} \{1 - (1 - e^{-c_i t})(1 - e^{-c_i^1 t}\, e^{-c_i t})^{n_i - 1}\}$$

and the functional equation becomes:

$$f_m(Z) = \text{Max.}\{[1 - (1 - e^{-c_1 t})(1 - e^{-(c_1^1 + c_1)t})^{n_1(z_1)}]\cdot f_{m-1}(Z - Z_1)\}$$

Next, we may generalize the problem and consider a redundant series system made up of nonidentical subsystems, or systems where stand-by units are dissimilar



$$R(t)_{system} = \prod_{i=1}^{m} \left\{1 - (1 - R(t)_i)(1 - R'(t)_i\, R(t)_i)^{n_i - 1}\right\}$$

$R(t)_i$ = Reliability of i^{th} component
$R'(t)_i$ = Reliability of Fault-Detection Device of i^{th} Component
m = Number of Components in Series
n_i = Redundant Degree of i^{th} component

2.) Complex System With Fault Detection & Switching.

Appendix IV

from the main as well as other stand-by units. This method is also applicable for a simple analysis of repairable systems. By this we imply that repair and consequent nonavailability of stand-by units is allowed. Reduced reliability or probability of nonfailure is then a representation of the availability of the redundant part. Since availability can be expressed as a probability density function, which for many maintenance schedules assumes a Poisson distribution as well, it can be introduced as an additional series component in each branch and, in fact, be incorporated into the probability of nonfailure of the switching device. If we, furthermore, allow redundancy of the nonidentical stand-by as well as main components, the general system becomes as shown in Fig. 3 and has a reliability of:

$$R(t)_{system} = \prod_{i=1}^{m} \{1 - (1 - R(t)_{1i})^{a_{1i}} \prod_{k=2}^{n_i} (1 - R^1(t)_{ki} R(t)_{ki})^{a_{ki}}\}$$

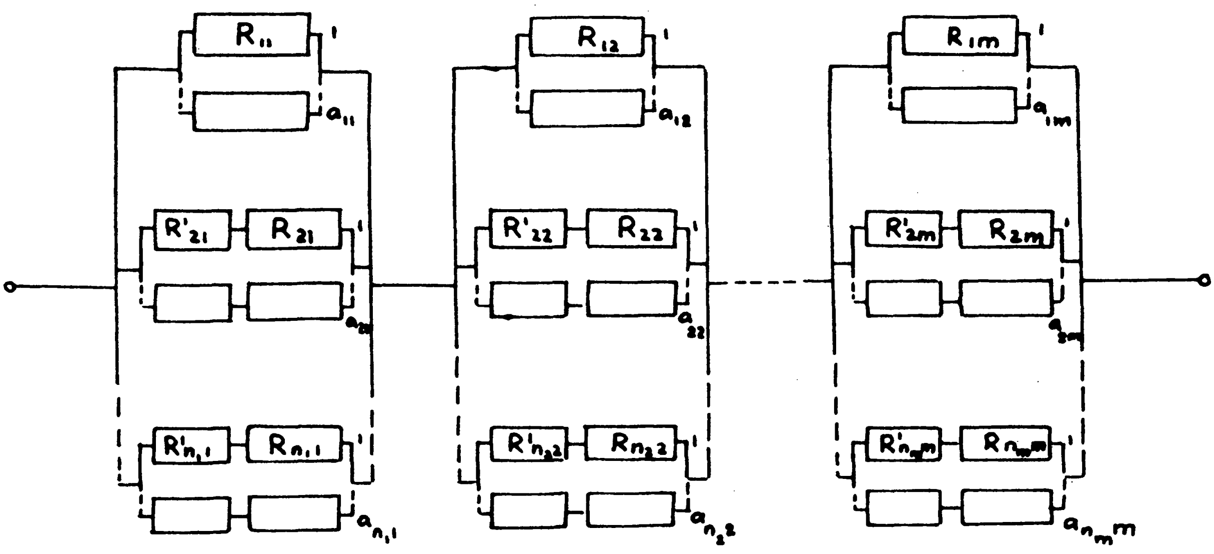
$$= \prod_{i=1}^{m} \{1 - (1 - e^{-c_{1i}t})^{a_{1i}} \prod_{k=2}^{n_i} (1 - e^{-(c^1_{ki}+c_{ki})t})^{a_{ki}}\}$$

and the recurrence equation then becomes:

$$f_m(Z) = \text{Max.}\{[1 - (1 - e^{-c_{11}t})^{a_{11}} \prod_{k=2}^{n_1} (1 - e^{-(c^1_{k1}+c_{k1})t})^{a_{k1}}] \cdot f_{m-1}(Z - Z_1)\}$$

In many systems we find it advantageous to sustain normal operations with multiple parallel components. If K_i is now the number of i^{th} components required for normal operation, then the reliability of a complex system with identical redundant components is:

$$R(t)_{system} = \prod_{i=1}^{m} \{1 - (1 - e^{-c_i t})^{k_i} (1 - e^{-(c^1_i+c_i)t})^{n_i - 2k_i + 1}\}$$



$$R(t)_{System} = \prod_{i=1}^{m} \{1 - (1 - R(t)_{1i})^{a_{1i}} \prod_{k=2}^{n_i} (1 - R'(t)_{ik_i} R(t)_{ik_i})^{a_{ki}}\}$$

$R_{ki} = R(t)_{ki}$ = Reliability of k^{th} Redundant part of i^{th} Series component
a_{ki} = Degree of Redundancy " " " " " " "

3.) Complex Systems With Non-Identical Parallel Subsystems And Subsystem Redundancy.

Appendix IV

which results in the functional equation of the form:

$$f_m(Z) = \text{Max.}\{[1 - (1 - e^{-c_1 t})^{k_1}(1 - e^{-(c_1^1+c_1)t})^{n_1-2k_1+1}] \cdot f_{m-1}(Z - Z_1)\}$$

Reliability of Complex Systems with Component Maintenance

The problem of maintenance scheduling and spare-part inventory control has been the subject of many intense studies. In the following notes, the author only attempts a simple and rather confined analysis of the effect of rigid repair policies on the availability or reliability of a complex system using the dynamic programming approach. The assumption will be made that total repair resources consisting of facilities, parts, and manpower are strictly divided among the different component parts of the system, resulting in both a constant repair rate for each component as well as a maximum number of a particular component that can be repaired simultaneously. Use of resources once allocated will not be allowed for the repair of other components. It is admitted that this approach is nonrealistic; but if allocated resources were to be used for several different components concurrently, the intersection of the statistical distribution of failure events would have to be introduced. In our present study we consider the case of repairing components subject to random failure only in order to obtain maximum system reliability within the constraints of total available repair resources. The case of preventative maintenance which can obviously be staggered to use resources optimally is not considered. This is a problem in which game theory and linear programming will be found useful as well. If we again assume a constant failure rate C_i for component i and a constant repair rate r_i (p_i), we can say that the number of failures of components i occurring in unit time after x_i of components i have failed is $(n_i - x_i)\ C_i = C_{x_i}$. Similarly, r_{x_i}, the repair rate of components i, will be equal to $x_i r_i$ up to the limit of simultaneously repairable components i, which is denoted by $X_i(p_i)$. Let us again assume the K_i parallel components are required for normal operation. Therefore, the largest possible number of concurrent failures is $(n_i - K_i)$. If we now consider the probability of having x_i failures in time $(t + \Delta t)$, then this is equal to the sum of the probability of having:

a) x_i failures in time t and no failures or repairs in the time between t and $(t + \Delta t)$,

b) $(x_i - 1)$ failures in time t and one failure in the time between t and $(t + \Delta t)$,

c) $(x_i + 1)$ failures in time t and one repair in the time between t and $(t + \Delta t)$,

or

$$\overline{R}_{x_i}(t + \Delta t) = \overline{R}_{x_i}(t)[1 - C_{x_i}\Delta t][1 - r_{x_i}\Delta t] + \overline{R}_{x_i-1}(t)[C_{x_i-1}\Delta t]$$
$$+ \overline{R}_{x_i+1}(t)[r_{x_i+1}\Delta t]$$

This difference equation can be solved by iteration procedures, or by converting it into a differential equation and, thereafter, transforming it using Laplace Transformers and solving the set of resulting linear equations by the use of the Gauss-Jordan Method. We then obtain values of $R_{x_i}(t)$ for all possible values of x_i in terms of r_{x_i}.

The reliability of the system of m series subsystems of components each with a redundancy i (Fig. 1) is then:

$$R(t)_{system} = \prod_{i=1}^{m}\left(\sum_{x_i=o}^{n_i-k_i}\overline{R}_{x_i}(t)\right) = \prod_{i=1}^{m}[1 - \overline{R}_{n_i-k_i+1}(t)]$$

where $$\sum_{x_i=o}^{n_i-k_i+1} \overline{R}_{x_i}(t) = 1$$

and $$\overline{R}_{n_i-k_i+1}(t) = (1 - R(t))^{n_i-k_i+1}$$

also $$\overline{R}_o(0) = 1 \quad \text{and} \quad \overline{R}_{x_i > o}(0) = 0$$

If we are interested in optimizing system reliability within the total repair resources available ($P = \Sigma\, p_i$) and, as $\overline{R}_{x_i}(t, p_i)$ = probability of x_i failures in time t is a function of p_i, we, therefore, can write:

$$f_m(P) = f_m\, R\,(t, P)_{\text{system}} = \text{Max.}\left\{\left[\sum_{x_1=o}^{n_1-k_1} \overline{R}_{x_1}(t, p_1)\right] \cdot f_{m-1}(t, P - p_1)\right\}$$

Obviously, this solution is only valid if p_i is specialized for use on component $_i$. If we fix the repair rate r_{x_i} and the total number of required operating components K_i, we again can obtain optimum system reliability for a total system allocation of Z where $n_i(Z_i)$, or the degree of redundancy, or the quality of the individual components are functions of the allocation for the stage Z_i . Stated in functional equation form, we obtain:

$$f_m(Z) = \text{Max.}\left\{\left[\sum_{x_1=o}^{n_1-k_1} \overline{R}_{x_1}(t, Z_1)\right] \cdot f_{m-1}(Z - Z_1)\right\}$$

where $Z = \sum_{i=1}^{m} Z_i$ and n_i a function of Z_i.

Analysis of Component Failure

Although we only considered catastrophic component failure in the preceding work, it is seldom that a component will fail only as a result of a single failure, even t. In fact, most components will have a multitude of possible catastrophic failure events, each with its own failure rate. The component failure rate is, consequently, the sum of the failure rates of all the possible catastrophic events. If $F(t)_{ij}$ is the probability of failure of the jth event of the ith component in our system, then assuming constant failure rates of events:

$$F(t)_{ij} = (1 - e^{-c_{ij}t}) = \overline{R}(t)_{ij}$$

where C_{ij} = failure rate of event j of ith component, normally $0 \leq F(t)_{ij} \leq 0.1$, and $\overline{R}(t)_i = F(t)_i$ = probability of failure of ith component,

$$F(t)_i = \sum_{j=i}^{s} (1 - e^{-c_{ij}t}) \cdot = \overline{R}(t)_i$$

Breipohl of Sandia Corporation (Ref. 3) has found that component cost and probability of failure can be related by an expression of the form:

$$Z_i = \frac{A_i \exp - B_i\,(1 - e^{-c_i t})}{(1 - e^{-c_i t})}$$

where K_{1i} and K_{2i} are constants. Similarly, assuming that the probability of failure of the component is the sum of the probabilities of failure of the possible failure events,

$$Z_i = \sum_{j=1}^{s_i} \frac{A_{ij} \exp - B_{ij} (1 - e^{-c_{ij}t})}{(1 - e^{-c_{ij}t})} = \sum_{j=1}^{s_i} Z_{ij}$$

and the cost of the system of m series components, each with redundancy n_i,

$$Z_{\text{system}} = \sum_{i=1}^{m} n_i Z_i = \sum_{i=1}^{m} \sum_{j=1}^{s_i} n_i Z_{ii}$$

$$= \sum_{i=1}^{m} \sum_{j=1}^{s_i} \left[n_i \frac{A_{ij} \exp - B_{ij} (1 - e^{-c_{ij}t})}{(1 - e^{-c_{ij}t})} \right]$$

As formed previously,

$$R(t)_{\text{system}} = \prod_{i=1}^{m} R(t)_i = \prod_{i=1}^{m} \left\{ 1 - \left[\sum_{j=1}^{s_i} (1 - e^{-c_{ij}t})^{n_i} \right] \right\}$$

for series events constituting failure

$$R(t)_{\text{system}} = \prod_{i=1}^{m} R(t)_i = \prod_{i=1}^{m} \left\{ 1 - \left[\prod_{j=1}^{s_i} (1 - e^{-c_{ij}t})^{n_i} \right] \right\}$$

for parallel events constituting failure.

If we are now interested in finding the minimum cost for a system designed for a set value of reliability, dynamic programming will give us an optimum allocation of component and event reliability for minimum system expenditure:

$$f_m(R(t)) = \text{Min.} \left\{ \left[n_i \sum_{j=1}^{s_i} \frac{A_{ij} \exp - B_{ij} (1 - e^{-c_{ij}t})}{(1 - e^{-c_{ij}t})} \right] + f_{m-1} \, {}^{R(t)}\!/\!{}_{R(t)_1} \right\}$$

where $R(t)$ and $R(t)_i$ are obtained from the equations previously developed.

The method can again be generalized for the case of complex systems with nonidentical parallel subsystems and including switching devices. We then obtain Total Cost of System:

$$Z_{\text{system}} = \sum_{i=1}^{m} \sum_{j=1}^{s_i} \left[n_i \frac{A_{ij} \exp - B_{ij} (1 - e^{-c_{ij}t})}{(1 - e^{-c_{ij}t})} \right]$$

$$+ \sum_{i=1}^{m} \sum_{j=1}^{s_i'} \left[(n_i - 1) \frac{A^1_{ij} \exp - B_{ij} (1 - C^1_{ij} t)}{(1 - e^{-c_{ij}t})} \right]$$

where s_i' = the number of failure events in switching device i and system reliability.

$$R(t)_{\text{system}} = \prod_{i=1}^{m} \left\{ 1 - \left[\sum_{j=i}^{s_i} (1 - e^{-c_{1ij}t}) \right] \prod_{k=2}^{n_i} \left[\sum_{j=1}^{s_i^1} (1 - e^{-c^1_{kij}t}) + \sum_{j=i}^{s_i} (1 - e^{-c_{kij}t}) \right] \right\}$$

when component failure events in series and

$$R(t)_{\text{system}} = \prod_{i=1}^{m} \left\{ 1 - \left[\prod_{j=1}^{s_i} (1 - e^{-c_{iij}t}) \right] \prod_{k=2}^{n_i} \left[\prod_{j=1}^{s_i^1} (1 - e^{-c^1_{kij}t}) + \prod_{j=1}^{s_i} (1 - e^{-c_{kij}t}) \right] \right\}$$

when component failure events parallel.

Similarly, again the recurrence equation allocating component and event reliability for minimum system cost,

$$f_m(R(t)) = \text{Min.} \left[\left\{ \left[n_1 \sum_{j=1}^{s_{11}} \frac{A_{1j} \exp - B_{1j} (1 - e^{-c_{11j}t})}{(1 - e^{-c_{11j}t})} \right] \right. \right.$$

$$\left. \left. + \left[(n_1 - 1) \frac{A^1_{1j} \exp - B^1_{1j} (1 - e^{-c^1_{11j}t})}{(1 - e^{-c^1_{11j}t})} \right] \right\} + f_{m-1} \, R(t) \big/ R(t)_1 \right]$$

Conclusions

An approach to the solution of complex system reliability problems under constraints has been presented using the functional equation of dynamic programming for the optimal set of multivariable analysis. The equations so obtained are in a form easily programmed for digital computer application. A next step, directed towards a more realistic model, should introduce distribution laws for the failure events and consider out-of-tolerance failure distributions. Design parameters in the form of tolerances must be analyzed as functions of cost and probability of failure for particular components. Finding the intersections or overlap of the failure density distributions of failure events and component failure should also permit a more realistic approach to maintenance scheduling.

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SYMBOLS

$R(t)$	= Reliability or probability of nonfailure in time t.
$f(t)$	= Failure rate.
C	= Failure rate or failures per unit time (constant).
i	= 1, 2, ... m Number of components in series.
n_i	= Number of redundant i^{th} components or degree of redundancy of i^{th} component.
$f_m(Z_i)$	= Maximum of m stage multivariable process with total allocation Z.
a_i	= Redundancy of redundant component K_i.
Z	= Total system allocation.
Z_i	= Allocation for i^{th} component.
p_i	= State of i^{th} component or repair resources allocated for i^{th} component.
q_i	= Decision made in i^{th} state.
x_i	= Number of failures of i^{th} component.
K_i	= Number of i^{th} components required for normal system operation.

r_i = Repair rate for i^{th} component.

$\overline{R}(t)$ = Probability of failure in time t.

P = Allocation of repair resources for total system.

$X_i(p_i)$ = Maximum number of component i simultaneously repairable.

t = Time.

A = Constant.

B = Constant.

$F(t)_{ij}$ = Probability of failure of j^{th} event of i^{th} component.

C_{ij} = Failure rate of j^{th} event of i^{th} component.

s_i = Number of failure events of i^{th} component.

s'_i = Number of failure events of i^{th} switching device.

C'_i = Failure rate of i^{th} switching device (constant).

C_i = Failure rate of i^{th} component (constant).

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DISCUSSION

KARASSIK (Worthington): I want to take exception to some of the statements you have made with reference to once-through boilers. You mentioned that a majority of today's land based steam electric power plants use once-through boilers. As of right now I would say less than 1 or 2% of the installations are once-through boilers, and less than 5% of those on order are of that type.

Second, you stated in the paper that simplicity of controls and quicker responses is a big advantage. This probably will ultimately be the case. At the present time, the simplicity of controls is not the advantage. The advantage is cost. The controls themselves have been the item that has been giving the greatest amount of headaches on land based steam electric power plants with once-through boilers.

Third, you make reference to the fact that the once-through boiler is particularly applicable, suitable for marine use, because of low load operation. This may be true again of this particular or special once-through boiler that you are studying. Those installed are actually suffering from very poor low load economy. A once-through boiler today requires 25 to 33% minimum flow, and as a result when the load is less than that, it requires dumping steam back through a deaerator and condenser. The ratio of loads in a utility compared to that in a marine application is much less. That is, the low load of 33% can be tolerated as a minimum in a land based station. I imagine it could not be tolerated in marine application.

FRANKEL: I'm in no position to argue with you about utility station statistics. I've only noticed that of some of the largest plants, the largest utility stations constructed in the last 10 years, a vast majority have been once-through boilers (plants of half a million or more kilowatt).

With regard to the no-load requirement, I agree that the no-load requirement of the once-through boiler is higher than the water-tube boiler, for instance.

I don't want to argue the point of 33% or 25%. In my paper I also advocated a base load generator in order to reduce or eliminate dumping altogether. You require aboard ship a large amount of electric power. This base load generator would normally take 10 to 15% of full load power. According to some of the information I have, no load of 10 to 15% would result in an ample through-flow in the design of the nature I have proposed. Obviously, the no-load through-flow requirements depend very much on the form of the design, and will vary greatly among the utility stations and the comparatively small marine power plants. Did I answer your question, sir?

KARASSIK: As regards controls?

FRANKEL: Oh! With regard to controls, I agree with you that some of the utility stations still have large amounts of trouble during large fluctuations of loads. Those who use these boiler plants as base plants maintain that they have little or no control problems. I agree with you that it is rather appropriate possibly to suggest such a plan where control functions and the controllability of the plant by itself is not yet fully known for a marine power plant, but one of the great obstacles has been the dynamics, or the ignorance of the dynamics of two-phase flow.

A great deal of research has gone into this field in the last 3 or 4 years, and its results, I can say, are very encouraging. We have great confidence presently in the transfer functions obtained for them which, I believe, will result in great confidence and reliance on the controllability of such boiler plants.

The following is a written discussion of the author's paper by Mr. Karassik.

The aims expressed by Prof. Frankel are without question valid and praiseworthy. Savings in cost, weight and space, coupled with increased simplicity of control and improved reliability will always be popular with shipowners, naval architects and operators alike. I am afriad, however, that I must find fault with some of the steps Prof. Frankel proposes to take to achieve these aims. Specifically I must take exception to several statements he made in his paper and in his verbal presentation of this paper:

1. The majority of utility steam-electric generating units *do not* at this moment use once-through boilers, but rather conventional drum-type boilers. Probably less than 1 or 2% of the units in service and less than 5% of the units on order today incorporate once-through boilers.
2. Simplicity of control *is not* the reason for the use of once-through boilers, but first cost is. In addition, for supercritical pressure installations of 3500 or

4500 psi throttle pressure, the once-through boiler is, of course, the only solution. But the control of these boilers still presents certain problems today and therefore this simplicity to which Prof. Frankel refers just *does not exist.*

3. Once-through boilers *are not* more economical than drum-type boilers at low load operation. As a matter of fact, the need to maintain minimum feedwater flows through the once-through boiler of up to 33% of rated flow requires dumping of steam back to the condenser whenever the unit load requires less than 33% steam flow. This characteristic of the once-through boiler is certainly a deterrent to economy at low loads. When applied to marine use, this objection becomes even more important because required steam flows range over wider limits than this 3 to 1 ratio.

The once-through boiler is acceptable and even desirable under certain circumstances for land-based steam power plants because of compensating advantages. I doubt, however, that it is going to be quite as acceptable aboard ships, especially aboard Navy ships until some of its present shortcomings are eliminated.

It is hoped that these characteristics can be eliminated in the future, at which time the once-through boiler will become more readily integrated into steam power plants, both on land and at sea.

KECECIOGLU (U. of Arizona): I do want you to explain to us what you mean by interaction. I don't know if we all understand your definition.

FRANKEL: Interaction is the result of response of one component, or the output of one component on the input and resulting response on the other. Now, interaction can be defined from the reliability analysis point of view in which the failure rate and possibly performance criteria of the different components are affected by the action of other components. From a control point of view, this obviously normally results in dynamic interaction. This means dynamic response of one component as a result of dynamic response or a variation dynamic response of another component in line.

KECECIOGLU: Can this be taken care of by proper component stress analysis, as well as, by considering the application and the operation environment of the component? Is this what you are referring to?

FRANKEL: Yes, sir, it has to be. The one point is that it has to be included at the design stage and not as an afterthought. What I would like to mention .....

KECECIOGLU: This is the first time I've heard of this combination.

FRANKEL: All I would like to say is I believe we should get away from trying to design components just on the basis of state points. That means we have a certain state point and certain outputs of a component, and we design a component accordingly. We should definitely try to find out at an initial stage what are the possible additional inputs by other components in the series or even parallel system, and how this will affect the performance from the reliability point of view as well as the dynamic control point of view of this particular component.

KECECIOGLU: I couldn't quite get how your switching reliability or standby reliability was incorporated in your reliability equations. You're taking the switching reliability as 100%.

FRANKEL: I simply assumed this simple analysis, and, as I mentioned, I used constant failure rates. If we take a simple case of identical components in a number of redundant identical components, I assumed that reliability of the nonworking components is actually in series with the reliability of the switching device and fault detecting device which will start this component should the working component fail; and, therefore, the reliability of the stand-by component has to be reduced by multiplying it by the reliability of the switching and fault-detecting device. You see, the fault detecting device will only have the reliability or probability of detecting the fault and actually starting the switching of 90%.

In this case I obviously have to put in the reliability of the stand-by component as its own reliability multiplied by 90%. This means its own reliability is reduced.

HIRSCHKOWITZ (U.S. Merchant Marine Academy): If I'm permitted to ignore the benefits of the compactness and perhaps the flexibility design in the weight savings of the once-through boiler, you talk about response time and possibly taking 15 minutes to level off things of this order. Going back many years by having a high inertia boiler, you have inherent reliability in the system that you don't have to design for and I'm always at a loss as to why people, with respect to ships, insist that these pressures have to remain so level. I've maneuvered several ships in my time through various harbors and really the pilots have no concern to the boilers anyway. They ring the bells, sometimes 56. Another captain can take it into the dock with 3, so these things seem sort of unimportant to me unless we drop the pilot; whether it took 15 minutes or 20 minutes or an hour to level off, it didn't really matter. I have somehow a feeling that the sophistication that would be required in the controls of a once-through boiler — I'm talking ignorance now because I haven't seen these controls, but I can only imagine — would somehow destroy this approach to high reliability.

Secondly, on your loops that you talk about like feedwater controls and things of that nature. Unless we're talking about a completely automated ship where there is not time to respond, where a person watching the system is not included in the concept, feedwater regulator with a boiler of reasonable size doesn't have to get involved in here if it's slow, it could be a rather simple thing even if the water level drops a little bit, as long as it is not going out of sight, as long as it doesn't carry over. If the inertia of the system is big enough you have these things.

Unless we insist on this reduction in size and a reduction in weight, these problems are being manufactured by us. If we could only go backwards and get a lot of reliability.

FRANKEL: I agree with you that we aren't really worried about fluctuations in pressure; but I am worried with regard to the resulting and consequent fluctuations in superheat temperature and, besides this, talking of the water level dropping a little bit more or less, during 14 years on steamships, I remember that carryover was always the result of a fast maneuver or emergency maneuver when the water level did drop out of sight.

Now, I agree with many of the statements that have been made before that we really know very little about the controllability of such a boiler. All I can say at this stage is that from a theoretical point of view it looks as if the control functions can be reduced to an absolute minimum, possibly a single loop and possibly just a loop maintaining constant fuel to feed ratio and fuel to air ratio, which may be all that is required.

We can't do it now, but I hope that if a large amount of research goes into this for a few years we may accomplish this; and I believe particularly the fact that we are more or less married to steam power plants and that the competition from large output, slow-running diesel engines from Europe is increasing in all power ranges and may result in the required additional research. In a way we had better get moving trying to get a low weight, easy to control, possibly fully automated power plant. Fully automated power plants, whether we like it or not, are going to come, because obviously we'll have to follow our competitors; and the trend has been in most of the foreign countries towards automation in merchant ships; and I believe, particularly with our high labor costs in this country, we will have to follow suit. I personally believe we had better look at a different type of boiler plant to do it.

HAUSCHILDT (Bureau of Ships): Your paper has been on merchant ship application. How do you feel about this for naval ship application, since we have a possible or even greater degree of maneuverability requirement and also possibly a less part load condition?

FRANKEL: As I mentioned before, theoretically it looks as if this plant is much more easily controllable and maneuverable than a plant with a high inertia boiler. Added advantages are obviously low weight, low volume, and the engine room could obviously be placed in a low head space as the boiler could practically be put into the funnel of whatever shape.

Other advantages are that this type of boiler plant could be shut down in practically zero time with no refactory maintenance, etc. It has the additional advantage that its temperature, its outlet temperature, can actually be adjusted, for instance, which would also definitely reduce and facilitate, for instance, maintenance of steam turbines, start-up procedures, etc. Therefore, I believe that if we do solve the practical control problems and get some reliable data on the response of the two-phase phenomena on which as I mentioned, we actually know very little, and knew practically nothing a few years ago, I'm very hopeful that this boiler plant is going to stay and possibly be the one that will propel naval vessels of the future.

MC INTOSH (USCG): If this low flow is a real bottleneck, why can't that be solved by going to more than two boilers so that one or more may be cut out when demand is reduced?

FRANKEL: The low flow problem would not be solved, because you would have two boilers each requiring anything between 10 and 33% of zero flow....

MC INTOSH: What if you operate just one boiler?

FRANKEL: If you just operate one — yes, but there again, it is a control problem of two boilers. The great advantage of a single boiler is really the large reduction of the control elements. The minute the boiler plant works by itself it requires all of its auxiliaries, and do not forget that the feed pump which has to overcome a pressure drop in a once-through boiler of 300 to 500 pounds to overcome the pressure drop is a monster of an auxiliary requiring about 300 to 400 horsepower. A plant of this nature would actually require no load just for its auxiliaries after start-up of close to 20% of its full-power rating for a plant of 22,000 shaft horsepower. Therefore, the trouble is only in port requirement, because the start-up is practically instantaneous and takes minutes in this particular plant. I don't think that this is a really major problem.

HIRSCHKOWITZ: Would you care to comment on the type of burners on light off techniques that you might envision for such an arrangement?

FRANKEL: In our little study here, we envisioned a full turn-down ratio burner, in particular, with regard to full automation of this particular plant. We considered the spill-type pressurized burner which is used by Walton in England and many places around here. Compared to the steam-assisted burner, we definitely find the latter has many advantages: efficiency, for instance. There is a large energy loss in the return oil with this spill-type burner. Beside this, we found that the controllability of the steam-assisted burner is superior; and we would definitely recommend a steam-assisted burner with a turn-down ratio of about 20 which can be achieved quite easily.

COHEN (Naval Boiler and Turbine Lab): Where you mentioned the inertia of the boiler being a problem, don't you really mean the inertia of the forced draft blowers? Really this is much more of a problem than the inertia in the boiler. Secondly, you also in the paper mentioned that improved atomization should result in being able to burn Bunker C in the supercharged boiler. Aren't there other problems, just to mention a couple, such as vanadium pentoxide in the fuel gas or soot in the gas turbine?

Also in the turbine following concept, I might mention that it hasn't been universally rejected as you mentioned in the paper. We at the laboratory recently completed an analogue computer study of such a plant which successfully met all the objectives.

Also lastly, you mentioned the difficulty of matching frequency response data or to calculate a transfer function, or vice-versa, really? We have successfully done this for many power plant components.

FRANKEL: Starting with the last question, we have this difficulty obviously in the dynamics of two-phase flow, particularly where many other details do come in as you are probably aware. I'm glad that the Naval Turbine Laboratory has had success with this supercharged boiler.

I have some experience with Velox boilers and was rather surprised that the Naval Turbine Lab chose a supercharging pressure of 67 pounds compared to the usual Velox pressure of 37 or 40 pounds. At the time I participated in combustion problems and combustion experiments on Velox boilers, we had quite a lot of combustion problems (apart from the technical problems in the gas turbines) within the boiler and in actual combustion within the combustion chamber at these high, supercharging pressures. Does this answer your questions?

COHEN: Just one other thing, in talking about supercharging you mentioned an increase of 13% in efficiency. Did you mean this just as a result of the supercharging itself or the combination of the monotube and supercharging?

FRANKEL: A combination obviously. Ordinary boilers have 88% efficiency. How can you increase it by 13%? You can't have more than 100 yet. So this obviously is a total plant on the same efficiency basis including some gas turbine.

WHAT PRICE RELIABILITY?
—An Inquiry into Land-Based Steam Power Plant Practices

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INTRODUCTION

By all standards, including that of the reliability of the propulsion machinery of its ships, our Navy is reputed to be the best in the world. And, if this is true, it certainly did not happen by accident. No one has ever pulled a rabbit out of a hat without carefully putting it there in the first place. But this superiority may well be on a relative and not on an absolute basis. How reliable *is* this propulsion machinery? Apparently not to a degree satisfactory to the Navy personnel, since it has thought it sufficiently important a subject to convene this meeting.

The propulsion machinery of a ship is essentially a complete steam power plant combined with a propeller or propellers, driven through a set of reduction gears and it is to the reliability of the steam power plant itself that we shall address ourselves now. It is the purpose of this paper to examine the practices of land-based central steam stations and to determine how the knowledge of these practices can help us in improving the reliability of naval vessel propulsion machinery.

It would be most fruitful if we could examine these practices with respect to every single component of a land-based steam power plant, that is with respect to the boiler, the turbine and the many pieces of equipment which comprise the Fluid Handling Group of the plant. We would consider practices related to the design, installation and operation of the condenser, the circulating pumps, condensate pumps, vacuum-producing equipment, deaerator, boiler feed pumps and closed feedwater heaters, to say nothing of the gate valves, check valves, regulators and other related equipment.

Such an undertaking, I am afraid, would be beyond the scope allotted to a paper such as this, to say nothing of the fact that it would be beyond my own competence and capabilities. I have chosen instead to limit my inquiry to the practices regarding reliability with respect to a single piece of equipment: the boiler feed pump. I submit, however, that we can derive considerable help in examining this one piece of equipment. Reliability, to an engineer, is a way of life. You cannot divide reliability into compartments. You cannot have reliability of the steam turbine if you do not care sufficiently to design reliability into the boiler feed pumps, the deaerator or the condensate pumps.

At this point I would like to present a brief statement of the conclusions I reached following my inquiry into land-based steam power plant practices with regards to the reliability of boiler feed pumps:

1. Reliability is very definitely related to the price of the equipment and of such protective or supervisory controls that may be installed. Generally, savings in the price of this equipment can only be achieved at the expense of the reliability of the boiler feed pumps.
2. Improvements in pump efficiency and a gain in reliability are fundamentally incompatible.

3. There is no clear-cut trend that might represent the general practice of all U. S. utilities. Because reliability is to some degree an intangible characteristic, it has been and still is a very controversial subject. Opinions range from a total disregard of such an intangible to the use of "value analysis" in which price — and efficiency — play a very secondary role.

SIMILARITIES AND DIFFERENCES

But before we can gain any insight from an investigation of land-based practices, we must set down what are the similarities and what are the differences between a central steam power station and a naval vessel and between the organizations that make policy decisions in the two cases. Oddly enough, there are more differences than similarities.

On the side of the similarities, we can start with the obvious fact that we are looking at a steam power plant in both cases. We can also state that in each case the power plant must be capable of operating over a wide range of loads and — if I stretch the point a little — that rapid fluctuations in load might be expected on land as well as aboard ship. Finally, the fundamental laws that govern the hydraulic and mechanical design of the equipment that comprises the steam power plant apply on land and at sea.

When we come to the differences, I am not certain whether I can reasonably be expected to prepare a complete catalog. But some of the most important differences are the following:

1. Navy vessel steam power plants generally operate at lower steam pressures and temperatures than large modern central stations.
2. Weight and space are relatively unimportant factors in the case of land-based stations, but of utmost importance aboard ship.
3. No one has yet insisted that a central steam station be made capable of withstanding a torpedo amidship and continue to send out kilowatts, albeit at a reduced rate.
4. Operators of land-based steam power plants resent unscheduled outages and especially those which shut a unit down. But these plants are generally part of an interconnected grid system with available reserve capacity. The shut-down of a unit is not a matter of life-and-death, quite literally, as for a ship or for a fleet.
5. A land-based plant is expected to be on the line for relatively long periods of time, possibly a year, without requiring maintenance or overhaul. A ship will generally be available for overhaul more frequently. But the time for a scheduled overhaul is quite unpredictable and, in case of war, any outage must be treated as unscheduled outage.
6. Certainly facilities and personnel for even routine maintenance are more amply available in the case of land-based plants.
7. "Maintenability," that is, ease of maintenance from the point of view of the space around the equipment is certainly more of a factor aboard ship than on land.
8. The Navy and the investor-owned utilities have vastly different purchasing policies. There is no such thing as "type approval" for land-based steam power plants. On the other hand, a privately owned utility reserves for itself the right to evaluate intangibles in choosing between two or more suppliers of equipment "apparently" designed to perform the same service equally well. It does not have to buy on a strictly numerically evaluated basis if it does not wish to do so.

I could continue to list more differences. But I think that I have proved my point, which is that major differences exist and that they should be taken into consideration when trying to apply any lessons we may learn from our examination of land-based steam power plant practice.

WHAT IS RELIABILITY?

There are probably as many definitions of reliability as there are engineers ready to define it. But the one definition that I have always preferred — and that I have frequently quoted — says that the reliability of a particular component or system of components is the probability that it will do what it is intended to do under certain operating conditions and for a specified period of time.

When it comes to applying this definition to the reliability of a boiler feed pump, we find that we can go so far and no further. We cannot assign a significant number to the probability that the pump will do what it is intended to do, that is, feed the boiler under *all* the possible circumstances that can — I did not say "should" — befall the installation.

Why is this? Why can we not say that statistically the reliability of such-and-such a pump is 99.3%, or some such number? For the simple reason that utilities have not assembled any statistics that would let us do this.

The amount of statistics compiled by the utility industry has by now reached very respectable proportions. Some of this is self-selected, that is, compiled by the choice either of the utilities themselves or of utility associations. A very great portion of it is imposed on the utilities by federal or state governing bodies. But, unfortunately for our purpose, these statistics do not include sufficient information to provide us with factual data on the availability of equipment other than the two major components of a steam power plant, that is, the boiler and the turbine — generator. The availability record of a vital component such as the boiler feed pump cannot be determined with any degree of accuracy whatsoever. We are forced, therefore, to speak of relative degrees of reliability, of relative probabilities of unscheduled outage and of the relative advantages of alternate design or application solutions.

But once we are willing to accept qualitative interpretations of the reliability of boiler feed pumps, we can use this qualitative concept in describing the general experience that has been observed in utility steam power plants. In one of my previous papers[1] I had presented a curve which describes qualitatively the possibility of failure of a piece of equipment such as a boiler feed pump at the end of a given number of operating hours. This curve (Fig. 1) serves mainly the purpose of focusing our attention on the classification of all failures into three separate groups:

(a) Infant mortality
(b) Random failure
and (c) Wear-out

Experience in land-based steam power plants has shown that it is the "infant mortality" which is the greatest enemy of boiler feed pump reliability. In addition, an analysis of the failures that beset a boiler feed pump in its early life will reveal that the overwhelming majority of these failures can be traced to sources external to the pump itself, such as:

1) Dirt in the piping (feedwater, condensate injection or lubrication).
2) Improper warm-up procedures

[1] "Criteria of Equipment Reliability Evaluation" by I. J. Karassik, presented at the June 14-18, 1959, Semi-Annual Meeting of the ASME, St. Louis, Missouri. (Worthington Reprint RP-1115)

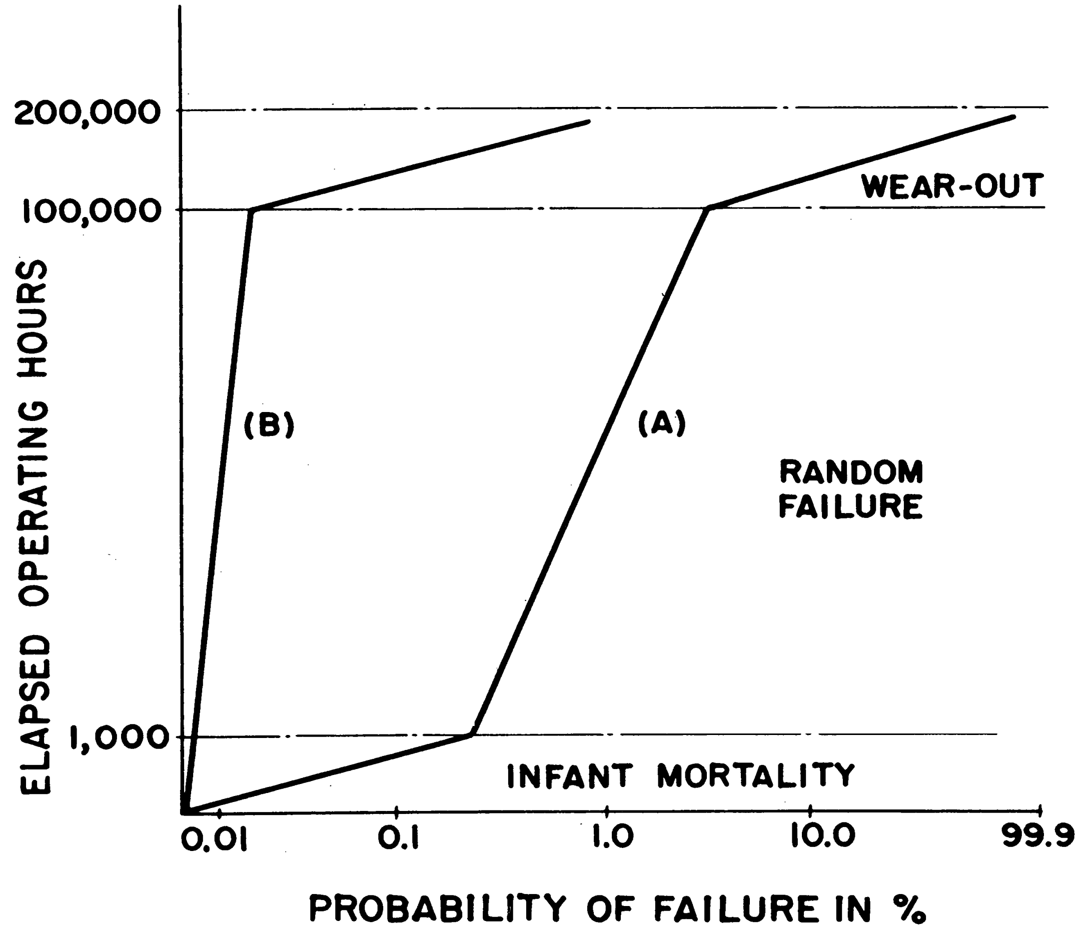

Fig. 1. Failure curves can be developed, once frequency of failures has been established.

3) Check valve failures.
4) Failures of the auxiliary power system.
5) Sudden and unusual load changes.
6) Unit trip-outs.

Yet, the effort to eliminate or to minimize circumstances contributory to this group of failures is not always apparent.

You will note that under this list of six contributory circumstances only the first two would normally be expected to be eliminated with time and therefore not to contribute to random failures in the later life of the boiler feed pumps. Check valve or auxiliary power failures, sudden load changes and unit trip-outs may all be slightly more frequent in the early stages of operation, but they remain probabilities throughout the life of the steam station. Why then did I list them under "infant mortality" causes?

The fact is that in too many instances the early warnings of the possibility of these contributory causes to boiler feed pump failure are disregarded. Steps to remedy the situation are only taken after one or more pump failures have convinced someone that the warning was justified. I am not speaking of any one particular utility, nor necessarily of the majority of utilities, but the fact that this happens frequently is to be deplored.

Maybe these early warnings are not sufficiently loud and clear? This could well be, but I must explain here a psychological barrier that is sometimes erected in front of the manufacturer who feels it is his duty to bring the potential dangers to the attention of the buyer. Only too frequently has such a manufacturer found that he was the only one to talk of these dangers and to act the "Jeremiah."

And when this happens, he may also find that he has planted seeds of suspicion in the buyer's mind that this manufacturer's equipment is probably less sturdy than that of his competitors, less reliable since he is the only one to worry about such routine matters as unit trip-outs or sudden load changes.

Once, however, the equipment fails because of these external circumstances, corrective measures may be quite costly, beyond even the damage done to the equipment itself. You will find, for instance, cases where repeated failures of check valves during the early life of the plant are corrected by the installation of a second set of check valves in series with the first. If the belt breaks, the suspenders will hold the pants up.

Where auxiliary power failures cause excessive unscheduled pump outage because of consequential damage, expensive electrical interlocks and second-line defenses may be set up. If sudden load changes are too unhealthy for the boiler feed pumps that have been selected and installed, controls are superimposed to limit the permissible rate of load change in the future. And yet, it is quite possible that some other boiler feed pumps could have survived the sudden load changes much better.

RELIABILITY vs. AVAILABILITY

We have been speaking of reliability in rather general terms. We have defined reliability as best we could and we have noted that failures which reduce reliability can be classified into "infant mortality", random failures and wear-out. And we have also noted that a large proportion of infant mortality failures of boiler feed pumps can be traced to sources external to the pumps themselves. But while the source may be external, we must accept the fact that the relative vulnerability of the boiler feed pumps must be a factor in whether a particular circumstance will or will not cause a failure.

Our major concern, however, is with availability. This can be defined as the percentage of time that the equipment is in "operational readiness." Thus, availability incorporates the effect of both forced and scheduled outage.

By virtue of this definition, availability is not necessarily synonymous with reliability. For one thing, the concept of reliability does not incorporate within itself the factor of down-time. If two pieces of equipment can each be expected to suffer one failure in 8000 hours of operation, they could be considered as having the same degree of reliability. But if one requires 80 hours of down-time for maintenance after the failure and the second requires only 40 hours, these two machines will have an availability of 99% and 99.5% respectively.

We can refine this concept still further. Let us consider 3 boiler feed pumps on which we have the following data:

Pump	A	B	C
One failure per	16,000 hrs.	8,000 hrs.	40,000 hrs.
Down-time after failure	48 hrs.	40 hrs.	80 hrs.
Normal maintenance outage each	40,000 hrs.	40,000 hrs.	80,000 hrs.
Maintenance down-time	48 hrs.	40 hrs.	80 hrs.
Total down-time per 80,000 hrs.	336 hrs.	480 hrs.	240 hrs.
Down-time in %	0.42	0.60	0.30
Availability in %	99.58	99.4	99.7

There is an interesting lesson to be learned here. While frequency of failure and of scheduled outages for maintenance as well as the down-time required for repairs or for preventive maintenance all enter into the calculation of availability, they cannot be given equal weight in our considerations of the "intangible" reliability of the three pumps.

The exact importance to be attached to each of these components of "unavailability" will vary considerably between various utilities and between various installations, because each installation has a different situation with regards to the permissible unavailability of a boiler feed pump. Different utilities may have different margins of reserve of installed capability. In addition, the seriousness of an interruption will be quite different in the case of an installation of three half-capacity pumps (one of which is a spare) than if only two such pumps are installed. As a matter of fact, we shall presently see how this consideration can be used in evaluating the advantages of spare equipment.

Generally, however, down-time for preventive maintenance is of considerably less importance than down-time following an unscheduled outage by virtue of the fact that all utilities have reserve capability and can schedule preventive maintenance to suit their particular load pattern. This is also true of Navy propulsion machinery to a certain degree, although for somewhat different reasons. Fundamentally, it is important to realize that preventive maintenance in the case of boiler feed pumps does not have the characteristics of emergency. If a pump can run 40,000 or 80,000 hours before the internal clearances need be renewed, a few months' delay in carrying out the necessary repairs will have no significant effect on the integrity and availability of the power plant.

HOW CAN WE IMPROVE INSTALLATION RELIABILITY?

Improvement of the reliability of a boiler feed pump installation can be achieved by four parallel approaches:

1) By removing or minimizing hazards of equipment failure through proper choice of design and materials and through more rigorous quality control.
2) By designing into the installation itself safeguards against hazards that threaten the integrity of the boiler feed pump through the interference of external circumstances.
3) By removing or minimizing hazards through monitoring or protective controls incorporated into the system.
4) By providing spare or standby equipment that will permit operation of the unit served by the boiler feed pumps during any down-time caused by unscheduled outage.

It should be very obvious that each one of these avenues of approach must cost money or, in some cases, efficiency which, in turn, can also be translated into mone Let us consider for instance the first item on our list. And let me state at the very outset of this analysis that regardless of the reliability of a given piece of equipment it can always be improved through more expensive design, more expensive material more research and still more thorough quality control. There is not a boiler feed pump commercially available today that could not be made still more reliable if cost and commercial considerations were not a factor. Nor is there a boiler feed pump that could not be made more reliable by increasing internal clearances at the cost of reducing the pump efficiency. Lumping price and efficiency considerations together, I have on occasion illustrated this statement by plotting the qualitative relationship between cost and probabilities of failure in a curve such as shown on Fig.

At first blush this sounds like a very embarrassing and damaging admission. Bu a few moments of objective analysis will show that there is nothing embarrassing or

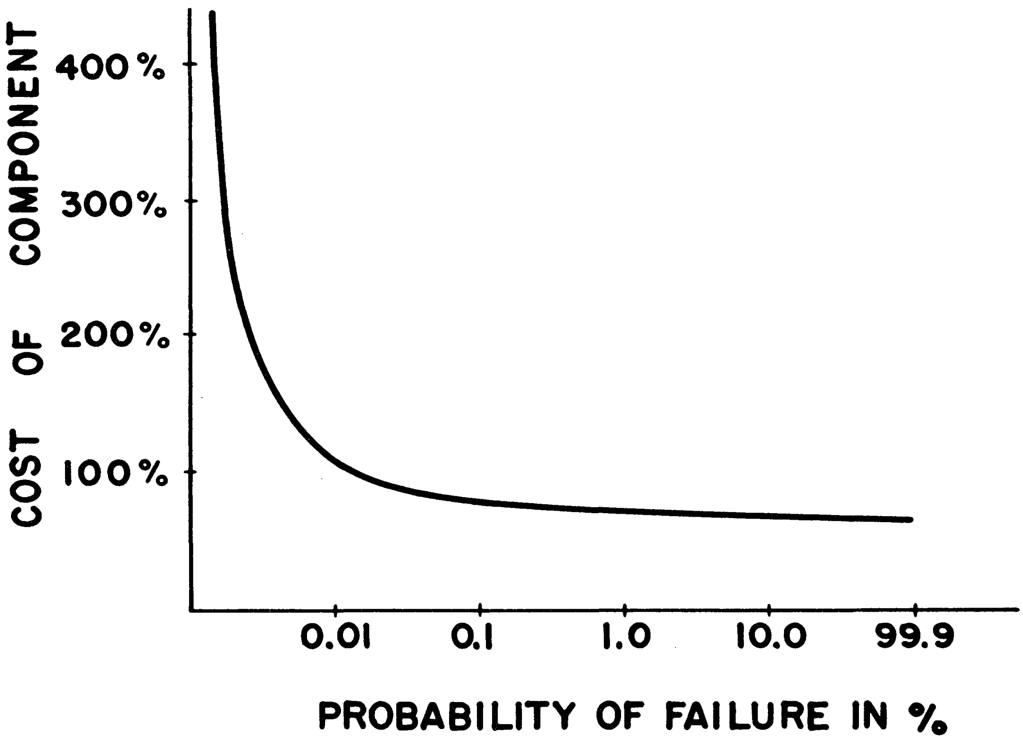

Fig. 2. Relationship between cost and probability of failure.

unethical about the position of a manufacturer who knows that he could design and build better equipment but also knows that he could not sell it. If it is true that people generally get the kind of government they deserve, it is equally true that a buyer gets the equipment he is willing to pay for.

The decision on the type of product a manufacturer wishes to produce rests on a vastly complex body of separate criteria. These criteria cannot be examined without first establishing some basic facts and formulating some basic assumptions with respect to the market the manufacturer intends to serve and the economic relations which dictate the relative success or failure of his enterprise.

The first assumption we can make is that in our free enterprise system, manufacturers are in business to make money. I might add, parenthetically, that they are given no insurance against the possibility that they will lose money and that too often the vagaries of the market place cause them to do just that.

And the first fact that we must consider is that in our particular economic system, ESSENTIALLY LIKE PRODUCTS WILL BE SOLD FOR ESSENTIALLY LIKE PRICES. This means, in other words, that the level of prices for equipment that is manufactured by one Corporation will be established by its competition. It may be difficult at times to accept the idea that competition must be viewed from the user's standpoint and not from that of an individual manufacturer. But regardless of whether one considers a particular competitor as a lower grade manufacturer or not, whether one considers his product as inferior or equal, he is an "equal" competitor if he is so considered by the prospective customers.

We must therefore translate our first fact into its equivalent: "COMPETITIVE PRICES ARE SET BY THE LOWEST COST MANUFACTURER WITH AN EQUALLY ACCEPTABLE PRODUCT." Note very carefully this "EQUALLY ACCEPTABLE" qualification, for it is definitely a fact that not all customers have the same understanding of what constitutes "equal acceptability."

If we combine this with our assumption that all companies are in business to make money, we are inevitably forced to restate our first fact into "LIKE PRODUCTS

MUST BE MANUFACTURED FOR LIKE COSTS." This is the basic reason why an increase in design, research, material, quality control or field service costs will penalize a manufacturer who wishes to build and *sell* more reliable equipment than he does now.

There is, of course, some balm in Gilead. You remember that I stated that not all customers have the same understanding of what constitutes equal acceptability. There are among utilities companies which give greater weight than others to the intangibles of service and reliability. And this is the only reason why all boiler feed pumps are not alike, why there are gradations available in the excellence of design, in the ability to withstand unfavorable conditions of operation, in brief, in the reliability of boiler feed pumps.

You might ask why shouldn't a manufacturer of boiler feed pumps follow the practice of the consumer-goods industry and offer two or more standards of reliability for utilities to choose from. This is quite customary for such products as tires, batteries or hot water tanks. In this latter case, for instance, you can get quotations on a wide variety of types, each carrying a different life-guarantee, and each with a different price. For example, you can choose from the following:

	Guarantee	Price
1. Plain galvanized steel unlined 30 gallon tank	1 year . .	$ 52.00
2. Plain galvanized steel glass-lined 30 gallon tank. .	10 years. .	$ 65.00
3. Plain galvanized steel, glass-lined 30 gallon tank . with deluxe controls and heavier metal gauge	15 years. .	$ 97.00

And if you wish to pay more and get still more for your money, you can choose from copper, cupro-nickel, stainless steel or all-monel tanks.

Let us plot these figures of price versus guaranteed life. After all, since the manufacturer guarantees different periods of operation, the probability of failure in a given period of time must also be different. Is it not remarkable to what degree this curve (Fig. 3) resembles the imaginary curve of cost vs. probability of failure plotted in Fig. 2?

Unfortunately, designing several lines of boiler feed pumps, each with a varying degree of reliability is not very practical for a variety of reasons. Not the least of these would be the psychological disadvantage of classifying utilities into what might be interpreted as first, second and third class citizens. Nor is it very practical for a manufacturer to react to the preference of just one customer who is willing to spend considerably more for a redesigned product that is normally part of a standard equipment line. If the entire cost of the redesign and of the special handling is assessed against this one customer, along with whatever additional cost in labor and material may be involved, the increased reliability will cost toomuch and the customer may not be able to justify the purchase. If this cost is allocated against a greater number of units in the expectation of selling these units in the future, the manufacturer runs the risk of having to ultimately write off most of the cost of the redesign because of optimism on his part with respect to the number of potential customers for this improved product.

The above should be understood as a generality and does not eliminate the possibility of such a transaction in some special case. It would be a tempting challenge to be able to design a super-reliable pump with a complete disregard for cost considerations. I am not quite certain of the lines along which such an attempt would proceed. One does not break certain restraints all too suddenly. The habit of weighing what every departure might do to the cost of a product will have to be eliminated before one could address oneself to such an undertaking.

Let us return for a minute to the curve of costs vs. probability of failure on Fig. 2. It illustrates very vividly the fact that there is a point where very minor savings

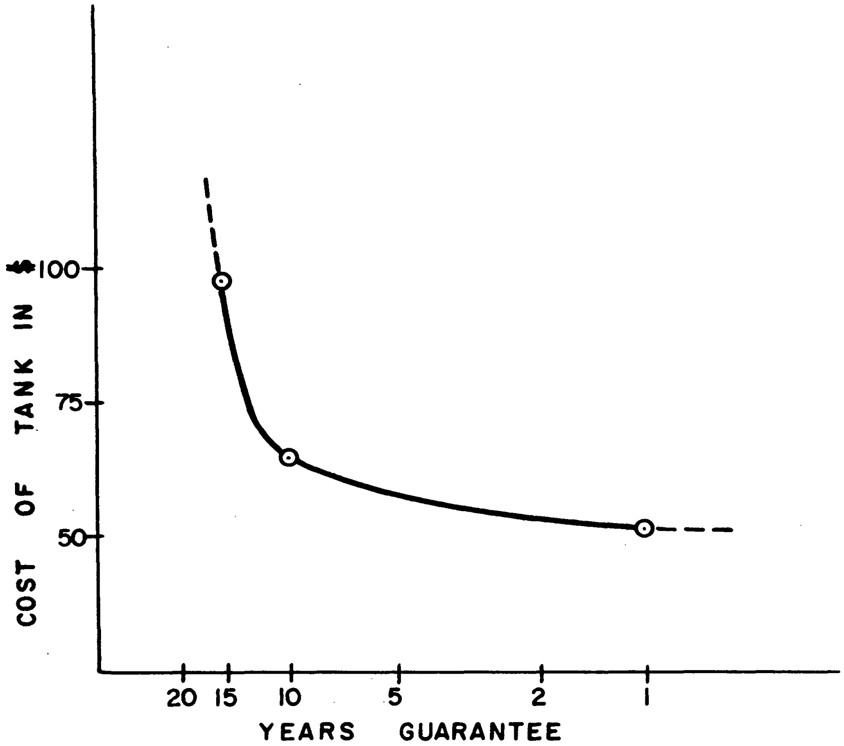

Fig. 3. Relationship between cost of hot water tank and guaranteed life.

will result in a tremendous increase in the probability of failure. There is also an area beyond which any further reduction in this probability can only be achieved at an increase in cost which cannot be justified. As I once said in the paper referred to earlier, the general shape of the curve on Fig. 2 can teach us the futility of buying too cheaply or too dearly.

OTHER MEANS OF IMPROVING RELIABILITY

We have mentioned that in addition to improving the boiler feed pump itself, there are three other means available to us for improving the overall reliability of the installation. One of these is to design into the installation safeguards against hazards from sources external to the pump itself. Probably the best example of doing this is the use of sound judgment and thorough analysis in designing the installation so as to provide adequate suction conditions under all possible circumstances. The transient operating conditions that prevail in an open feedwater cycle installation immediately following a sudden load drop are well understood today and there is absolutely no reason why the margin necessary to provide against these transient conditions cannot be calculated accurately.

Nor is there any reason or excuse to employ excessive optimism in establishing a safe and sound value for the required submergence. But adequate suction conditions may frequently entail the expenditure of additional costs in the layout of a land-based central steam station and I must unfortunately observe that occasions still arise from time to time where the reliability of the boiler feed pump proves to be impaired because these additional costs were not considered to be justified.

When it comes to the application of monitoring or protective controls which would remove or minimize some of the hazards which can interfere with the reliability of operation, I feel that the majority of today's boiler feed pump installations could be more liberally supplied. This is particularly true with respect to controls which would counteract the effect of insufficient NPSH or of inadequate condensate injection sealing supply. While some boiler feed pumps are capable of operating in a flashed condition with an excellent chance of coming through unscathed and while this transient condition may be of relatively short duration, it is still preferable to provide corrective controls that will obviate this condition. As to inadequate condensate injection sealing supply, this condition will almost always result in a partial or complete failure. Therefore, if proper protection is to be afforded the boiler feed pump, these two controls are essential.

In general, because of the increasing complexity of the boiler feed system, increasingly sophisticated control systems are required to protect the boiler feed pumps. But at this moment, there is very little agreement among utilities as to the need of supervisory and monitoring controls for this service. It is therefore commercially impractical for the manufacturer to include any such controls as standard equipment, lest the user assume that these controls are necessary for one make of pumps and not for another. Of course, I have some definite ideas regarding what controls are essential if reliability is to be improved, but whether many utilities would purchase them if they were included with the boiler feed pumps is questionable

STANDBY EQUIPMENT

The final area I wish to explore is the question of standby or spare equipment. Until a few years ago it was a standard practice for utilities to install either two full-capacity boiler feed pumps or three half-capacity pumps. In the first case, if one pump — or its driver suffered a casualty, the second pump would permit the unit to carry full load without interruption. In the case of three half-capacity pumps, it was assumed that the two pumps in service would seldom suffer damage simultaneously and therefore the half-capacity spare pump provided adequate protection. In my opinion this assumption was fully justified.

But as the size of the units served by these boiler feed pumps grew and as the operating pressures increased, the cost of providing spare equipment mounted very rapidly. At the same time it was observed by the utilities that the reliability of the pumping equipment had been improving very markedly. The temptation to eliminate spare equipment became very strong because of the important dollar savings that could be achieved. Today, when half-capacity boiler feed pumps are used, the overwhelming majority of the utilities install but two pumps and no spare. If a pump designed to handle the full capacity requirements is used, a partial spare may be installed, capable of carrying from 33 to 60% of the normal flow requirements. And several installations of full capacity boiler feed pumps have been made without any spares whatsoever.

My personal opinion is that an installation of two half-capacity boiler feed pumps without spares is fully justified for a land-based central steam station. In most instances, an unscheduled outage will reduce load capability of the unit to about 60% rated and the deficiency can be easily compensated from the reserve capability of the system. The use of a single pump with no spare, on the other hand, appears to be less justifiable at this moment, especially if means are not being employed to protect the installation against failures caused by circumstances external to the pump itself.

But maybe we can examine the wisdom of these decisions numerically. Maybe we can calculate the cost incurred by an interruption in service and compare it to the savings resultant from the reduction in the costs of the installation? Two papers have been presented before technical societies within the last few years that contain data which can throw light on this matter. The first of these[2] indicates that an

[2] "Economic Choice of Generator Unit Size" by L. K. Kirchmayer and A. G. Mellor. Paper No. 57-A-154, presented at the Annual Meeting of the ASME, New York, December 1-6, 1957.

increase in the forced outage rate of a main unit of 1% requires that the reserve capacity of the system be increased from 4 to 5% if the same standard of system reliability in meeting the load is to be maintained. The second paper[3] assumes that the forced outage of a boiler feed pump and all of the directly associated equipment is 1% and calculates the cost of providing the necessary reserve capability for a number of different pump arrangements. The paper reaches the following conclusions:

Arrangement	Increase in forced outage of main unit	Required Increase in total capability in % of main unit
1) One 100% Capacity pump	1%	4%
2) One 100% and one 50% Capacity pumps	0.406%	1.62%
3) Two 50% Capacity pumps	0.802%	3.208%
4) Three 50% Capacity pumps	0.012%	0.048%

These figures can be applied to the installation of a 300 MW, 2400 psi unit which is assumed to cost $125/kw. If a 4% increase in reserve capacity is required for each percentage point increase in the main unit's forced outage rate, the following dollar penalties have been calculated:

One 100% Capacity Boiler Feed Pump	$ 1,500,000
One 100% and one 50% Capacity Pump	610,000
Two 50% Capacity Boiler Feed Pumps	1,200,000
Three 50% Capacity Boiler Feed Pumps	18,000

From the foregoing one might assume that the cost of a 50% capacity pump would have to reach $890,000 (the difference between $1,500,000 and $610,000) before it is worth while to eliminate it and operate without any spare whatsoever. Or that this same spare would have to cost $1,182,000 (the difference between $1,200,000 and $18,000) before it would be sound practice to use two half-capacity pumps instead of three. Wherein lies the fallacy if this not be so?

The problem, of course, is that we do not know exactly the forced outage rate of boiler feed pumps, as we have already said before. It is strange to note that in other industries some very thorough statistical information has been compiled. I would like to cite one paper[4] for instance which provides a total of 46 separate tables listing electrical equipment failure statistics in industrial plants. An interesting example taken from this paper gives the annual failure rate from all causes for large induction motors 250 hp and larger as 9.86%. This rate was not established from a small sampling: 46 plants in USA and Canada were involved in the survey and the total number of motors in question was 1,420.

Obviously, the record of electric motors in a utility will be superior to that of motors installed in industrial plants. At least, I am totally unfamiliar with such a high percentage of failures. But if this record is better, it seems to me that it might be ten times, or twenty times better, but not 100 times better. Thus, the assumption of a 1% rate of forced outage for a boiler feed pump and all of its related equipment, including its driver, does not seem to be so outrageous. But until we decide to accumulate more accurate statistics in this connection than we have today,

[3] "Auxiliary Drives and Factors Affecting their Selection" by J. J. Heagerty. Paper 61-989, presented at the AIEE-ASME National Power Conference, San Francisco, California, September 24-27, 1961.

[4] "Report on Reliability of Electric Equipment in Industrial Plants" by W. H. Dickinson, Paper No. 62-61, in the July 1962 Transactions of the AIEE.

we shall not be much wiser and we shall not have any better criterion in deciding whether to include a spare pump or not than we have today.

We should take advantage of the analysis developed to evaluate the effect of possible forced outages on the additional investment required for reserve capacity and see whether it cannot shed a new light on making a choice between several boiler feed pump offerings. Let us return to the example of a 300 MW, 2400 psi unit with a cost of \$125/kw. We have seen that an increase of 1% in forced outage rate requires the addition of 4% in reserve capacity, or 12,000 kws. Thus, an increase of 1% in forced outage should be penalized by \$1,500,000. If one boiler feed pump design were to be as little as 0.10% less reliable than another, it should be assessed a penalty of \$150,000 for purposes of comparison with the more reliable equipment. Because this figure is of the same order of magnitude as the price of two boiler feed pumps of half-capacity designed to serve such a 300 MW unit, we can state our conclusions in another way: if two such pumps were offered free of charge, it would not be economical to accept the gift, if these pumps were 0.10% less reliable than some others. We have already stated that it is difficult to establish an exact difference in reliability between two or more boiler feed pump designs, because of the lack of adequate statistical evidence. But it would seem to me that any design which has greater reliability by virtue of lower shaft deflections and larger internal clearances should certainly merit an evaluation of these intangible characteristics.

CONCLUSIONS

There is no question that a great many differences exist between naval ship propulsion machinery and that installed in a land-based central steam station. There are also differences in the operation of this machinery as well as in the manner in which it is selected and purchased. Nevertheless, I believe that in our desire to improve the reliability of naval ship propulsion machinery we can learn a great deal from the practices employed in central steam stations. We can learn much of what we should do and we can see the hazards to which we expose ourselves if we fail to do so.

Before new approaches can be developed to the improvement of the reliability of steam power plant components, the major causes of failure must be defined more systematically and with more precision than they have been in the past. A more comprehensive analysis must be made of all the interactions between the various components of the steam plant. A greater store of statistical data on availability will be required to establish a more exact basis for reliability evaluation.

But first and foremost, if we wish to endow this equipment and these plants with optimum availability, more than lip service must be given to the evaluation of intangibles in the selection of equipment and in the design of the system in which it is to operate. We must free ourselves of the prejudice that we can get the best, the most efficient, the most reliable equipment — at the lowest bid price.

DISCUSSION

DUNN (Electric Boat): In the first place I don't think the comment that Schirra made is a laughing matter at all. I don't think he made it really. I think that is a big story. I've heard it any number of times, and I don't think it is really fair to laugh at it. By laughing at it we infer that the lowest bidder does have a reduced quality or reliability piece of equipment. What Schirra should have been thinking of as he lay on his couch or stepped into the capsule was the fact that a great deal of money may have been spent to convert that capsule and machine from three sigma which is 99.7 by doubling it making it 6 and then it is quite obvious somebody added one for good measure in making it seven sigma. For us at least this is a better way to think than any laughable aspects of low priced bidding.

I have a couple of additional comments that I would like to make. You say the feed pump can be made more reliable at an increased cost. Over and over again people dealing with vendors hear this statement, and I would like to call to the attention of all vendors the fact that some of us users of their equipment would like to know the answer to the question "more reliable than what?" We have listened to your sales engineers and I see some of them around here who have been operating for many years. They tell me how reliable their equipment is, how much quality is put into their equipment. I feel I have a right to know at no increased cost how they know that their equipment is as reliable as they say, what is the basis for their statements, how their management evaluates the engineering and design departments performance in the design phase to judge for themselves whether they will allow their sales engineers to go out and make these claims. On this basis, then, if I knew how reliable their equipment actually was that they were trying to sell me I would better know whether I could afford to pay more dollars to make it still more reliable. A trade off if you will.

Third Comment: The expense of being redundant, I think that the kinds of things that we're talking about, shipboard machinery which have long mean times between failures, relatively long compared to the electronics or aerospace piece parts business. I think we would be much better advised to look at these failure rate curves in terms of not infant mortality and random failure, but in the light of their proper context as a debugging period, which is a term we all use for this so called infant failure period. Then in the random failure area this is what we all term the useful life. This is the useful life of the equipment. Going back to what I said before about the sales engineer, I feel that I have as a user the right to ask a vendor what his judgment — what his specific quantitive claim is of the useful life of his piece of equipment.

Then I can evaluate the need for spare parts along with it, and I would be in a lot better shape reliability wise than I've ever been before.

KARASSIK: I'll plead guilty to have made laughing matter of Schirra's supposed comment. However, my extenuating circumstances are that we can frequently learn by going to extremes. I was driving at the extreme interpretation of the meaning of the remark: the cases where no evaluation whatsoever is made of the intangible contents of the package that is being supplied.

With reference to your second comment dealing with "more reliable than what?", my reference to price or cost included a reference to efficiency which can be translated into a cost since it affects fuel costs. There is no question that if I were to double the internal clearances at the running joints of a pump, I would reduce the possibility of contact between stationary and rotating parts that may be caused by external circumstances. Therefore I would lengthen the life between overhauls by reducing the possibility of random failure. When it comes to wear-out of high pressure boiler feed pumps, we can give it less attention, since we have documented cases of pumps running from 100,000 to 150,000 hours before renewable internal parts need to be replaced.

I did say that it is difficult to assign a meaningful number to the degree of reliability of a piece of equipment such as a boilder feed pump. If we eliminate the debugging period and eliminate random failures I can predict the ultimate life between overhauls quite accurately. But random failures do occur. And in my lexicon, random failures do not refer so much to failures of a component but rather to failures caused by external hazards. One example would be the failure of a pump which flashes and which is shut down in a flashed condition so that it coasts down going through its critical speed while steam bound.

This can be a random failure that may occur after 10 years of operation and internal contact under these conditions will lead to failure. I certainly would not call this a wear-out failure, nor would I put it under the heading of "debugging," because I assume that provisions would have been taken to prevent flashing in the early life of the pump. Therefore when this occurs 5 or 10 years after installation, it is a random failure.

It would be most gratifying to the designer and the manufacturer if a tabulation could be produced giving you an exact predicted useful life and an availability report with plus and minus tolerances. But unfortunately, the exposure frequency is insufficient to have such a report mean anything. It is easy to make predictions when you are dealing with 200,000 transistors or diodes because statistics based on such large numbers are meaningful. But by the time we have finished building 20 or 30 pumps of a particular type, pressures have gone up, capacities have gone up, certain other characteristics have changed and we are building a different pump. So that when I am reduced to 20 or 30 items, the sampling size is such that I cannot make accurate predictions. This is the problem.

DUNN: That's quite true, but there are certain predictions you can make from small samples and even the samples that cross the board in terms of pumping. Lately I've looked at approximately 2,000 pump incidents of all kinds during this so called debugging period or infancy failure period, and while I don't have all the data reduced at the present moment, it looks very much like 80-85% of the failures in the debugging period were quality control oriented production failures. There may be 5% design oriented troubles and malfunctions. The overall design aspect of the pump business seems to be very well taken care of, and I think that with the proper quality control, which the pump manufacturer can give to us, a life expectancy in reasonable terms based on the limiting items (or worst case parameter) can be predicted. I'm not talking about exact numbers, but reasonable terms from which we can establish performance and endurance margins and look to a preplanned repair or overhaul period, plan for maintenance and repair in such a way that the life expectancy of these shipboard components can be repeated in cycles throughout a long period of time perhaps equal to the life of the ship itself.

KARASSIK: It would be quite interesting to have access to such a large statistical sample and I would definitely welcome the opportunity of examining it at some time.

I am not in the position to state whether the incidents of failure caused by quality control occur in the same ratio for marine application and in land-based utilities. I think not and I would say that in the case of high pressure boiler feed pumps, my experience indicates that about the reverse percentage of that you have quoted would prevail.

Now you have yourself mentioned something which lends support to my thesis that increased reliability can be achieved at an increase in cost. You said that in your opinion, increased quality control would effectively reduce the number of failures. I submit that increased quality control does justify the extra cost incurred. But while some customers are prepared to pay this extra cost, I have yet to find an engineer who will tell me "I will pay X dollars for such and such a percentage increase in reliability." He will say that since this increase cannot be proved, he will have to use his judgment as to what is an acceptable minimum. Then if 3 or 4 manufacturers meet this acceptable minimum, they will be judged equally in the balance when bids are compared and the decision will be made strictly on price.

This is the problem that we face. We have not yet learned to deal with intangibles as readily as we deal with little black scribbles on a piece of white paper. We fail to remember that if we are going to deal strictly with those things that can be represented numerically and do not permit ourselves to exercise judgment, we engineers — I on this side of the lectern and you on the other side — we engineers will become superfluous. We shall be replaced by computers. I can visualize a time when every user will have a computer connected with the computer of every manufacturer. After we go home at five o'clock, the East Cupcake Power and Light Co. of Illinois computer will send a message to Worthington and all other manufacturers to the effect that it requires let us say a pump or a condenser. The computers of all the manufacturers will spin the wheels of their computers through the night. By morning the bids would have been sent in; the computer at East Cupcake, Illinois would have analyzed the bids and awarded the contract. In the morning, when I come to the office, I will learn that we have received an order to build such and such boiler feed pumps and we have lost some others. This is what will happen if we do not learn that engineers should be allowed and even "enjoined" to use our judgment in making decisions.

DUNN: Judgment is certainly an integral part of this business, but there are wavs to determine the confidence that one places in a "good engineer's" judgment. While you may not be able to get a precise numerical substitute for judgment, you can assess, in close to numerical fashion, the confidence that you place in someone's judgment. This is based on your past experience with that person, and the, let's say, the consistency and validity of his claims.

KARASSIK: Let us say then that we both, manufacturers and users have to refine our methods of analysis, and means of applying our judgment, and I accept the challenge. I cannot say that I shall deliver that within so many days, months and years I will be able to produce more sophisticated data on life expectancy of equipment, but let me assure you that as a goal it is a very worthwhile one and I repeat, I accept the challenge to try.

JACKSON (Dept. of Defense): When you were speaking of limited samples, to what extent do you use overstress analysis?

KARASSIK: Within practical limits we apply not only overstress analysis but also in certain cases destructive testing of components when they are first designed as the first in a series.

Let us consider boiler feed pump impellers, for instance. Years ago, we would test all impellers at 15% overspeed. This approach was found inadequate. If we did not destroy or distort an impeller at 15% overspeed, we had no assurance that the limit would have been reached at 17 or 20%. Instead, we now test the first impeller of a new design at 100% overspeed, having first checked out with overstress analysis whether this impeller could stand this overstressing. Then we feel it is no longer necessary to test each consecutive impeller at 15% overspeed.

The problem of having a limited number of samples is only part of the picture. Design analysis and a few tests can give us reasonable assurance that a design is sound. This does not, however, give us any assurance that the parts are repeated with the necessary fidelity to the design. It gives us no assurance as to the homogeneity of the material, as to the accuracy of the tolerances or as to the accuracy of the assembly process. If it operates at its ideal best, a quality assurance program is directed at giving us this particular assurance.

But I have been speaking of land based power plants which differ one from the other as much as night and day. This is different from the case of navy equipment. Here you build a type ship, after which the design will be repeated a number of times. This seldom happens with land-based power plants.

It is these differences between plants, between systems, arrangements, piping and valving arrangements, between the choice of other components that magnifies our problem. Yesterday we heard much said about "interaction." The listing of possible interactions that affect a boiler feed pump, from the point of view of other pieces of equipment or of events arising outside the boiler feed pump would cover pages and pages and pages.

Thus, this matter of the small sampling does not give us a problem from the point of view of design or design verification, but rather from the point of view of, (a) interaction with other equipment, and (b) the vagaries of operation which may be encountered. For in my opinion, it is not practical to design a boiler feed pump that will withstand any and every cataclysm that might — and has — been visited upon a boiler feed pump.

One might think, for instance, that a block of wood about 4" in size should not find its way into the boiler feed system of a power plant. It would be illogical if I were to design a pump that could stand that particular overstress. But such an event has taken place more than once.

I could list other examples. Overstress analysis is a sound tool as far as design is concerned, but it covers only a portion of my problems, — and I have so many others to face. So many others that are a more serious and more frequent deterrent to the integrity of the equipment involved, that I am looking for more tools. One of the tools that I am looking for is a sense of logic on the part of the designer of the system involved.

FRANKEL (MIT): First of all, I believe we should all be indebted to Mr. Karassik for so ably focusing our attention on the relationship between the reliability economy and performance.

Part of my question has just been answered, but I would like to make a comment, particularly with regard to random failure, chance-failure period during the life, the useful life as it has been developed for our pump or sub-system.

Somehow I believe the audience may get away with some impression that chance failures are mainly environmental while, on the other hand, and this is the big difference between consumer goods and goods like a feed pump, for instance, that all feed pumps — the one with the higher reliability and those with lower reliabilities — are actually designed for the same wear-out period which means for the same total useful life. But obviously, the lower reliability and possibly a lower cost pump will have a much higher chance rate of failure which, thereby, though both of them may work in the same environment, the lower reliability pump will definitely have some inherent factors which increase its chance of rate of failure.

With this point of view, I would also like to focus attention on some very important work done by Briepohl of Sandia Corp. in California and other people which are quoted in his work. Your plot # 2 which you said is just qualitative, can actually provide a quantitative measure. These people have investigated a large number of electronic and simple mechanical components and have come up with the conclusion that a gap-relating cost and probability of failure of your form is applicable in every case and can actually, and I believe I used it in my paper on page 95, be expressed as an exponential function to the minus a constant times the probability of failure divided by the probability of failure. I believe this should be a very useful tool in getting quantitative cost functions and their relation to reliability.

KARASSIK: Thank you very much. I shall address myself to this last comment to begin with. I am very pleased to be given this reference. Maybe a qualitative curve that I dreamed up one night may prove to be reasonably practical and can be replotted in more quantitative terms.

I would like to correct the impression I may have given that I consider all random failures as environmental. What I meant is that they may well be a combination of environmental and inherent. Where, for instance a forging inclusion will cause shaft failure after five years of operation, the failure will frequently be triggered by an environmental factor. Now it is quite correct, as Professor Frankel has said, that different degrees of reliability will make a piece of equipment more or less resistant to an environmental triggering mechanism. You can put it as simply as this: we can place 5 pumps side by side to operate continuously at base load conditions, with no interruption of load or any other changes that might distract them from their normal appointed duties. All pumps will probably last about the same number of years, or at least have a life of the same order of magnitude. But if you now start imposing certain external stimulants, introduce certain hazards, then one pump will fail first, then a second pump, then the third and finally all five pumps will fail. There is no pump in the world that I personally could not wreck. You can always develop externally imposed conditions that are severe enough to wreck any boiler feed pump, including one that could be built with complete disregard for cost, at 3 or 5 or 10 times the normal cost for such a pump. You can always wreck anything. The human race has long ago proved this to its disadvantage.

Thank you, gentlemen.

RELIABILITY IN THE AUTOMOTIVE INDUSTRY

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Two design methods are presented here to provide for the variability in the life of automotive components: one based on life at a given stress, the other on stress at a given life. The former is expressed in terms of Design Life Factors; the latter is given as Design Stress Factors. Knowing the mean value of the stress or of life for a given part, the design life or design stress may be calculated. This can be done for a given percent reliability and thus for the maximum number of failures that one can economically tolerate. Of the two methods more information is available for the Design Life Factors than the Design Stress Factors. The study is based on test data on automotive components (axles, crankshafts, gears, valves, etc.) and on some material specimens.

Introduction

In the design of automotive components to resist fatigue a number of factors are used to account for surface finish, type of loading, mode of loading, size, etc. These factors have been experimentally determined for various materials and operating conditions. However, because of the inherent scatter in the engineering properties, each of these factors represents a mean value computed from the scatter of the observed data.

The performance of an automotive part in service is judged by the number of failures or time to failure of a small number of components rather than by the mean life. This, in turn, depends on the degree of scatter. Thus in a meaningful design it is essential to incorporate a reliability factor which would provide for this scatter.

Reliability can be defined as the probability of a product performing a specified function, under given conditions, for a specified period of time, without failure. Thus, the function of the part, the operating conditions, and the time of operation are all important aspects of reliability.

In statistical terms reliability is the converse of the probability of failure. If the probability of failure is 1 the part will fail and, therefore, the reliability is zero. Similarly, if the probability of failure is zero the part will not fail and reliability is 1 or 100, if expressed in percent. Thus, 80% reliability means 20% failure, 95% reliability means 5% failure, etc.

Thus, reliability implies an avoidance of failure. In the case of automotive components this means principally fatigue failures as most automotive parts are subjected in service to repetitive loading.

The Problem of Scatter

It is the fundamental characteristic of manufactured parts that they exhibit variation in life when subjected to identical loading conditions. Aside from the

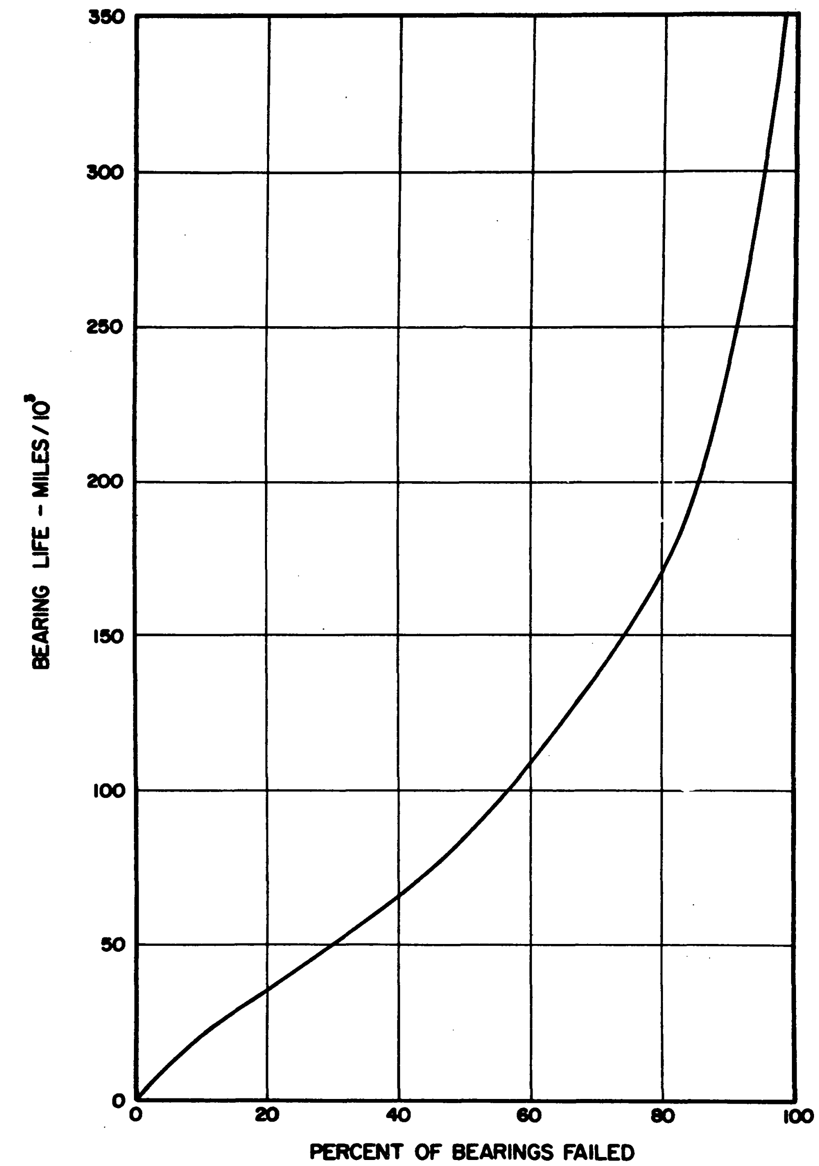

Fig. 1. Life Expectancy of Ball Bearings.

variations resulting from the human error in testing, or from the limitations of the test equipment, the principal variation lies in the parts themselves. No two pieces produced are exactly the same, no matter how refined is the process. Although the differences may be small, they nevertheless exist.

This is illustrated (1) by the anti-friction bearing in truck front wheel spindles (Fig. 1). Although the rated life is 100,000 miles, 10% of the bearings may fail under 20,000 miles and another 10% will last over 200,000 miles.

All automotive parts show scatter, some to a greater degree than others. Engine exhaust valves, for example, exhibit considerable variation in life (2). In Fig. 2 each group comprises a number of engines, all of the same design, which were tested in fleet operation. The life of a valve shown is an average between the first and the second valve that failed in a given engine. Similar tests conducted on the dynamometer (Fig. 3) under controlled conditions show about the same degree of scatter as in fleet operation. This suggests that scatter is an inherent characteristic of a fabricated part.

Faced with this problem of scatter it has been intuitively recognized that it will not suffice to evaluate an automotive part from a single sample and a recourse

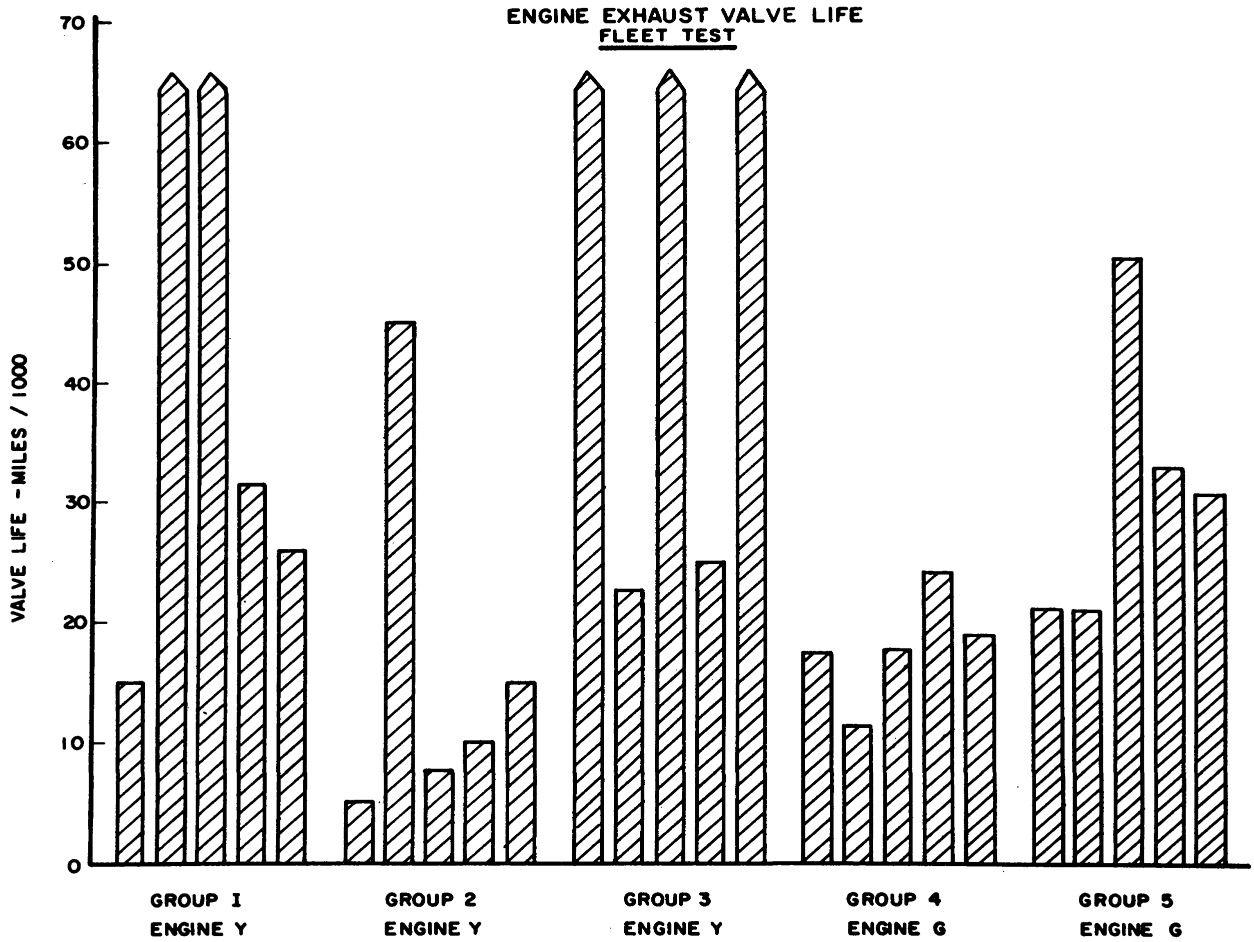

Fig. 2. Scatter in Life of Engine Exhaust Valves - Fleet Test[2]

must be taken to an average (mean) of a number of samples. This has been a widely accepted method of evaluating data.

The use of the average alone has a serious drawback. Half of the specimens have a life lower than the average and, therefore, a design based on the mean value implies a 50% reliability, that is, half of the parts will fail. This would be an intolerable condition and the reason that 50% failures do not occur in actual practice is because in design calculations based on mean values generous factors of safety are provided.

This has been recognized for some time and efforts have been made, mostly on empirical basis, to focus attention on the low end of the scatter band. Fig. 4 refers to considerable data accumulated on bevel and hypoid gears (3). The recommended design line for maximum of 5% failures is obviously much more meaningful than the commonly used mean line which implies 50% failures. The only shortcoming of the method shown in Fig. 4 is that it necessitates collection of considerable amount of data, generally beyond the capacity of an average industrial laboratory.

This study was undertaken to provide a useful basis for design of automotive components which would provide for the inherent scatter in their lives.

The Log-Normal Distribution

Meaningful analysis of scatter data necessitates the use of statistical methods (11,12,13). The dispersion, or scatter, is described by a numerical factor. The usual measure of standard deviation can be used in connection with a normal

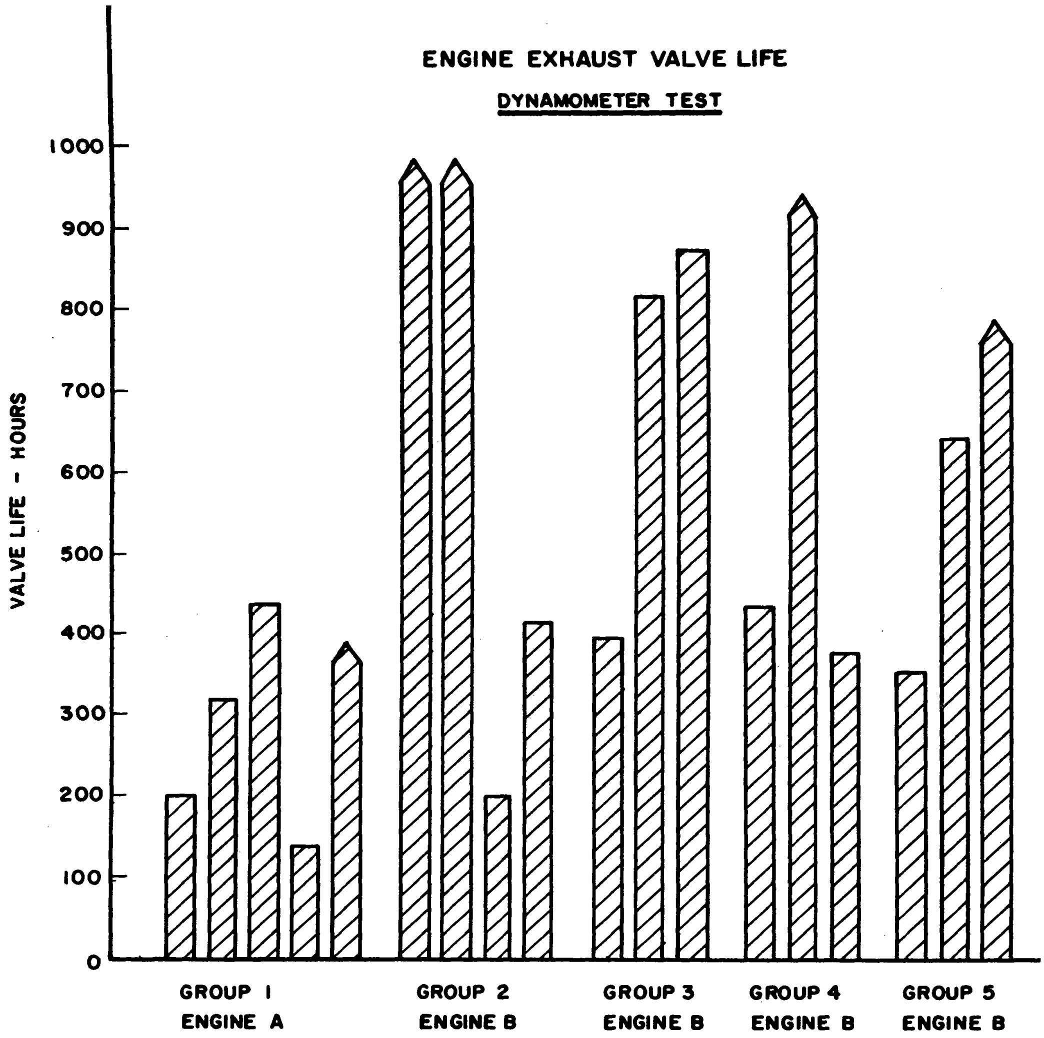

Fig. 3. Scatter in Life of Engine Exhaust Valves - Dynamometer Test[2]

distribution of occurrences, or the log-standard deviation can be used with the log-normal distribution. More refined methods, such as the extreme value distribution, the probit method, or the staircase method of fitting a regression line to the distribution did not offer any higher accuracy in the present case.

For this study the log-normal distribution was used (5,6). The reason for using this distribution is that fatigue data plotted on a linear scale show a skewed distribution (4,5) (Fig. 5). The explanations for this are numerous. One of the most logical ones is that a limit of values exists at the lower end of the scale, as a part cannot have negative life or strength, while no limit exists at the upper end of the scale. Further, one very large value will have greater effect on the mean than will several small values. Thus, the mean tends to be above the peak value of occurrences.

Through the use of logarithms, a skewed distribution of values can be transformed into a close approximation of a normal distribution. The statistical methods applicable to a normal distribution are much simpler, are readily applied and are as meaningful as other statistical methods. For this reason, all statistical operations have been performed with the logarithms of life. Several checks performed with the present data confirmed that the resulting distributions were nearly normal.

Fig. 6 shows one such check. Line A is a plot of bevel gear data based on a log-normal distribution of fatigue life of bevel gears. This method, discussed

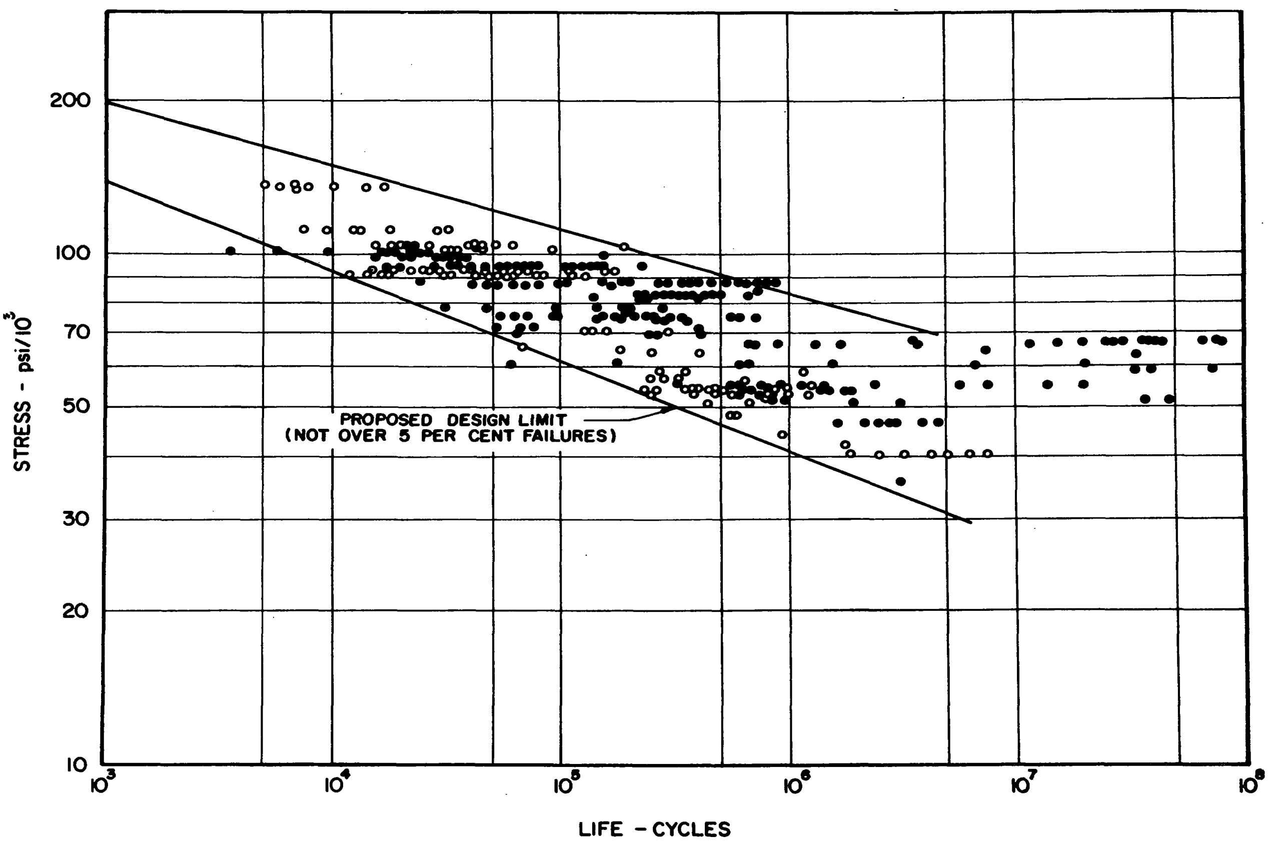

Fig. 4. Scatter in Life of Gears-Bevel: Solid Circles; Hypoid: Open Circles[3]

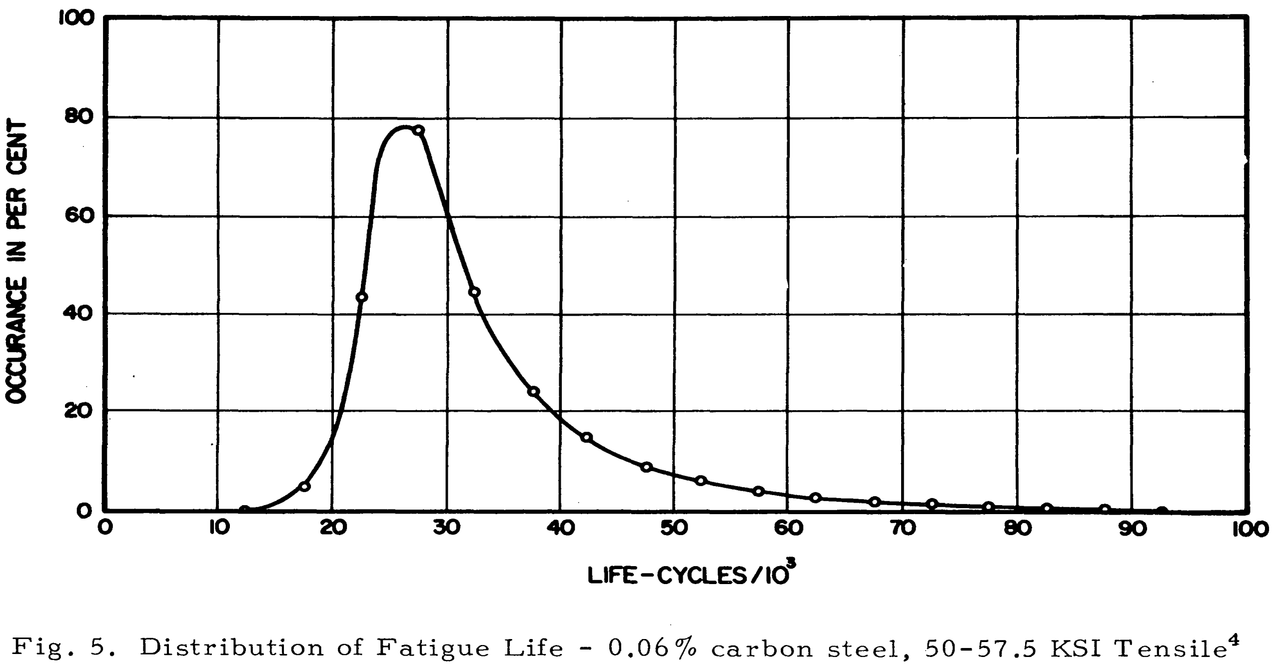

Fig. 5. Distribution of Fatigue Life - 0.06% carbon steel, 50-57.5 KSI Tensile[4]

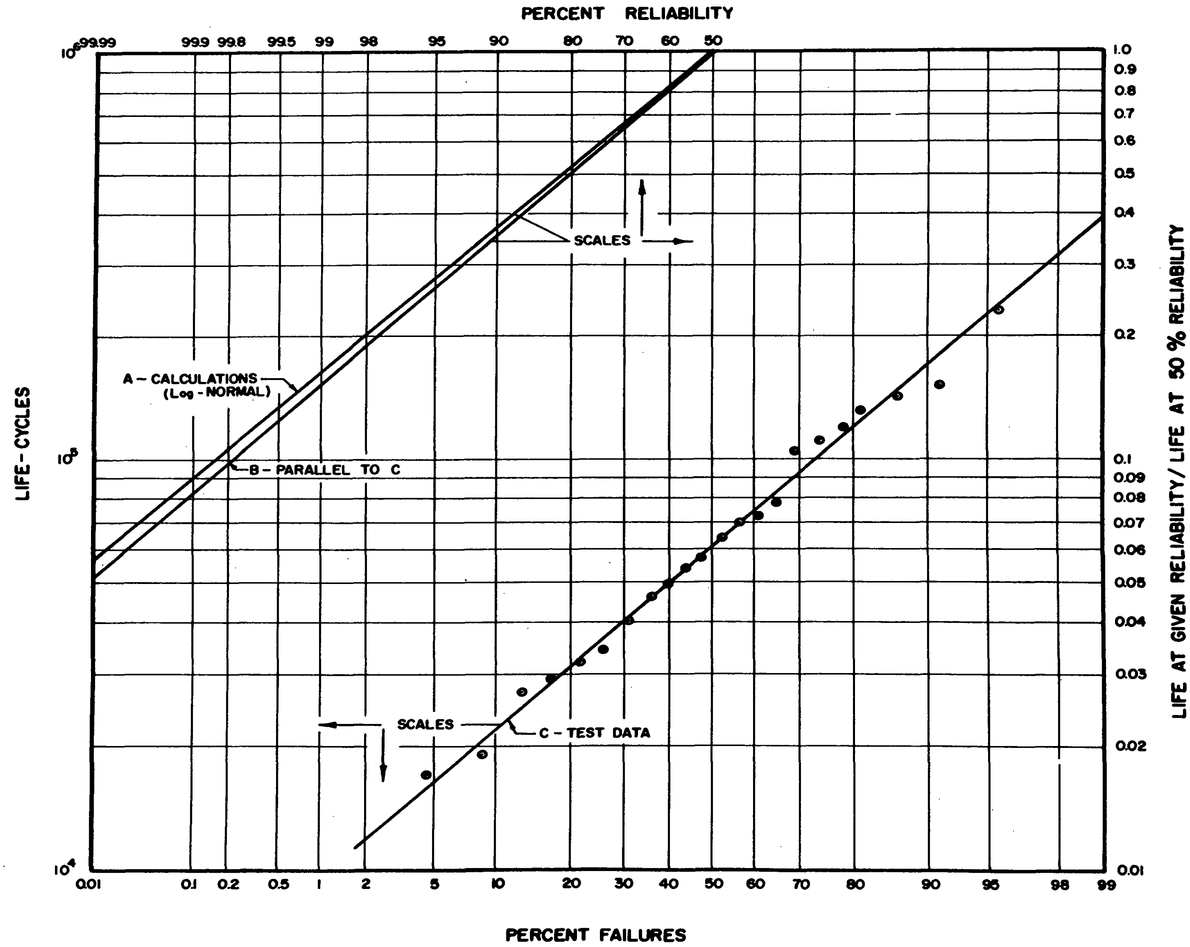

Fig. 6. Experimental and Calculated Distributions of Bevel Gear Fatigue Life.

later in the report in greater detail involves the conversion of life N to log N and then finding $\overline{\log N}$ (which is mean log N) and log N (standard deviation of log N) and plotting the ratio of $N/\overline{N}$ (right hand ordinate in Fig. 6) for different levels of reliability (upper abscissa).

Line C is a plot of the life test data on a logarithmic probability paper. This method is based on the fact that the cumulative distribution function of any log-normal distribution will plot out as a straight line on the logarithmic probability paper. Hence we can tabulate our data in the order of failure and assign them rank $R = m/n+1$, where m is the ranked order of a given specimen from a total of n specimens in a sample. That is, if we have specimens whose lives are N_1, N_2, N_m .... N_n we assign them ranks $1/n+1$, $2/n+1$ .... $m/n+1$, ... $n/n+1$ respectively.

These ranks are then plotted on the Percent Failures (lower abscissa) and the lives on the life-cycles (left hand ordinate) and we plot these on the logarithmic probability paper. Next we fit the best straight line to this plot (by least square method). If the distribution of the data were log-normal we would get a straight line on this paper. This is the case as evidenced by line C. Now if we wish to plot the ratio of $N/\overline{N}$ for different reliabilities, all that is necessary is to draw a line through 50% probability, parallel to line C. This produces line B which is quite compatible with the calculated line A.

The statistical analysis of fatigue data can be handled in two ways. A distribution of stresses at a fixed life can be found, or a distribution of life at a particular stress. In the first phase of this study the distribution of life values at a given stress was considered. Most of the test data obtained on automotive parts was available in this form. Generally, the load or stress was set and the parts were run to failure. The number of cycles was then recorded for each part and the scatter observed. This may have been repeated at a different stress level or at several levels.

The most common measure of dispersion of occurrences is the standard deviation. The most important property of this parameter is that 68.23% of the occurrences in the normal distribution are contained within the range of one standard deviation above and below the mean value; 95.45% within $\pm$ 2 standard deviations; and 99.73% within $\pm$ 3 standard deviations.

This property is especially useful for the purpose of finding a scatter factor, as the maximum percentage of failure can be readily expressed as a certain number of standard deviations from the mean. Furthermore, because the number of occurrences outside of a particular standard deviation range includes, both parts of very long life and parts with short life, a range of three standard deviations is equivalent to a failure level of .135%, or a reliability level of 99.865%; two standard deviations 2.28% failures, or 97.72% reliability level; 1.645 σ gives 5% failures or 95% reliability; 1.282 σ gives 10% failures or 90% reliability; 1 σ 16% failures and 84% probability; and .675 σ gives 25% failures and 75% reliability level.

Standard deviation in the log-normal distribution can be calculated from the following expression (5):

$$\sigma_s = \left(\frac{\Sigma\,(\log N)^2}{n} - (\overline{\log N})^2 \right)^{\frac{1}{2}}$$

where

σ_s = standard deviation (actually the "best estimate" of the population variance)

$\overline{\log N}$ = mean value of the logarithms of the life values to the base 10

$\Sigma\,(\log N)^2$ = summation of the squares of the logarithms of the life values

n = number of specimens in sample

Knowing $\overline{\log N}$ and K σ (where K is the desired multiple of σ) by subtracting K σ from $\overline{\log N}$, the log of life corresponding to a given reliability is obtained. By taking the anti-log of this life and dividing it by the mean life (50% reliability) the Design Life Factor is derived. This factor is thus life at a given probability divided by life at 50% probability and it is plotted against percent reliability on a log-probability paper. Figs. 7-15 constitute these plots for various cases.

Reliability of Automotive Components

In this manner, considerable data obtained over a period of years, and also some found in literature was analyzed in the manner indicated above and plotted as shown in Figs. 7-15. In these graphs the upper and lower abscissas represent percent reliability and percent failures respectively. The ordinate gives Design Life Factors. The original data, from which these calculations were made were expressed as life in cycles, hours, miles, etc. A low design factor means high scatter and a high factor considerable uniformity in the life of the parts tested.

The stress or load levels corresponding to the above lives were the design stresses under which these parts were tested. In most cases these were located somewhat below the midpoint between the ultimate tensile strength and the endurance limit of the material. In those cases where lives were available at several stress levels the information was so noted and further analyzed.

The analysis of Figs. 7-13 suggests the following. Because of manufacturing variations one would expect more scatter in actual automotive parts than in laboratory specimens. This was found to be approximately the case, as shown in Fig. 14. Within the manufactured parts themselves (Figs. 7-11), those parts which are subjected in service to contact loading (ball bearings, hypoid and bevel gears, engine valves) were found to have lower life factor than parts under flexural loading (crankshafts, springs, fans, etc.) Fig. 15. Note that the transmission helical gears had a relatively high life factor (Fig. 7). In this case the critical load was flexural as all the failures occurred at the root of the tooth.

The wide scatter of members under contact loading has been generally recognized (7). In the case of ball bearings life is inversely proportional to the ninth power of the unit pressure, so that even a slight variation in the bearing geometry or in the applied load may produce considerable variation in life.

Finally, no particular interrelationship can be expected within the various automotive components (Figs. 7-11). The scatter in life and thus the value of the design life factor at any reliability level depends on many things: the complexity of design at the critical section, uniformity of material properties, method of manufacture, quality of the inspection methods, etc. and these will differ among the various automotive parts. Thus a shaft whose critical section is a keyway would be expected to show higher variability in life than a shaft whose critical section is a large fillet. The size and the method of grinding the fillet can be carefully controlled and therefore, the shaft should show a relatively good uniformity in life. On the other hand it is more difficult to control the manufacture of a keyway and more scatter can be expected. Thus, in Fig. 7 higher design life factor was obtained for

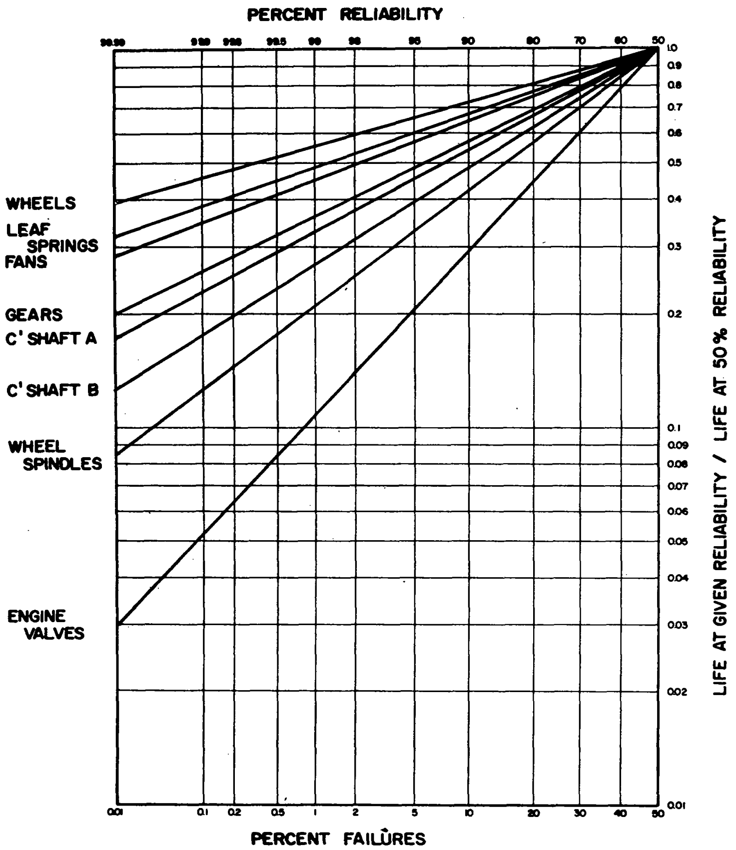

Fig. 7. Design Life Factors at Different Reliabilities - Some Automotive Components.

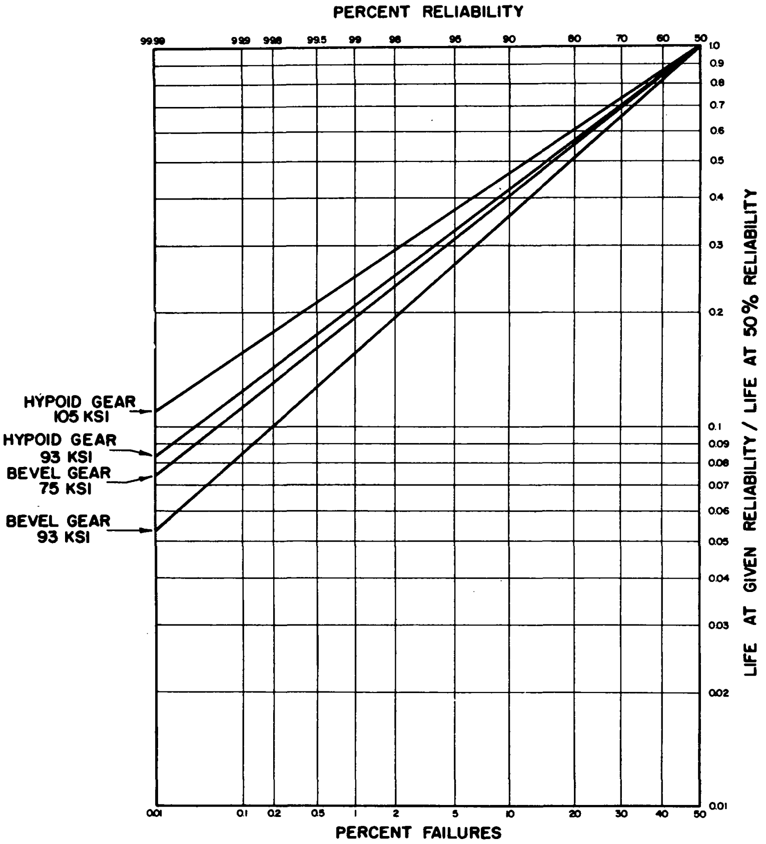

Fig. 8. Design Life Factors at Different Reliabilities - Bevel and Hypoid Gears

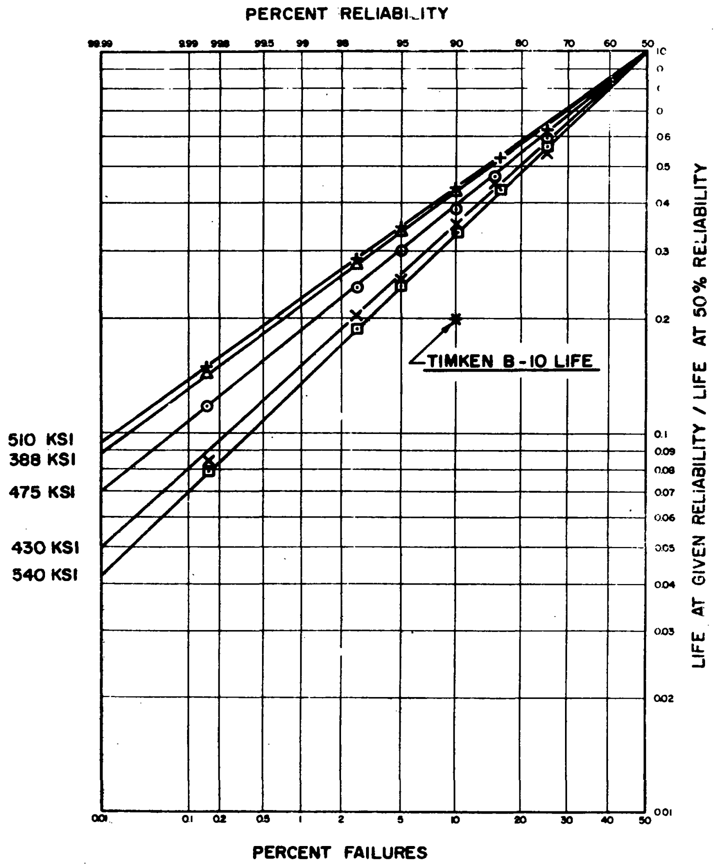

Fig. 9. Design Life Factors at Different Reliabilities - Radial Ball Bearings.

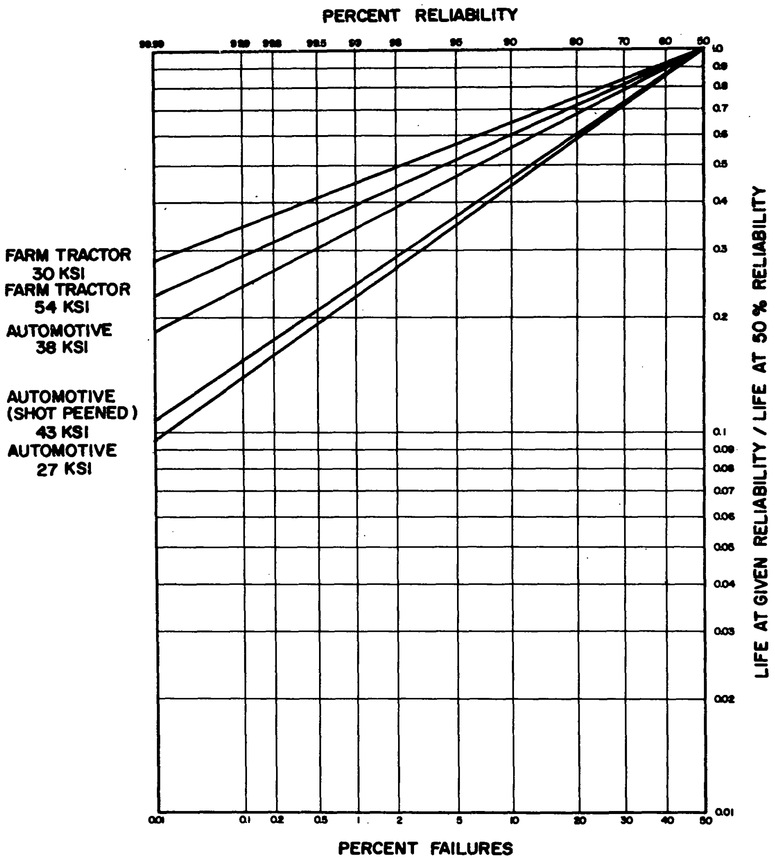

Fig. 10. Design Life Factors at Different Reliabilities - Rear Axle Shafts

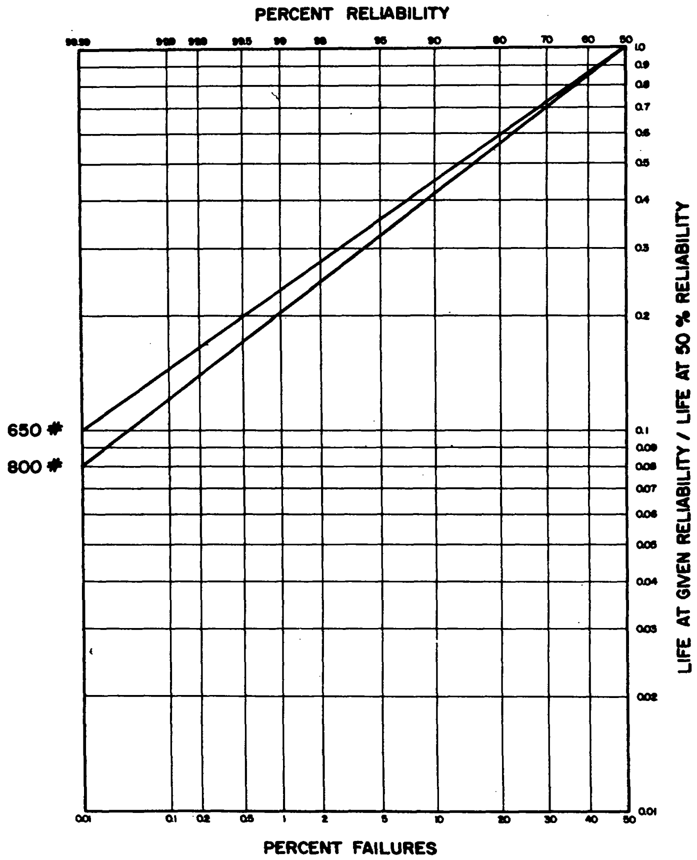

Fig. 11. Design Life Factors at Different Reliabilities - Roller Chains

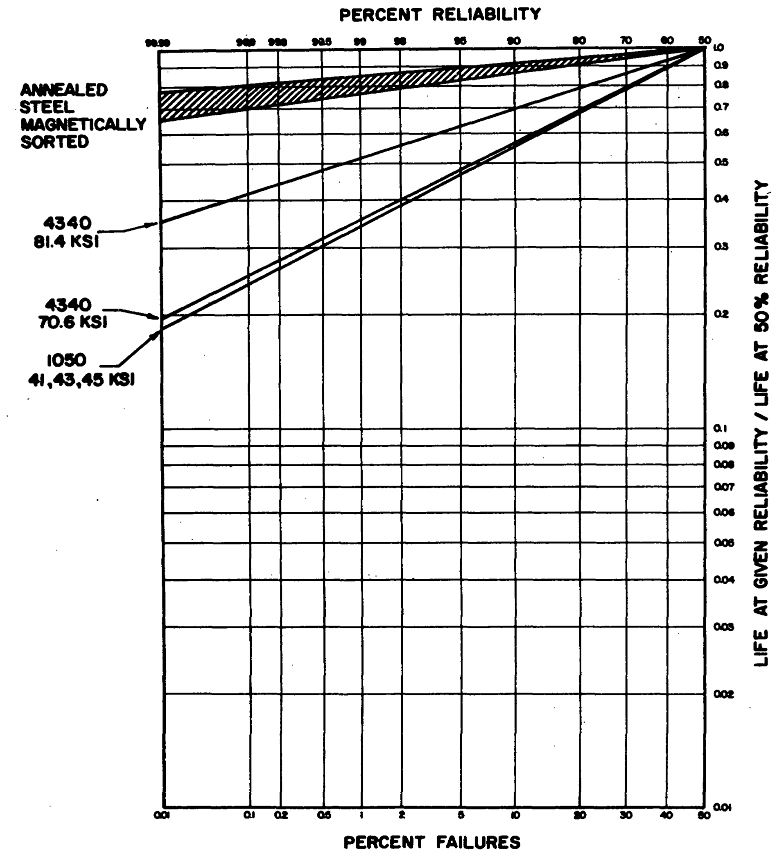

Fig. 12. Design Life Factors at Different Reliabilities - Some Steels

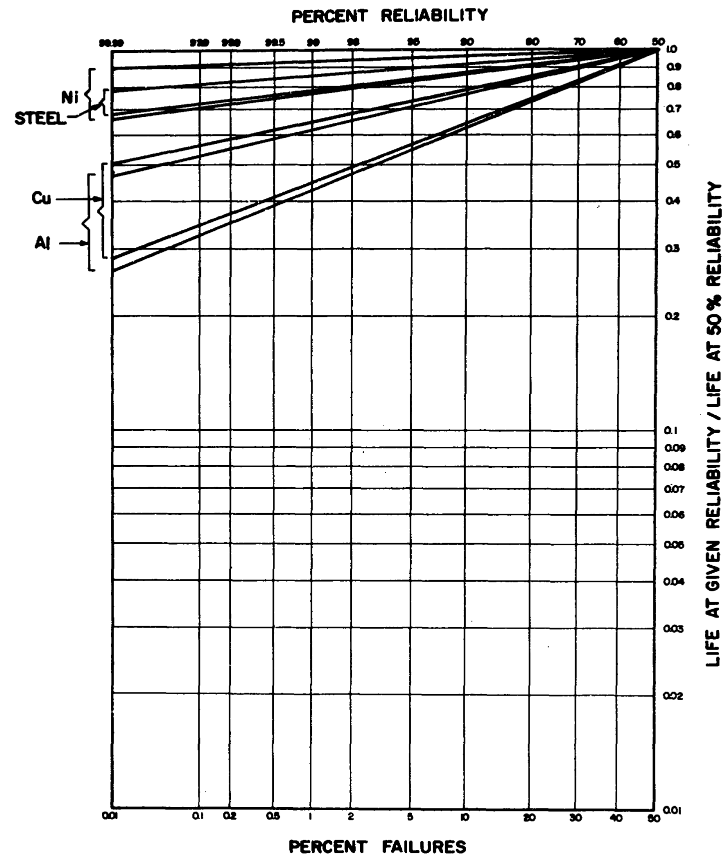

Fig. 13. Design Life Factors at Different Reliabilities - Annealed Steel and Nickel Magnetically Presorted.

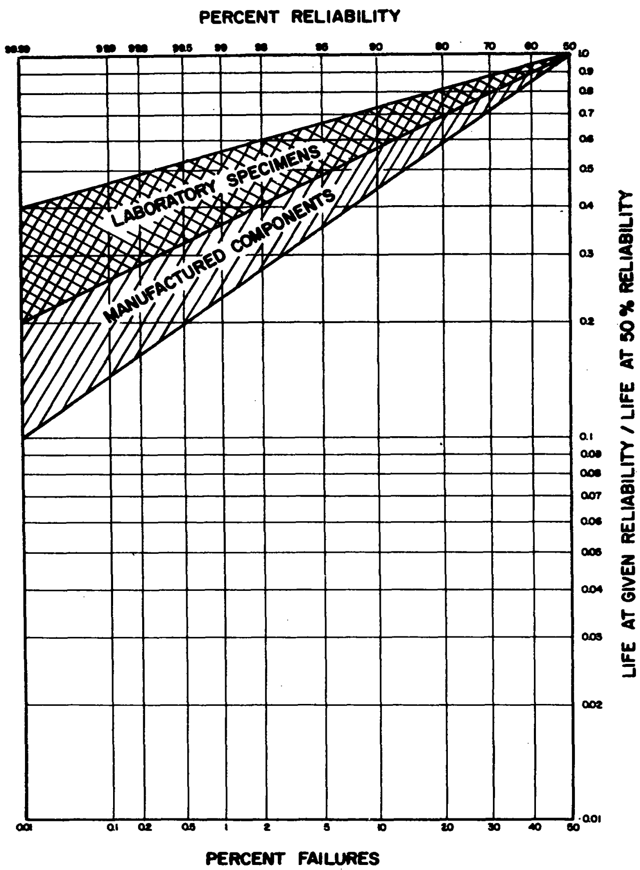

Fig. 14. Design Life Factors at Different Reliabilities - Comparison Between Laboratory Specimens and Manufactured Components

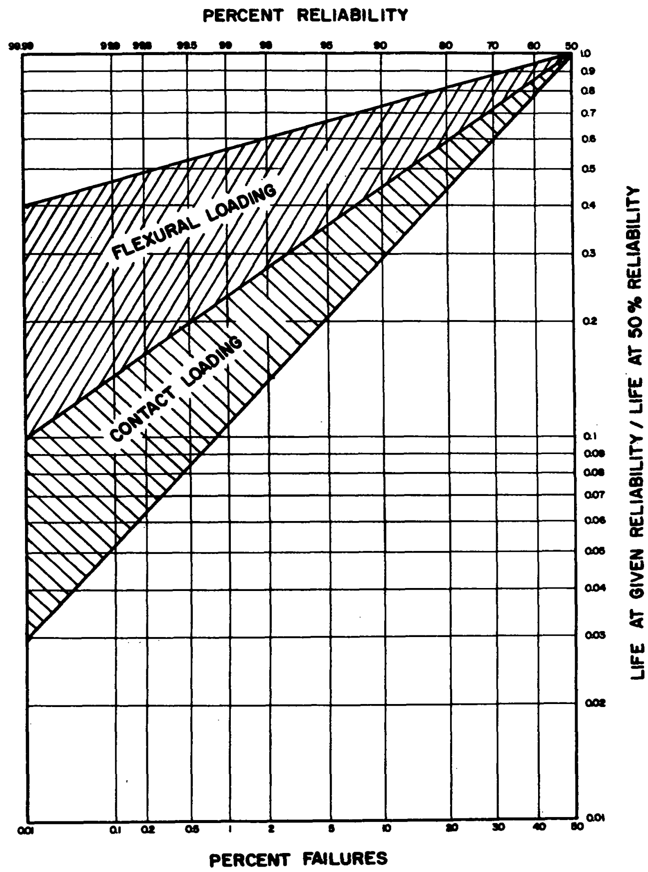

Fig. 15. Design Life Factors at Different Reliabilities - Comparison Between Parts Subjected to Flexural Loading and Contact Loading

a cast wheel, where in the region of maximum stress there was no abrupt changes in section than in wheel spindles which had an 1/8″ fillet radius.

For design purposes 95% reliability may be used in design calculations of parts whose failure can be regarded principally as an economic loss but not a loss of life of vehicle occupants. Into the first category belong body, engine, transmission, and some drive line components. For the latter (steering and suspension parts) a 99.9% reliability should be used instead. The information in Figs. 7-11 is tabulated under the two headings in Table 1.

Table 1. Design Life Factors
Automotive Parts

	Reliability	
	95%	99.9%
Wheel	.66	.45
Leaf Spring	.60	.38
Fan	.57	.35
Transmission Gear	.49	.26
Crankshaft - Design A	.46	.22
Crankshaft - Design B	.40	.18
Front Wheel Spindle	.33	.13
Engine Exhaust Valve	.20	.05
Hypoid Gear, 93 KSI	.33	.13
Hypoid Gear, 105 KSI	.37	.16
Bevel Gear, 75 KSI	.32	.11
Bevel Gear, 93 KSI	.27	.084
Ball Bearing, 388 KSI	.34	.13
Ball Bearing, 430 KSI	.26	.08
Ball Bearing, 475 KSI	.30	.11
Ball Bearing, 510 KSI	.34	.14
Ball Bearing, 540 KSI	.24	.07
Axle Shaft, Automobile, 27 KSI	.35	.14
Axle Shaft, Automobile, 38 KSI	.46	.24
Axle Shaft, Automobile, Shot Peened	.37	.16
Axle Shaft, Farm Tractor, 30 KSI	.57	.34
Axle Shaft, Farm Tractor, 54 KSI	.52	.29
Roller Chain, 650 lb	.35	.14
Roller Chain, 800 lb	.32	.12

The Design Stress Factors

For design purposes some problems are best solved in terms of the life at a given stress, while others in terms of stress at a given life. Figs. 7-13 and Table 1 provide the information for the former. That is, at any required level of reliability the design life as a fraction of the mean life can be determined. In this manner, the average life of automotive components, conventionally obtained from laboratory tests, can be translated into a more meaningful design life where reliability and thus the maximum percent of failure can be stated.

The other category of design problems calls for the solution of stresses at a given life. To do this it is first necessary to establish the variation in the life scatter with the stress level. A casual review of literature (8,9,10) indicates that in some cases scatter is independent of stress, while in other cases it may either slightly or pronouncedly increase or decrease with the stress level (Figs. 16-18). The same was found to be true in the present case.

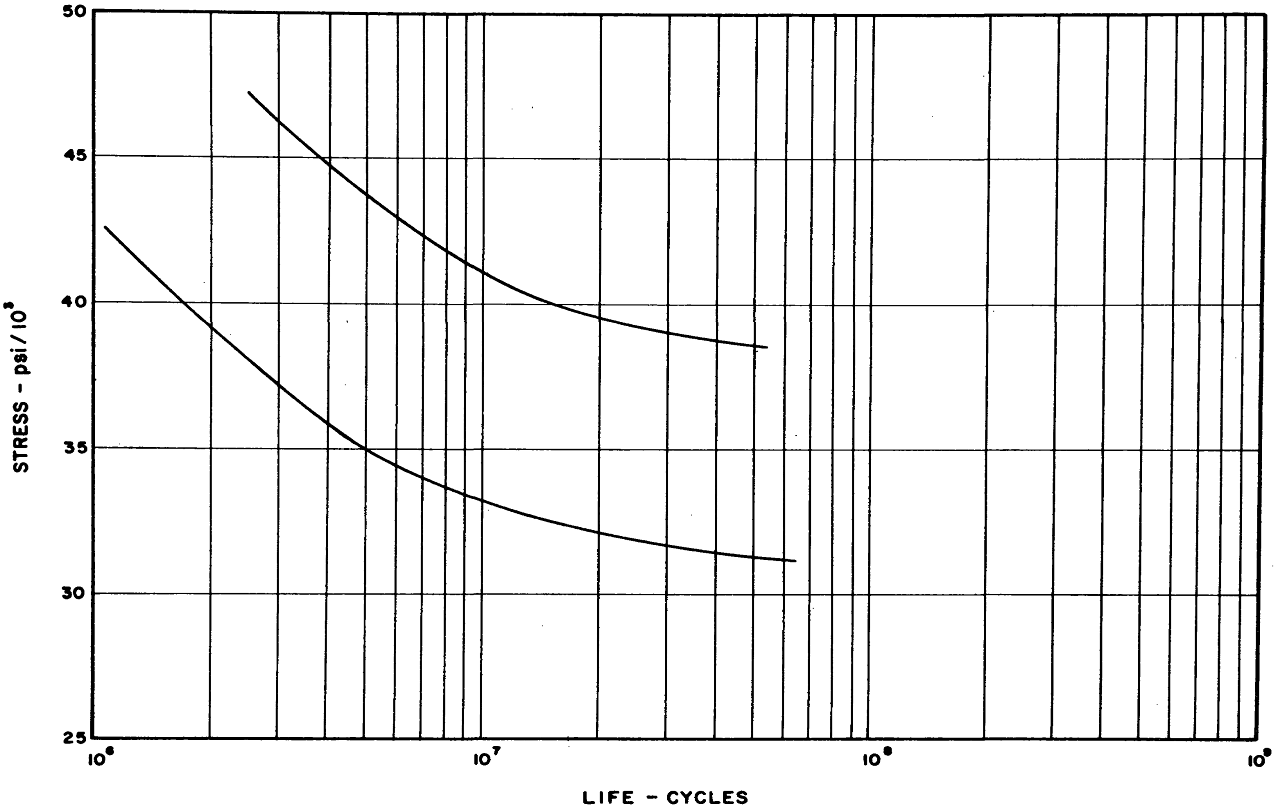

Fig. 16. Scatter Band of Monel Metal[8]

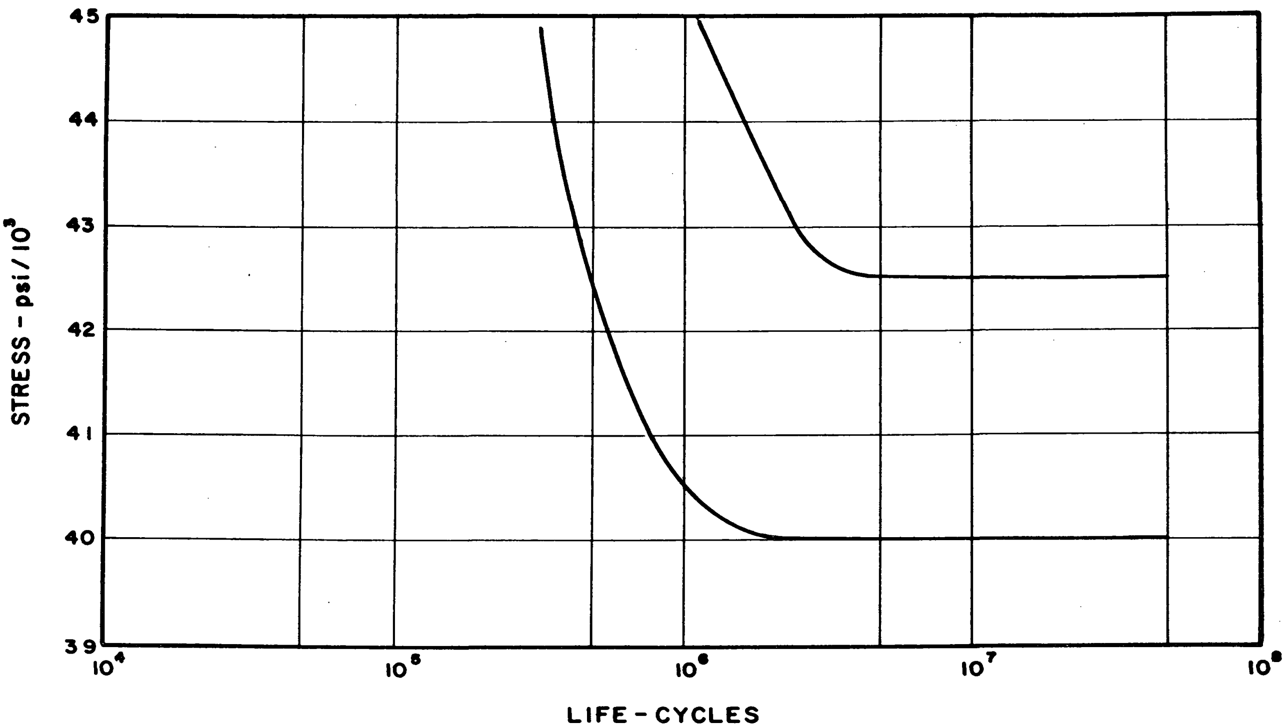

Fig. 17. Scatter Band of 1050 Steel[9]

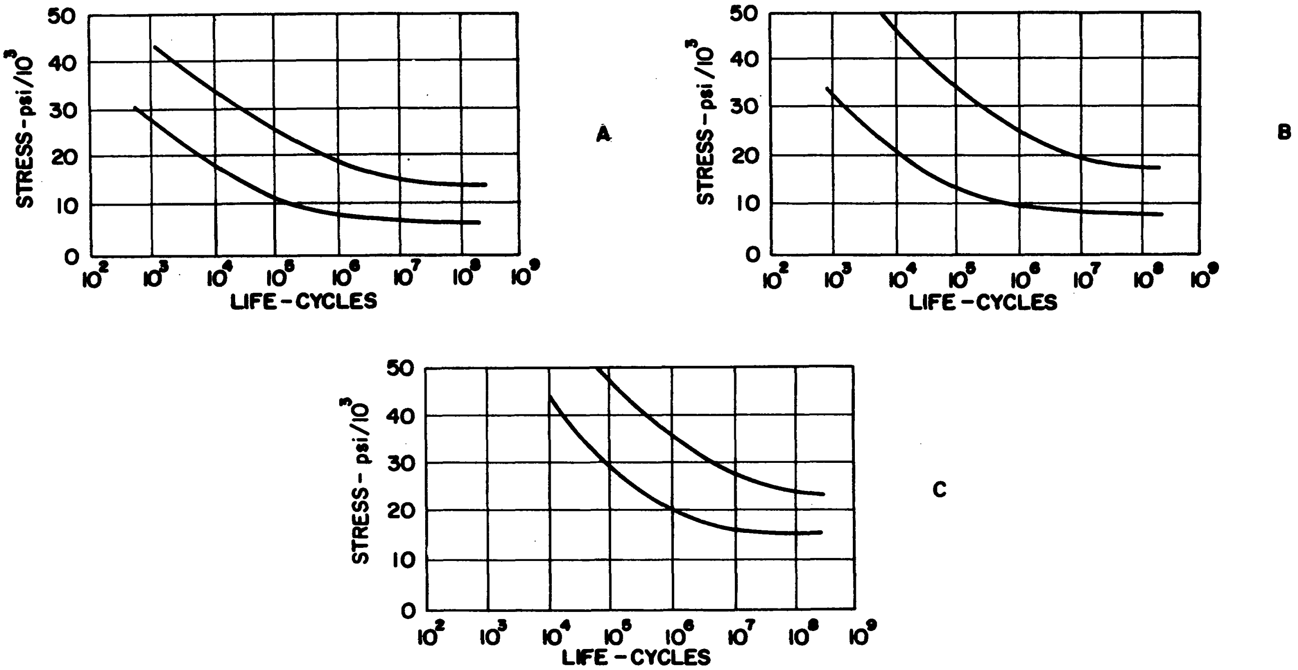

Fig. 18. Scatter Band of Aluminum Alloys -
A: Sand Cast; B: Permanent Mold; C: Wrought

It is significant to note (Figs. 12 and 13) that in the case of materials magnetically sorted for permeability and tested in torsion the design life factor: 1) is largely independent of stress: 2) it is much higher for any level of reliability than materials not sorted. This reflects the uniformity of samples obtained. The benefit of this uniformity to the service life of automotive components is worth noting.

In Figs. 19 and 20, by dividing at a given life, the stress at 99.86% reliability (3 standard deviations) by the stress at 50% reliability (mean stress) a Design Stress Factor $S_{99.86}/S_{50}$ was obtained. This was repeated for various materials, with results as shown in Fig. 21. It will be noted that for the softer materials studied, in order to assure ourselves of 99.86% reliability, that is less than 15 failures in 10,000 pieces, it is necessary to design the part to approximately 80% of the average value of the fatigue strength.

For the 4340 steel the percent reduction in the fatigue strength will be considerably greater. Thus, for a life of 500,000 cycles the design should be based on approximately 50% of the mean fatigue strength found in handbooks for this particular life.

Similar study was made for ball bearings, with the results shown in Fig. 23 which was derived from Fig. 22. The design stress here should be 80% of the mean fatigue strength.

The above analysis was concerned with the finite life regions (sloping lines) of the stress-life relationships. Calculations were also made for the endurance limit regions of several automotive components. Here, however, the data were less well defined. In some cases the distribution was found to be normal, in other cases log-normal, and in a few cases not at all defined.

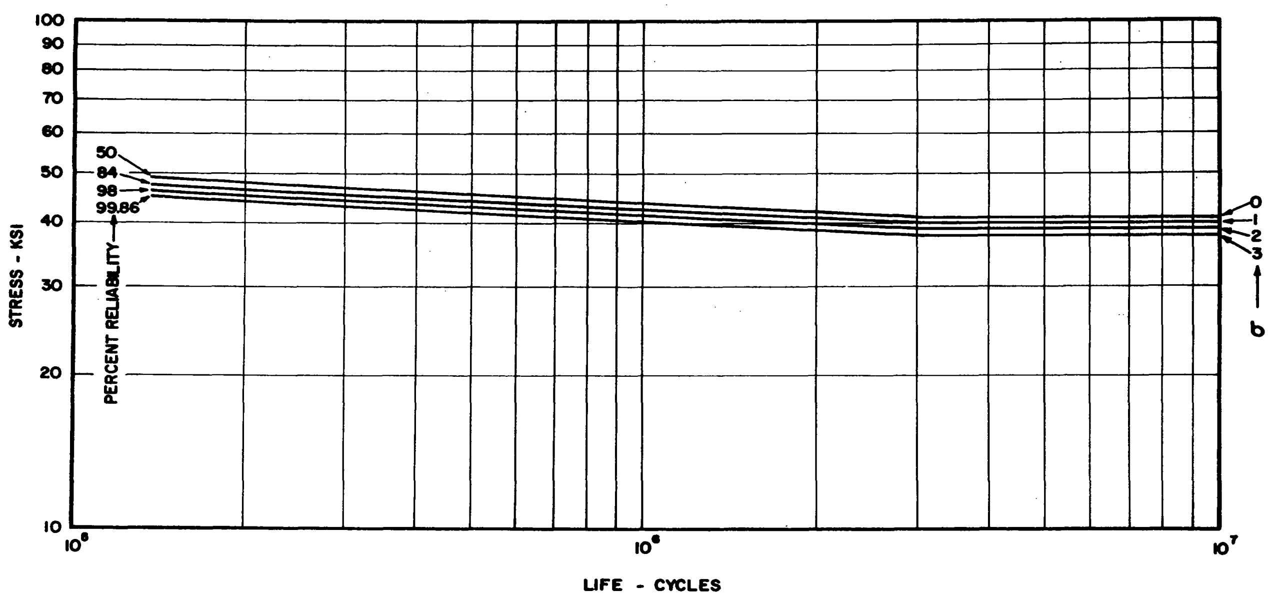

Fig. 19. Stress-Life Relationahip at Various Reliabilities - 1050 Steel, $81R_b$

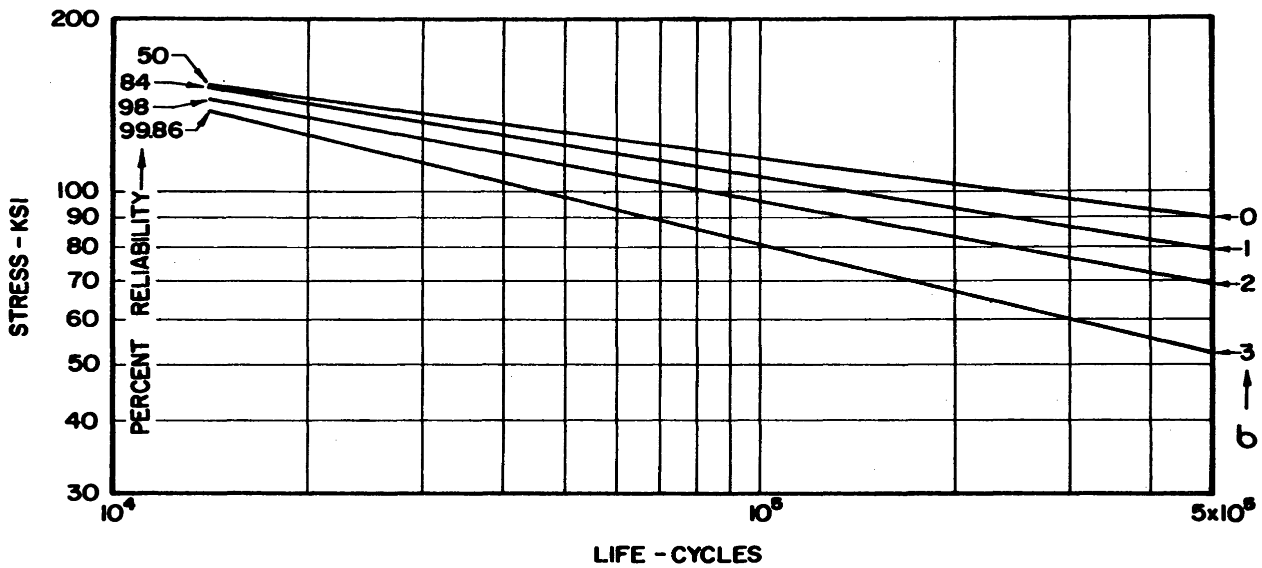

Fig. 20. Stress-Life Relationship at Various Reliabilities - 4340 Steel, 363 BHN

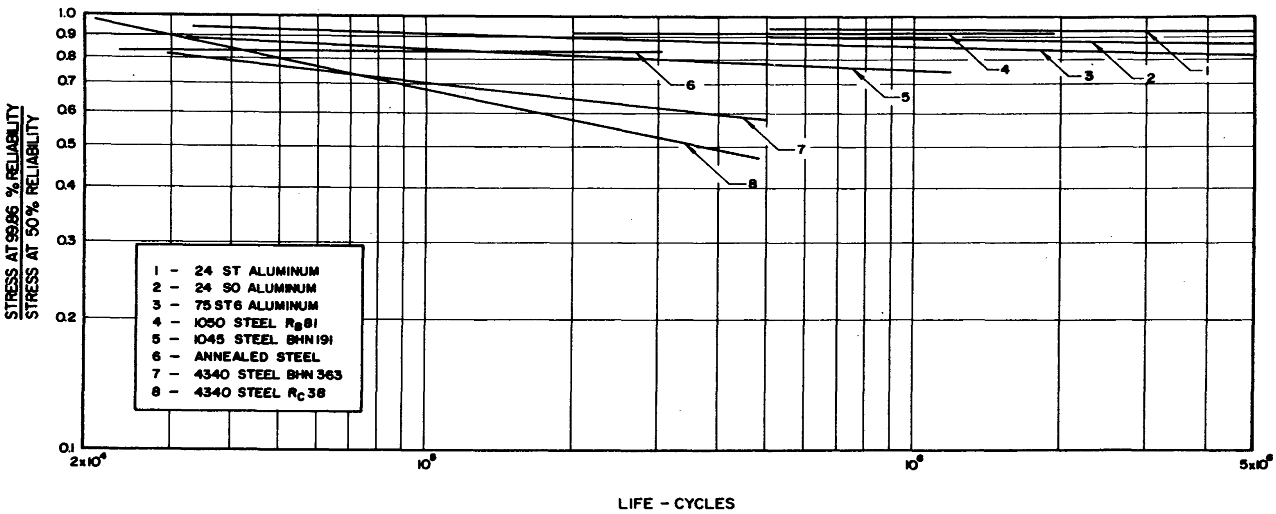

Fig. 21. Design Stress Factors at Various Lives - Some Materials

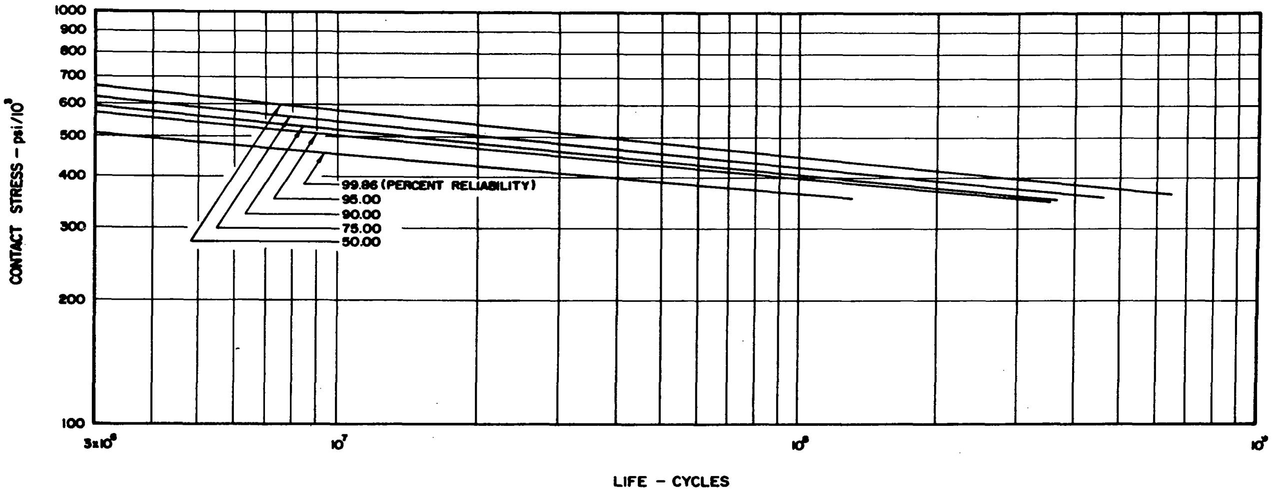

Fig. 22. Stress-Life Relationship at Various Reliabilities - Radial Ball Bearings

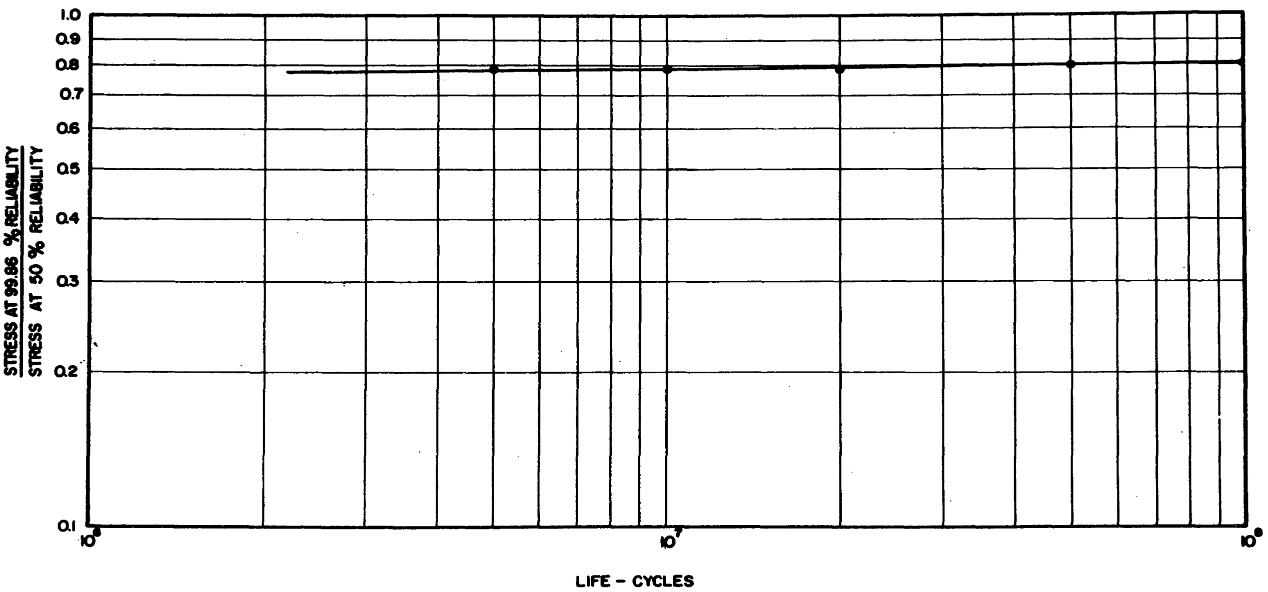

Fig. 23. Design Stress Factors at Various Lives - Radial Ball Bearings

Appendix

Some additional information on the characteristics of the automotive components studied:

Ball Bearings: 1/2″ radial, SAE 52100. Also see reference 15

Bevel and Hypoid Gears: See reference 3

Wheels: Truck, malleable iron

Leaf Springs: Truck, SAE 4068, 444-477 BHN

Gears: Truck, Transmission Helical, SAE 8620, carburized, 58-63R_c

Crankshaft: Automobile, SAE 1046, 228-269 BHN

Front Wheel Spindles: Automobile, SAE 1046, 248-293 BHN

Engine Valves: Truck, Exhaust Valves

Axle Shafts: Farm Tractor, SAE 8635, 269-321 BHN

Axle Shafts: Automobile, SAE 8650, 388-444 BHN. Also see reference 14

Roller Chains: American Standard No. 40

Fan: Automobile, 6 blade spider type, SAE 1020

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DISCUSSION

KECECIOGLU (U. of Arizona): I certainly would like to commend you for presenting such extremely valuable data. In the field of reliability everybody says all right, all this theory you are presenting is fine. Where is the data that substantiates it? Your paper presents such very, very valuable data and I hope you publish more of this type of data.

Now my question would be this: Have you, now that you have reliability versus design life data, instead of getting this data experimentally on manufactured pieces, tried to actually design this particular reliability into them, and then compare the reliability obtained from strength and stress distributions with the experimental results?

LIPSON: Yes...

KECECIOGLU: And then correlate this with the test data. How does it correlate?

LIPSON: Yes, I'll answer this briefly because Mr. West indicates to me that the purpose of discussion is discussion and not orations. So my answers will be real brief.

Yes, as a matter of fact what you suggested represents now a major effort in the automotive industry. Up to the present time the main application of reliability in the automotive industry was to the field of testing, Weibull plots, log normal plots, how many samples to choose, how to interpret data statistically and so on, but now we direct our effort in this direction that you indicated, and we employ the interference method, where we plot stress distribution which is generally normal distribution, and strength distribution which is normal distribution and overlap them by simple statistical methods, and determine the frequency of failure.

So first on the managerial level, one decides what is the percent of failure that can be economically tolerated. They may say in the case of a crankshaft, we can tolerate 1/10th of 1% failure, that is, one failure in a thousand. Armed with this, we go back and determine what should be the relationship between stress and strength on the basis of the interference method. But in this case may I indicate to you that the stresses we are talking about and the strength we are talking about are not just stress index or some nominal stress or some index that we hopefully think is a measure of stress.

It must be the real stress. What we call the significant stress — considering stress concentration and all the related factors. When we talk about strength, we mean significant strength which includes the effect of size, type of loading, mode of loading, surface finish, stress concentration and environment and so on.

That's why in the field of reliability, what we broadly call stress analysis, actually it should be called stress and strength analysis, plays such a very important role.

HUGHES (Allis-Chalmers): I see that you have data on parts under flexural and contact stresses. Do you have any data on strictly wearing or rubbing type parts? Parts that are not under any significant flexural stress or axial load, but wear because of the mating parts?

LIPSON: Yes, we do have some data, for example we have some data on the wear of cylinder liners, and the wear of pistons, piston rings, and of gears. We generally assume that they follow a normal distribution rather than a log normal distribution and we treat it accordingly. As a matter of fact we consider this problem so serious, so important that at the University we have a course devoted to just that particular problem, on the graduate level, because I strongly feel that — take for example gears — the design of gears in a few years will be predicted neither on Lewis formula or flexure loading at the root of the tooth, but on the limitations of pitting and wear on the pitch line, and the same is true of many other components. So this would be the real criteria on the design.

BAZOVSKY (Raytheon): Professor Lipson, you say that most of the distribution shown on the slides are log normal. Since the failures are probably caused by composite effects such as fatigue, premature wear and sudden breaks caused by some imperfection in the material, we can say it is a combination of chance failures and wear type failures or fatigue failures. If one considers the chance failures as exponentially distributed and the fatigue and wear type failures as normally distributed then the combination of such could produce a log normal distribution, wouldn't you say?

LIPSON: Yes, and they do follow log normal distribution. We are faced with this problem in the automotive industry and I'm sure it is true of any other industry that we have a choice of plotting the data as a log normal distribution and obtain a straight line the way I have plotted it right here or as a Weibull distribution, and Mondays, Wednesdays and Fridays the plot is Weibull, the rest of the week the plot is log normal.

BAZOVSKY: The Weibull distribution depends on how the alphas and betas, the constants in the Weibull distribution, are located. I would say that in most cases in the Weibull distribution you will find that the distribution applies if there is a very high incidence of early failure or infantile failures. When those are weeded out the remaining specimens will have then a more reasonable life, I would say. As to your paper it gives us the answer how to design mechanical parts for a required life and machines too. It supplies directly the answer to what to do and in which direction to move in the future.

LIPSON: Thank you. I would like to comment on your observation about obtaining a straight line, either Weibull or log normal. Really what we are interested in is this low percent of failure; the B 10 life or the B 5 life, because nobody can tolerate 10% failure, consequently we are not interested in B 50 life or even B 10 life. B 10 life, 10% failure will be catastrophic. We are interested in B 1 life. Consequently we are particularly interested in the shape of the curve, Weibull or log normal at low percentages, let's say B 1. Unfortunately Weibull doesn't follow a straight line at low life, neither does log normal.

In most cases if this is the Weibull curve, it drifts downwards, and this simply means that if you do follow the Weibull curve as a straight line at low life, you are always on the conservative side, but this is no comfort at all, because you can be very grossly overdesigned on this basis. Consequently we are particularly interested in why actually it deviates from this straight line. We feel that probably the reason for it is that the failures that we test, the parts that we test or the parts which are tested in the hands of the customer are not really random failures. They do not all belong to the same population. By this I mean that during the process of manufacture by one method or another, there is some inspection; visual inspection, electromagnetic inspection, and so on.

This automatically truncates the low end of the curve, and really what we plot is not normally distributed on log normal distributive curve, but the truncated curve. If we plotted simply parts as they come from production without any rejects by inspection I suspect we would get a straight line.

JACKSON (Dept. of Defense): We talked about obtaining field data. How do you get your field data? What is your source of field data and the means by which it is collected?

LIPSON: Up to about three years ago it was a very haphazard means of obtaining data, but now in connection with reliability programs instituted in all automotive companies, a well organized effort is being made.

The main source of field data is of course the field data obtained during the warranty period. So if there is a warranty period say of 24,000 miles and you go to your dealer, and simply say that something has failed, and he fixes this, of course he is compensated by the company. So he is very careful to write a report and send the broken parts. So in this manner we now have a very careful source of data.

I would like to emphasize again this data is restricted only within the warranty period, within 24,000 miles, and the rest of it we have to project what is likely to be the failures at 50,000 miles.

THAYER (Lykes Bros.): In order to have this extended warranty do you have to go to the dealer for certain progressive maintenance checks?

LIPSON: No. To the best of my knowledge, you don't.

THAYER: I wonder about some cars where they say this warranty, this extended guarantee, is only applicable if you follow a certain maintenance procedure.

LIPSON: I think Chrysler has that. Chrysler with a 50,000 mile warranty on the power drive, on the drive train...

THAYER: Isn't this in effect forced progressive maintenance which might help to cut out these random failures during this extended guarantee period?

LIPSON: Yes, this is a very well taken point. They modified the normal distribution curve that we get in this manner. That's true.

FRANKEL (M.I.T.): I think we are very indebted to Professor Lipson for more or less providing the evidence that mechanical failure rates are very different from electronic failure rates and will normally obey the Weibull distributive or log-normal distribution; and, therefore, that mechanical failure rates are time dependent.

I would like to ask Professor Lipson if his statement on wear-failure and his assumptions that this is still a constant failure rate and could probably be simulated by some process where failure distribution is normal — is not unlike dry friction of two services, but related events of different interacting components. I believe some very sketchy experimental data shows that this also would approach a Weibull distribution. I wonder if you would comment?

LIPSON: Well, as a matter of fact actually I can comment on it in this manner. In the first place when we plotted for example our data on the cylinder linear wear, the purpose of this investigation was to determine the effectiveness of induction hardening the bore in minimizing wear of cylinder bores. So we have considerable amount of data on non-induction hardened bores and induction hardened bores and we plotted this. When we did plot this it followed Weibull curves as you indicated.

My second comment will be that we are painfully aware of the various manifestations and complexity of wear aside from the fact that there is abrasive wear, and adhesive wear — the mechanisms are quite different. So the problem is most complicated, but there seems to be an indication that they follow Weibull.

DESIGN OF EQUIPMENT TO OPTIMIZE RELIABILITY FOR MANUFACTURER'S AND CUSTOMER'S MINIMUM TOTAL COST *

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ABSTRACT

For a class of pumps used commercially and aboard aircraft carriers, there exists an optimum level of reliability at which the total cost of this pump to the user is minimum. The reliability of these pumps is predicted and also calculated from field performance data. Actual manufacturer's and customer's cost data are developed, consisting of the manufacturer's cost before and after shipment and his profit, and the customer's cost of buying and maintaining these pumps including downtime cost. From a combined plot of this data, the optimum level of reliability to which the pump should be designed and manufactured is determined for both the manufacturer and the customer. A customer's purchasing philosophy should consider purchasing a product at this optimum level of reliability. Consequently the purchase should be neither at low initial engineering and manufacturing cost, which generally correspond to low reliability and high support cost; nor at very high reliability which corresponds to very high initial cost. In both cases the total cost of the product to the customer is higher than that at optimum reliability, assuming that the total monetary outlay over the life of the product is the major consideration.

INTRODUCTION - PRODUCT PROCUREMENT PHILOSOPHY

Generally, two basic philosophies have been used by the purchaser of products heretofore. One is to buy at the lowest selling price and the other is to buy the product at the highest possible reliability level. Neither philosophy is correct, because they both lead to a higher total cost of the product over its useful life. We believe that total monetary outlay over the life of the equipment by the user is the primary criterion to be used in the procurement or specification of a product, rather than purchase price alone.

Minimizing this total cost maximizes the utilization of available money and enables the acquisition of more units for the same money. For a given number of required units it minimizes the monetary burden on the taxpayer who after all is the source of our defense budget.

This study establishes the fact that there is such a level of reliability at which

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the total cost over the life of a specific product is minimum. This level of reliability is called the *optimum level of reliability*.

The objective of this paper is to develop the relationship between reliability and the manufacturer's and customer's total cost. This is shown first qualitatively and then quantitatively for one product, a pump used by the Navy. It is hoped that proving the existence of such an *optimum level of reliability* shall stimulate a serious reappraisal of manufacturers' design, manufacturing and purchasing philosophies and practices, and of customers' procurement philosophies.

The philosophy should not be that minimum purchase price nor maximum attainable reliability governs the procurement decision but that the *optimum level of reliability* does. Furthermore, procurement should be from that manufacturer who supplies a product whose *total cost* is minimum rather than initial purchase price. If this criterion is met by more than one manufacturer then the successful bidder should be selected on the basis of additional factors such as maintainability, availability, delivery time, etc.

It should be emphasized that the determination of this total minimum cost requires knowledge of the reliability of the product involved and explicitly its "bathtub" curve.

BACKGROUND – PRIOR STUDIES

Several investigators have developed the concept that an optimum reliability level does exist for certain cost considerations. We feel that Figs. 1 and 2 are the first attempts at developing a complete cost picture. Ryerson (1)* and RCA (2) developed the optimum reliability point for the minimum customer's total cost by adding together two reliability versus cost curves. One curve which increases with reliability represents the initial cost to the customer. The other curve which decreases with increasing reliability represents the support cost. The sum of these curves is similar to Curve C, Fig. 2 designated as "Customer's Total Cost." These two studies do not consider the manufacturer's total cost picture and no actual cost data in support of these curves are provided.

The existence of an optimum quality for the manufacturer has been pointed out by Delco-Remy (3). This was done by considering the decreasing cost of scrap, reoperation, and warranty as quality increases, plus the increasing cost of a program to achieve increasing levels of quality. The result is similar to the "Manufacturer's Total Cost" shown in Fig. 1, Curve C. This study does not consider the customer's total cost picture. We believe that reliability versus cost results in a more useful comparison than quality versus cost because reliability is a precisely defined scientific entity and includes the effect of variable quality.

Other work done by RCA (2), Delco-Remy (3), Winlund (4), Welker and Bradley (5), and Bosinoff (6), has resulted in the enumeration of some of the costs that must be considered to optimize reliability from both the manufacturer's and customer's point of view.

Miller (7), Carhart and Herd (8), Niles (9), Gyllenhaal and Robinson (10), Kalbach (11), Price (12), and Cox and Harter (13) have developed mathematical reliability versus cost models, however, the major obstacle faced in applying these has been the lack of adequate data. One of the objectives of this study is to develop the required data for a specific product and motivate the acquisition of data for cost optimization with regard to reliability in general.

Recently the philosophy that there is an optimum product reliability at which its total cost is a minimum has been extended to the possibilities of lowering this minimum cost and at the same time increasing the product reliability through the

*Numbers in parentheses refer to those in the Bibliography.

use of an *integrated, aggressive and scientifically applied reliability program* (1,2, 14). This is a very promising extension and should be the subject of other studies.

To date very few studies have been conducted wherein reliability versus manufacturer's and customer's cost curves were developed based on actual reliability and cost data. Cox and Harter (13) have developed curves between the reliability improvement effort and the reliability resulting from this effort for one shot devices. Stevens (15) has demonstrated that reliability improvement costs for electronic subsystems for military aircraft are far offset by maintenance cost savings based on actual cost figures. McLaughlin and Voegthen (16) have determined the support cost of special purpose and waveguide tubes in prime search radar and navigation aid and UHF communication at four Air Control and Warning sites located within the Central Air Defense Force Command. Forty percent of the maintenance hours were found to be required to maintain these tubes. It is concluded that the development of improved parts are most urgently needed to reduce the heavy burden of support and that the present practice of considering only unit purchase price of equipment is misleading. We must determine means of basing development and production decisions on the total cost of equipment for service life.

Bazovsky, MacFarlane and Wunderman (17) present an excellent mathematical analysis of the relationship between reliability and maintenance costs. They developed generalized mathematical linear and non-linear cost models for single components and also for complex equipment. They calculate the failure rates, the number of unscheduled and preventive maintenance actions, part wearout life, time between failures, maintenance time, part replacements, and costs of repair maintenance, preventive maintenance and the resulting total maintenance cost. In doing this they establish optimum reliability levels to minimize the total maintenance cost of a steam turbo-pump aboard a BuShips surface ship. They feel handicapped in applying their theory in its totality because of lack of sufficient high confidence level operating, reliability, maintenance and cost data.

Bracha (18) presents a comprehensive survey of papers and studies on the effect of unreliability on the support and maintenance costs. The survey shows clearly that such costs can be as high as twelve times the purchase price per year of the original equipment. He emphasizes that attainment of optimum reliabilities is paramount to save the taxpayer money on space and defense projects. He urges that short-range procurement policies should be replaced by long-range planning which considers the total system cost. This cost would include the cost of developing, procuring, and maintaining the entire system in which the equipment is operating. When studies show that the total cost of the system during its useful life would be less by spending more on the initial achievement of reliability and maintainability then the greater initial expenditure is worthwhile.

This sutdy is one of the first major efforts in this field, whereby actual reliability and cost data are used to draw conclusions on the reliability versus total-cost picture of a non-electric product, a Navy centrifugal pump. It is hoped that the investigators quoted previously may welcome the results presented and use the reliability-cost data to further substantiate their theories and findings on reliability-cost optimization.

MANUFACTURER'S RELIABILITY VERSUS TOTAL PRODUCT COST PICTURE

The manufacturer's costs may be simply divided into two major categories:

1. Cost before shipment.
2. Cost after shipment.

EFFECT OF LEVEL OF RELIABILITY ON THE MANUFACTURER'S TOTAL PRODUCT COST

R_{OM} = Optimum reliability for minimum manufacturer's selling price.

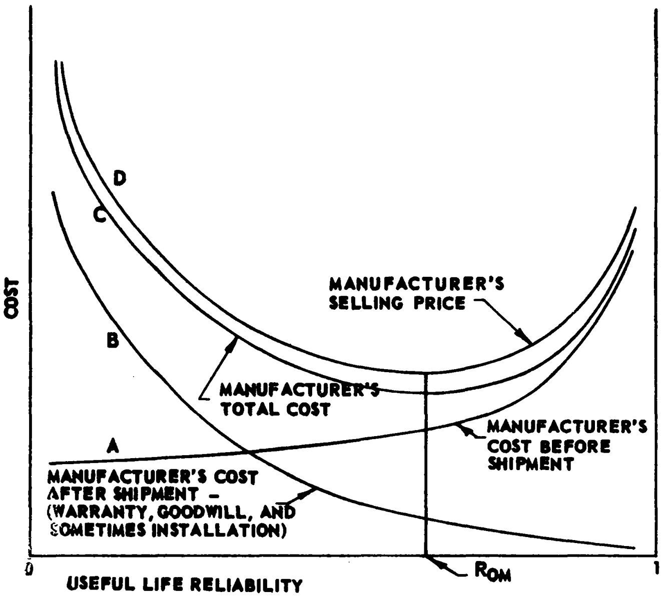


Figure 1.

The cost for a pump before shipment consists of the following:

1. Direct materials and purchased components.
2. Direct labor.
3. Manufacturing burden.
4. Research and development.
5. Engineering expense.
6. Engineering changes during manufacturing.
7. New patterns.
8. Small tools.
9. Shipping expense - packing.
10. Sales.
11. General and administrative.
12. Miscellaneous.

The cost after shipment consists of the following:

1. Warranty.
2. Good will.

As the reliability of a product is increased, the before shipment cost would also increase as shown in Curve A of Fig. 1. It is clear that higher reliability requires the use of better materials, slightly higher cost, more reliable purchased components, the expenditure of more research, development and engineering money. The same would hold for the remaining items, as more stringent controls, better tools, improved packaging, more surveillance and data feedback from the field and the customer would generally be required. These costs would increase slowly with increasing reliability but increase sharply as relatively high reliability levels are sought, particularly when improvement in the state of the art is involved.

The after shipment costs consist of the cost of (a) parts, subsystems and systems for inwarranty failures, (b) service shop charges, (c) travel expense, and

(d) other service efforts. These costs are frequently the result of inadequate or improper engineering, manufacturing, sales, purchasing, materials, quality control, inspection or service. In addition, misapplication, incorrect specifications, improper shipping practices, improper erection and startup procedures contribute to warranty costs.

Good will costs are incurred when the responsibility for a failure, malfunction or discrepancy cannot be ascribed clearly to either the producer or the customer and the producer absorbs part or all of the cost resulting from any corrective action.

As the reliability of a product is increased, failures decrease and so would the parts replacement cost, secondary failure cost, and the cost of the rest of the warranty items. At low levels of reliability these costs decrease sharply with increasing reliability and decrease gradually at high levels of reliability, thus giving rise to Curve B in Fig. 1.

The sum of Curves A and B gives the manufacturer's total cost, Curve C in Fig. 1. To this a profit is added and, assuming a given percent profit, Curve D, the Manufacturer's Selling Price, would be obtained.

A very important observation may be made for Curve D where a minimum selling price at a specific level of reliability, R_{OM}, exists. This level of reliability is known as the optimum level of reliability for the manufacturer. It is interesting to note that any deviation in either direction from this optimum reliability level results in increased cost to the manufacturer. A lower level of reliability results in higher costs because of higher after shipment costs, whereas a higher level results again in higher cost because of higher before shipment costs.

CUSTOMER'S RELIABILITY VERSUS TOTAL PRODUCT COST PICTURE

The customer's cost consists of his purchase price and the cost of operation and maintenance during the life of the product. The latter cost is composed of the following:

1. Maintenance materials and supplies.
2. Maintenance labor.
3. Parts replacement not covered by warranty.
4. Downtime due to malfunction and discrepant performance.

Generally, as the designed-in reliability of a product increases, the customer cost decreases because of a lower failure rate. Consequently, the fewer failures require less replacement parts and maintenance materials and supplies. This will produce Curve A in Fig. 2 which represents the cost of unscheduled repairs.

The customer could also have scheduled repair costs. These costs can increase or decrease with increasing product reliability as shown in the following discussion. It is assumed that the period between scheduled repairs remains constant. The scheduled repair cost will decrease if fewer parts are required as the product reliability increases. This case, shown by Curve B, Fig. 2, could occur if the parts lives were increased greatly when the product reliability is increased. The second case occurs if approximately the same parts are always replaced during scheduled repairs but the parts costs increase as the product reliability is increased. No curve is shown for this case.

For practical purposes, the operating overhead, installation and procurement costs may be considered independent of reliability. Consequently, they do not affect the trend of the customer's support cost but only shift the level of Curve C upward.

The customer's total costs will obviously be the sum of the initial purchase price plus the support cost over the life of the equipment. Curve C in Fig. 2 is

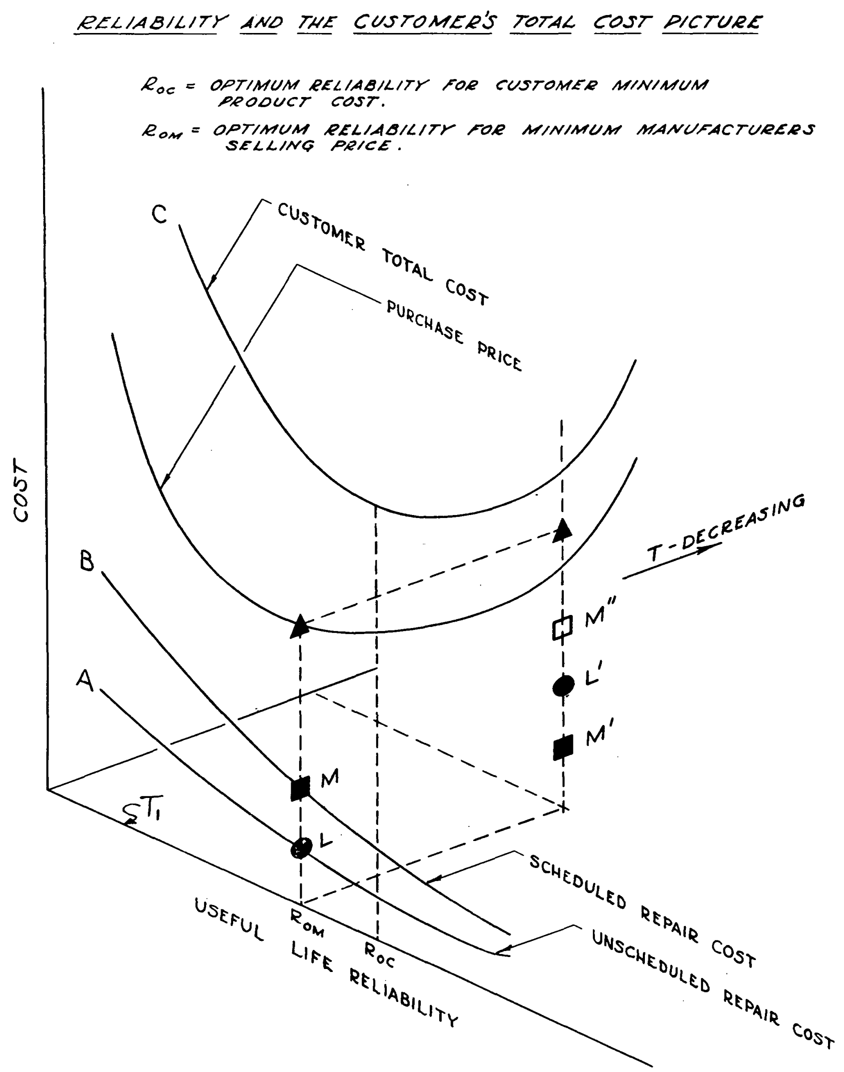


Figure 2.

obtained by summing Curves A, B and the initial purchase price curve. It is important to observe that Curve C provides an optimum level of reliability, R_{OC}, to the customer at which the total cost of the product to him is minimum. Furthermore, Curve C shows that any shift of useful life reliability from the optimum level increases the total cost. A shift to lower reliability results in higher support costs as well as higher purchase price, and a shift to a higher level of reliability will result in an increase in purchase price greater than the reduction of support costs. It can also be seen from Fig. 2 that the total cost will change, depending upon T, the period between scheduled repairs. Therefore, to truly optimize the total costs, both the "designed-in" reliability and the preventive maintenance program must be considered.

It is obvious that there is a shift in the optimum reliability level from that of the manufacturer's to that of the customer's. The competitive and profit picture would influence the decision as to which of these two optimum levels of reliability should be achieved by the manufacturer. It is also quite clear that every product should be designed, built, operated and maintained to obtain this optimum range of reliability. Considerable effort is required to reach this optimum and even when obtained, it will remain a never ending challenge to stay within this range of reliability.

Repair costs are a function of both random and wearout failure rates (22). Also, the scheduled repair cost is a function of four variables.

1. The frequency of the scheduled repairs.
2. The number of parts replaced during each repair.
3. The cost of the parts.
4. The time to make the repair.

The operational reliability of a product which exhibits wearout is dependent upon the interval between scheduled repairs or preventive maintenance actions noted as T. As T is increased more and more unscheduled part replacements will be made due to part wearout and equipment reliability will decrease. The limiting point occurs with no preventive maintenance. The failure rate of the product is then the reciprocal of the product's mean life plus the chance failure rate if all parts are replaced as they fail which gives a mix of part ages. As preventive maintenance actions are performed at more frequent intervals the part wearouts become fewer until only random failures are occurring. Preventive maintenance does not change the random failure rate. Therefore, the reliability of a product with a designed-in failure rate of λ_c can have a range of operational failure rates from λ_c, with a frequent preventive maintenance action, to the wearout failure rate of the product, $\lambda_c + \lambda_w$, which will occur with no preventive maintenance.

Fig. 2 shows in three dimensions the relationship between customer cost, useful life reliability and the period between preventive maintenance actions, T.

As T is decreased, the repair maintenance cost decreases because there will be fewer wearouts. But the preventive maintenance cost may increase or decrease, depending upon whether or not fewer parts and less labor are required with the more frequent preventive maintenance actions. These possible changes in costs are shown in Fig. 2 by points L and L'; and M and M' or M''.

Since the objective of this study is to consider the effect of the designed-in or useful life reliability of a product on its total cost the period between preventive maintenance should be maintained constant at the valve T_1 prescribed by the manufacturer. Therefore, the cost-useful life reliability plane at $T = T_1$ in Fig. 2 is the one of interest.

PRODUCT USED TO DETERMINE THE OPTIMUM RELIABILITY VERSUS THE TOTAL COST PICTURE

To optimize reliability with respect to total cost for a product, a product had to be selected that over its production history had undergone several changes that affected its level of reliability. These changes may consist of the following:

1. Design.
2. Material.
3. Purchased components.
4. Manufacturing techniques and processes.
5. Tools and skills.

After a careful investigation it was decided that a product that met the above requirements, was relatively simple in nature and also was used by the Navy was a centrifugal pump. From among the various types and sizes supplied to the Navy, Allis-Chalmers 5" x 4" KSK and SK pumps, illustrated in Figs. 3 and 4 were chosen. These pumps have horizontally split casing and a double suction impeller with a specific speed of approximately 1000. The Type SK pump is the basic model dating back to 1933 and the Type KSK is a redesigned model dating back to 1955.

For adequacy of design, cost and sales data, it was decided to confine the study to those pumps sold during the period of 1953 through May 1962.

Design and material changes made in these pumps during this period are given in Table I.

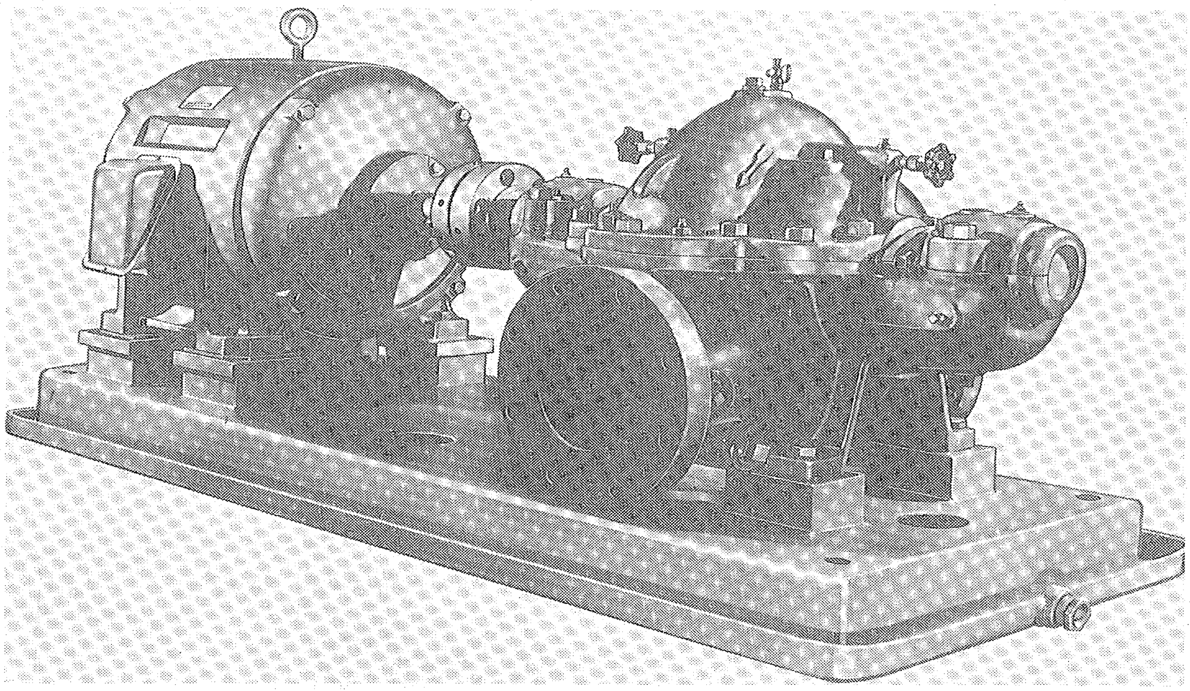


Figure 3. External View of 5x4 SK Pump Shown Driven by a Motor, Both Mounted on a Bedplate.

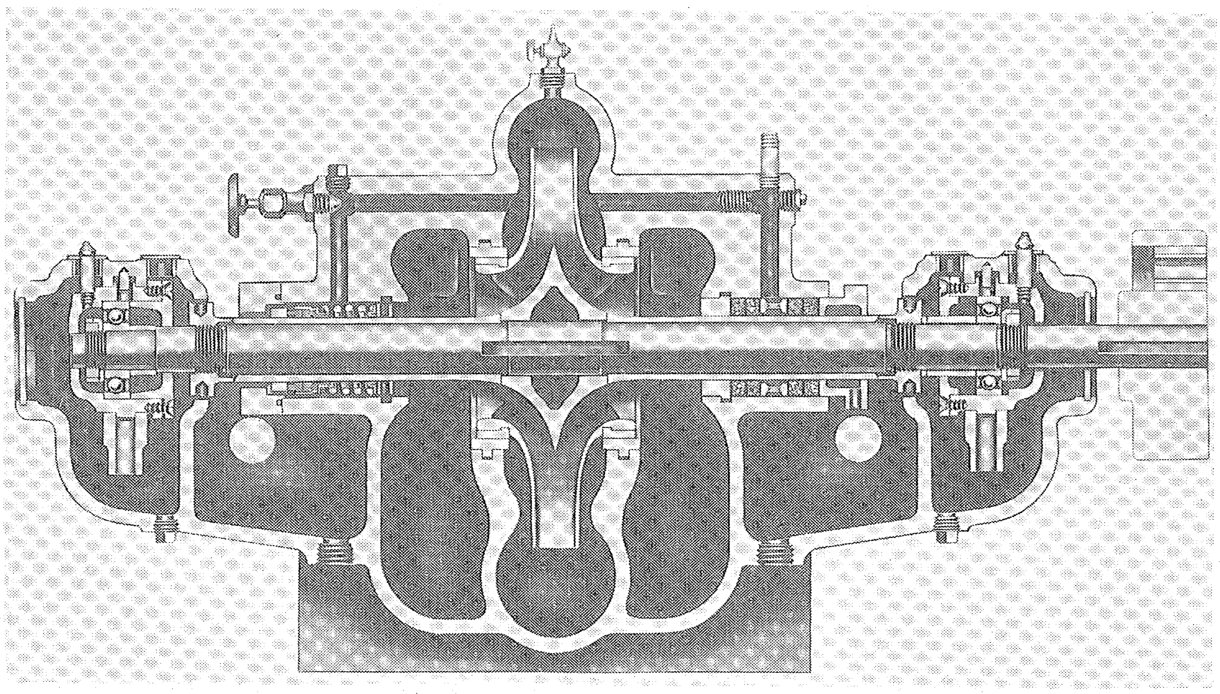


Figure 4. Sectional View of 5x4 SK Pump Showing Both Mechanical Seal and Packing.

TABLE I

PUMP DESIGN AND MATERIAL CHANGES

Component	No. of Changes	Type of Change
Impeller	2	Width - 15/16" and 3/4"
Impeller	Numerous	Dia. - 7" to 11" in approx. 1/8" increments
Shaft	2	Dia. - 1.467" and 1.687"
Shaft	3	Material - SAE 1045 Hot Rolled Steel Ni-Cu (Monel) Stainless Steel
Shaft Sleeve	2	Dia. - 1.468" I.D. and 1.687" I.D.
Shaft Sleeve	3	Length - Standard 5/8" shorter 3/4" longer
Shaft Sleeve	5	Material - Bronze AB Monel Stainless Steel ACM 141 (Allis-Chalmers Designation) SAE 1045 - Hot Rolled Steel
Bearings	4	Oil Lubricated Sleeve - "SB" Oil Lubricated Single Row Ball "B1O" Grease Lubricated Single Row Ball "B1G" Grease Lubricated Double Row Ball "B2G2"
Bearings	1	Snap Ring Added
Casing Ring	5	Standard Larger Bore and 1/8" Thinner Larger Bore and 1/16" Thinner Larger Bore and 1/8" Thicker Adjustable
Casing Ring	4	Materials - Stainless Steel Bronze E Cast Iron 25 ACM 144
Casing	4	Materials - Cast Iron 25 Stainless Steel Gun Metal Bronze EH
Packing and Seal	3	Materials - Soft Asbestos Fleximetal Mechanical Seal
Seal Cage	2	Standard - C.I. Gun Metal
Other Components	8	Angle Valve Instead of Aircock Tee Handle Oil Rings Alemite Collar Snap Ring for Bearing Bearing Adapter Adapter Cap Dowel Bearing Deflector

TABLE II

APPLICATION ENVIRONMENTS OF 5x4 TYPE SK AND KSK PUMPS

I. Type of Fluid Pumped

A. Description

1. Pure water
2. Water with significant impurities
3. Sea water
4. Brine (higher salt concentration than sea water)
5. Industrial chemicals
6. Petroleum products
7. Water with significant solids

B. Temperature ranges from ambient to 190°F.

C. Specific gravity from 0.9 to 1.9

D. Viscosity ranging from that of water to over 300 SSU.

II. Hydraulic operation from near shutoff to 35% in excess of rated flow.

III. Discharge pressures 0 to 250 psi.

Purchased component changes consisted of bearings, packings, and mechanical seals. Only minor changes were made in manufacturing techniques, processes, tools and skills.

Application environments of these pumps are given in Table II. These environments refer to the internal pump environment and affect the pump component and pump failure rate.

The operating environments of these pumps basically consisted of ground use by industry and shipboard use on aircraft carriers by BuShips. They are broken down in more detail in Table III.

TABLE III

OPERATING ENVIRONMENTS FOR 5x4 SK AND KSK PUMPS

Group	Ground Environment
1. a.	Condensate, Feedwater and Booster Pumps.
b.	Fire Pumps.
2.	General Service Pumps in Utility Companies.
3. a.	General Service Pumps in Other Industries Such as Cement, Glass, etc.
b.	General Service Pumps in Paper Industry.
4. a.	Irrigation Pumps.
b.	Stream Barker Service Pumps.
5. a.	Hydraulic Mining Service.
b.	Oil Company Service.
	Shipboard Environment
1.	Fuel Pumps (Navy).
2. a.	Fire Pumps (Navy).
b.	Tanker Fire and Butterworth Service Pumps.
3.	Catapult Water Brake Pumps (Navy).

PRELIMINARY PUMP GROUPINGS

The pumps selected for study were broken down into groups which reflected the changes in Table I. These groups essentially represent the coordinates to be

Allis-Chalmers Manufacturing Co.
CENTRIFUGAL PUMP RELIABILITY REPORT
Type KSK, 5x4, single stage, double suction, split casing pump

Customer__________
Customer Order No.__________
Shop Order No.__________
Shipping Order No.__________
Pump Serial No.__________
District Office__________
Date Shipped__________
Type Bearings__________
Speed__________
Impeller Dia.__________
Driver__________

Replaced or Repaired Part: Size & Name	Part No.	Date Part Replaced or Repaired	Amount of Running Time on Replaced Part - Hrs.	Reason for Replacement or Repair. If Wear, Estimate Amount of Wear, If Repair, Describe

Check one of following:

Average Pump operation per day	4 Hr ☐	8 Hr. ☐	16 Hr ☐	24 Hr ☐
% Impurities in pumped fluid	0%-- ☐	5%-- ☐	10%-- ☐	More ☐
No. of stops per day	0--- ☐	2--- ☐	10--- ☐	More ☐
Operation near shutoff	Never ☐	On Occ. ☐	Freq. ☐	
Pump operation (variation from rating)	5% ☐	15% ☐	35%-- ☐	More ☐

Date Placed in operation__________
Description of fluid being pumped:__________

Suction pressure:__________

Customer comments:__________

Downtime - cost/hr__________

INSTRUCTIONS

Complete as much of the report as possible using estimates if necessary.

List all parts that have been replaced or repaired on the pump identified by serial number at the top of the page.

Estimate the running time on replaced or repaired parts.

Give reason for replacement or repair of each part. Give an estimate of the amount of wear in terms of changes in dimensions or tolerances where applicable. Briefly describe repairs made.

Give the approximate operating range or variation from the rated point for the pump by checking the appropriate box.

Under "Comments" estimate total hours of pump operation, indicate vibration and noise levels and describe any unusual operating or environmental condition.

Under "Downtime" give cost/hour if the function of the pump is not performed due to pump failure, i.e., what would be the total cost of a pump failure due to loss of production, etc., if there were not any standby pumps. If a standby pump is used for emergencies, give its total cost per year.

Figure 5.

plotted on the total cost versus reliability curve. The primary changes which fashioned the preliminary grouping of the pumps are outlined in the following:

1. Shaft diameter was changed from 1.468″ to 1.687″, thus increasing the strength of the shaft.
2. Higher strength materials like stainless steel and Ni-Cu alloy were used for shaft, shaft sleeves and impeller.
3. Two types of fittings were used:
   a. Bronze
   b. Cast iron
   (The components, shaft sleeves, casing bushings, shaft nuts, casing rings and impeller are of bronze in the case of bronze fitted pumps and are of cast iron in the case of cast iron fitted pumps.)
4. Four different types of bearings:
   a. "B1G" - ball bearing with one row of balls, grease lubricated.
   b. "B10" - ball bearing with one row of balls, oil lubricated.
   c. SB - sleeve type bearings.
   d. "B2G2" - ball bearing with two rows of balls side by side and grease lubricated.

Two other important variables considered were impeller diameter and width. The effects of these two variables on pump head, radial load on the shaft and the bearings, bearing thrust load and pump brake horsepower are discussed under "Combined Pump Stress Analysis."

On the basis of above criteria, the pumps in this study were carefully studied and were divided into eighteen (18) preliminary groups.

DATA SOURCES AND ACQUISITION

The acquisition of the required basic pump performance reliability data was as follows:

1. From the Allis-Chalmers Fluid Dynamics Department the serial numbers of all pumps in the eighteen preliminary pump groupings were tabulated showing the information required to fill the upper portion of the form in Fig. 5.
2. The form in Fig. 5 was filled out and sent to Allis-Chalmers District Offices through which the pumps were sold. The District Offices requested that the commercial customer complete the form and return it. If after a prescribed time the forms were not received, a follow-up letter was sent out. In total 276 forms were sent out to commercial customers, of which 104 were returned filled out adequately, or 38%.
3. The serial numbers of pumps which had been sold to the Navy were grouped according to aircraft carriers on which they were installed. Letters were written through Lt. Commander Art Coyle, Power Branch, ONR, to the Commanders of the Pacific and Atlantic Fleets requesting that the Captains in command of the aircraft carriers have the form in Fig. 5 completed, and furthermore, supply Allis-Chalmers with a copy of their Machinery History Card, NAVSHIP 527 (Rev. 10-48), as shown in Fig. 6. Out of 151 pumps aboard Navy vessels information on 117 pumps was received or 78%. The

MACHINERY HISTORY CARD

FORM NAVSHIP-527

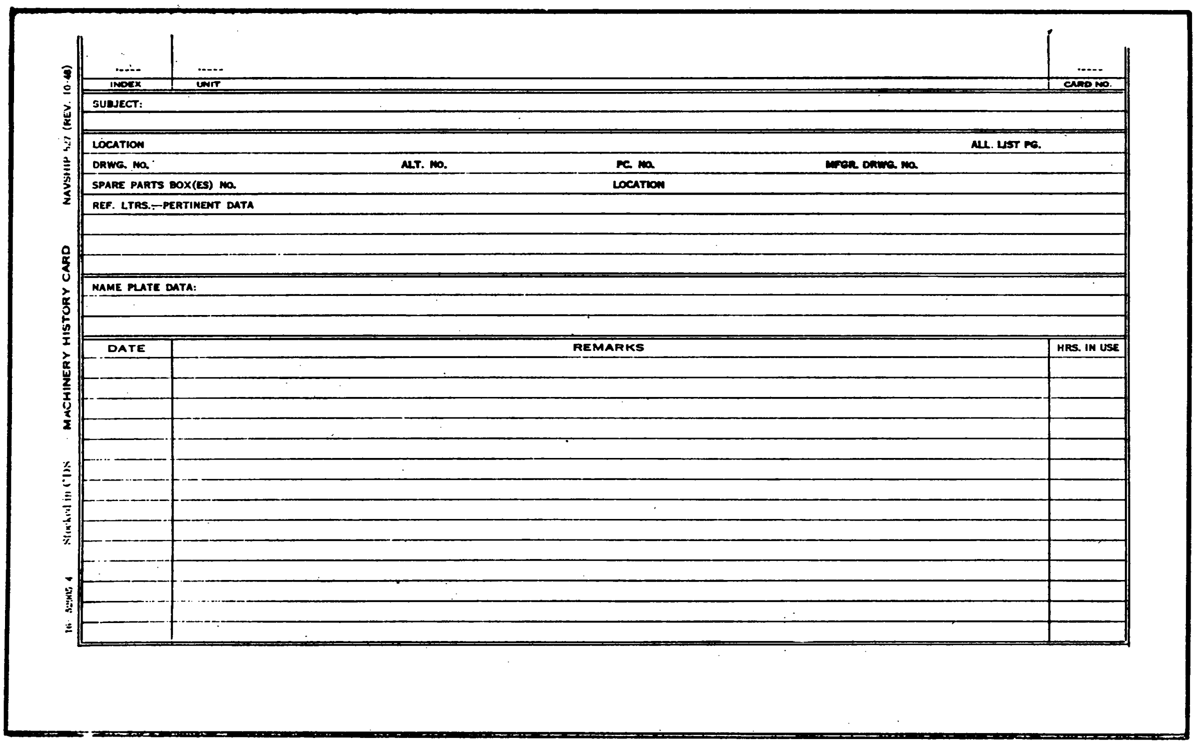


Figure 6.

outstanding cooperation of the carrier personnel in supplying us copies of their Machinery History Cards should be commended, as this information has enabled us to develop a bathtub curve for the fire pumps used aboard aircraft carriers. Data on the remaining pumps could not be obtained because either the vessels were at sea during the Cuban quarantine of the Fall of 1962 or vessels with pumps aboard were still under construction.

4. The Bureau of Ships, Code 706A, performed a search of their compilation of Reports of Equipment Failure, Form NAVSHIPS 3621 (Rev. 6-59) shown in Fig. 7 to obtain the total number of reported failures for the period under study.

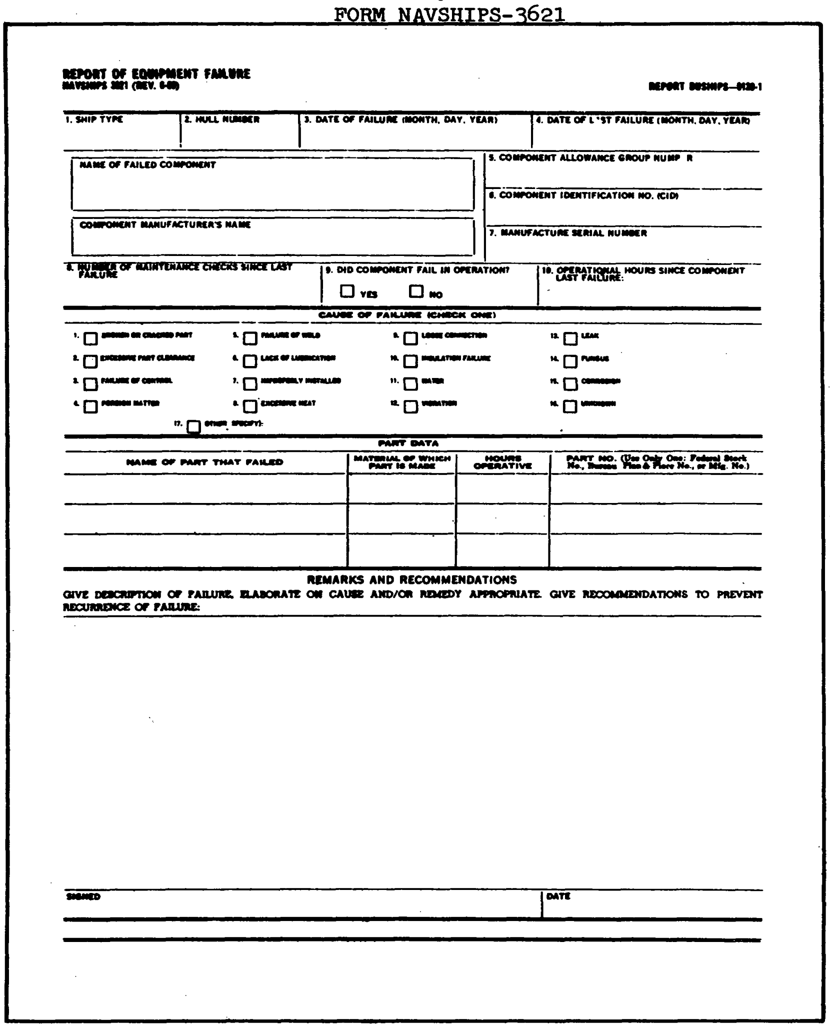
5. The Fluid Dynamics Department Renewal Parts file was searched for repair parts and date of order for all pumps sold commercially. This data was used to supplement the information received on the Centrifugal Pump Reliability Report.

The acquisition of the required basic pump cost data was as follows:

1. Works Accounting Department of the Allis-Chalmers Comptroller's Division supplied the pump specification cost. This cost was available directly from the original order forms and tickets and includes direct labor, materials, purchased components, and manufacturing burden.

2. The cost of research and development, net engineering, engineering changes during manufacturing, new patterns, small tools, shipping expense and certain miscellaneous items was obtained from the Accounting Department for all pumps built. These are accumulated for all company products on a product line basis and cannot be isolated for a specific model in that product line. Using the experience and judgment of personnel in areas where these

REPORT OF EQUIPMENT FAILURE
FORM NAVSHIPS-3621



REPORT OF EQUIPMENT FAILURE

REPORT BUSHIPS

1. SHIP TYPE | 2. HULL NUMBER | 3. DATE OF FAILURE (MONTH, DAY, YEAR) | 4. DATE OF LAST FAILURE (MONTH, DAY, YEAR)

NAME OF FAILED COMPONENT

5. COMPONENT ALLOWANCE GROUP NUMBER

6. COMPONENT IDENTIFICATION NO. (CID)

COMPONENT MANUFACTURER'S NAME

7. MANUFACTURE SERIAL NUMBER

8. NUMBER OF MAINTENANCE CHECKS SINCE LAST FAILURE

9. DID COMPONENT FAIL IN OPERATION? ☐ YES ☐ NO

10. OPERATIONAL HOURS SINCE COMPONENT LAST FAILURE:

CAUSE OF FAILURE (CHECK ONE)

1. ☐ BROKEN OR CRACKED PART
2. ☐ EXCESSIVE PART CLEARANCE
3. ☐ FAILURE OF CONTROL
4. ☐ FOREIGN MATTER
5. ☐ FAILURE OF WELD
6. ☐ LACK OF LUBRICATION
7. ☐ IMPROPERLY INSTALLED
8. ☐ EXCESSIVE HEAT
9. ☐ LOOSE CONNECTION
10. ☐ INSULATION FAILURE
11. ☐ WATER
12. ☐ VIBRATION
13. ☐ LEAK
14. ☐ FUNGUS
15. ☐ CORROSION
16. ☐ UNKNOWN
17. ☐ OTHER (SPECIFY):

PART DATA

NAME OF PART THAT FAILED	MATERIAL OF WHICH PART IS MADE	HOURS OPERATIVE	PART NO. (Use Only One: Federal Stock No., Bureau Plan & Piece No., or Mfg. No.)

REMARKS AND RECOMMENDATIONS

GIVE DESCRIPTION OF FAILURE. ELABORATE ON CAUSE AND/OR REMEDY APPROPRIATE. GIVE RECOMMENDATIONS TO PREVENT RECURRENCE OF FAILURE:

SIGNED | DATE

Figure 7.

costs are originated, the costs accumulated for the product line were apportioned to the pump groups under study.

3. The after shipment manufacturer's cost consisting of warranty and good will costs are compiled under a separate account at Allis-Chalmers. These were extracted by the Accounting Department for all the serial number pumps making up the groupings.

4. The customer support cost was determined as follows:

   (a) All parts replaced on the pump were requested to be entered on the Centrifugal Pump Reliability Report by the customer. These parts were then extracted from the reports for each group.

   (b) Costs of the replacement parts were provided by the Renewal Parts Section of the Allis-Chalmers Fluid Dynamics Department.

   (c) Hours to repair or replace each part entered on the Centrifugal Pump Reliability Report were found by using a teardown and assembly time chart or accessibility tree obtained from the Timestudy Section of the Allis-Chalmers Manufacturing Planning Department. They were complemented by values from the Pump Repair Section and the Service and Renewal Parts Section of the Fluid Dynamics Department.

(d) Downtime costs per hour to the customer were requested on the Centrifugal Pump Reliability report and ten customers responded. These are costs resulting from loss of production and stoppage of surrounding equipment. The product of the downtime cost per hour and the total pump repair time gave the downtime cost per group.

FINAL PUMP GROUPINGS

Due to insufficient data, evidenced by lack of customer report and/or hours of operation, to establish a reasonable confidence level in the calculated failure rates the pump groupings had to be changed. More specifically the changes are shown in Table IV and are discussed in the following:

Groups 1*, 2 and 3 were reapportioned into Groups I and II.

Groups 4, 5, 6 were regrouped as Groups III and IV.

Groups 7 and 8 returns were inadequate and those obtained could not be merged with the closest group, Group I, because of differences in casing and shaft sleeve material, higher discharge pressure flanges and the presence of an outside seal.

Group 9 was eliminated specifically because six of the seven pumps originally comprising this group were shipped to Puerto Rico and no reply to our reliability form was received. The one pump reported upon in Group 9 could not be combined with any other group because it is the only pump with sleeve bearings.

Group 10 was omitted since only 2000 hours of operation were accumulated on the nine pumps comprising the group.

Groups 11 and 12, which are Navy fire and catapult water brake pumps respectively, and had the most abundant returns were renumbered as Groups V and VI.

Group 13 consists of Navy jet fuel pumps aboard an aircraft carrier. However, inadequate data was provided and this group was deleted.

Group 14 consisted of stainless steel fabricated pumps sold to one company which had gone out of business.

TABLE IV

FINAL GROUPINGS

FROM PRELIMINARY GROUPINGS*

Final Group No.	Preliminary No.
I	1, 2, 3
II	
III	4, 5, 6
IV	
V	11
VI	12
VII	15, 16, 17
VIII	

*Prėliminary Groupings 7, 8, 9, 10, 13, 14 and 18 have been deleted. See text for reasons.

*Note: The original groupings are numbered in Arabic numbers while the final groupings are numbered in Roman numerals.

Table Va

FINAL PUMP RELIABILITY GROUPINGS AND THEIR COMPARATIVE DIFFERENCES

(Groups I through V)

Group	No. of Pumps	Shaft	Shaft Sleeve	Casing Ring	Ball Bearing
I	15	Small dia. shaft D = 1.468"	Small dia. 6-3/8" long	Bronze 'E'	B1G
II	17	Small dia. shaft D = 1.468"	Small dia. 6-3/8" long	Bronze 'E'	B1G
III	10	Larger dia. shaft D = 1.687" instead of 1.468"	Larger dia., longer by 3/4". Some pumps have ACM 141 shaft sleeve instead of bronze 'AB'	Bronze 'E'	B1G
IV	13	Larger dia. shaft D = 1.687" instead of 1.468"	Larger dia., longer by 3/4". Some pumps have ACM 141 shaft sleeve instead of bronze 'AB'	Bronze 'E'	B1G
V	90	Larger dia. tapered shaft of Ni-Cu alloy D = 1.687" instead of 1.468"	Larger dia., longer by 3/4". Shaft sleeve of Ni-Cu alloy.	Bearing bronze casing ring with larger bore but 1/16" thinner	B2G2

Group	Impeller	Casing	Others	Application
I	Larger dia. impeller. Wide outlet 7.0" to 9.0" dia. Narrow outlet 7.0" to 9.50" dia.	CI 25		Commercial
II	Larger dia. impeller. Wide outlet 9.0" to 11.0" dia. Narrow outlet 9.50" to 11.0" dia.	Some pumps in this group have BR 'EH' casing instead of C.I. 25	Some of the pumps in this group have outside seals	Commercial
III	Larger dia. impeller. Wide outlet 9.0" to 11.0" dia. Narrow outlet 9.5" to 11.0" dia.	CI 25		Commercial
IV	Larger dia. impeller. Wide outlet 9.0" to 11.0" dia. Narrow outlet 9.5" to 11.0" dia.	CI 25		Commercial
V	Larger dia. impeller of Ni-Cu alloy instead of bronze with wide outlet. Diameter = 10-1/8"	Gun metal casing instead of C.I. 25	Shaft nuts, ball bearing adapters, adapter caps, bearing caps, glands are of gun-metal instead of BR 'AB'	Navy pump used as fire pump

Table Vb

FINAL PUMP RELIABILITY GROUPINGS AND THEIR COMPARATIVE DIFFERENCES

(Groups VI through VIII)

Group	No. of Pumps	Shaft	Shaft Sleeve	Casing Ring	Ball Bearing
VI	16	Larger dia. tapered shaft of steel, D = 1.687" instead of 1.468"	Larger dia., longer by 3/4" shaft sleeve of Ni-Cu alloy	Bearing bronze casing ring with larger bore but 1/16" thinner	'B2G2 bearings
VII	20	Larger dia. shaft of carbon steel. D = 1.687" instead of 1.468"	Larger dia. sleeve with taper at key-way end and short by 5/8"	Larger bore and thicker by 1/8". It has 1/8" groove instead of tongue	Snap ring bearing retainer is used, B1G
VIII	14	Larger dia. shaft of carbon steel. D = 1.687" instead of 1.468"	Larger dia. sleeve with taper at the keyway end and short by 5/8"	Larger bore and thicker by 1/8". It has 1/8" groove instead of tongue	Snap ring bearing retainer is used, B1G

Table Vb Continued

Group	Impeller	Casing	Others	Application
VI	Wide outlet impeller of Ni-Cu alloy instead of bronze. Diameter 7" to 8.30"	Gun metal casing instead of C.I. 25	Shaft nuts, ball bearing adapter caps, bearing caps, glands are of gun metal instead of BR 'AB'	Navy pump, used as catapult fresh water brake pump
VII	Larger dia. impeller. Wide outlet 7" to 9.0", narrow outlet 7" to 9.5" dia.	CI 25	Spacer sleeve, ball bearing adapter, adapter cap, alemite collar, valve stem and straight dowel are not used, but bearing housing, bearing cover and deflector are used	Commercial
'III	Larger dia. impeller. Wide outlet 9.0" to 11.0". Narrow outlet 9.5" to 11.0" dia.	CI 25	Spacer sleeve, ball bearing adapter, adapter cap, alemite collar, valve stem and straight dowel are not used, but bearing housing, bearing cover and deflector are used	Commercial

Groups 15, 16, and 17 were merged and resplit into two groups, VII and VIII, to provide more population in each group.

Group 18 was deleted since the owners did not provide any reliability data although the entire group was sold on two orders.

The new and final eight (8) groupings are given in Table IV with the corresponding preliminary groupings and in Table V with the number of pumps in each group and the comparative differences between groups. In addition the following paragraphs describe a representative pump in each group.

Group I:

A bronze fitted pump having a small diameter shaft (D = 1.467″) of SAE 1045 annealed steel, "B1G" bearings, bronze impeller with a diameter ranging from 7.0″ to 9.0″ for a wide outlet and from 7.0″ to 9.5″ for a narrow outlet, cast iron casing, standard packing of soft asbestos and standard seal.

Group II:

A bronze fitted pump having a small diameter shaft (D = 1.467″) of SAE 1045 annealed steel, "B1G" bearings, bronze impeller with diameters ranging from 9.0″ to 11.0″ for a wide outlet and from 9.5″ to 11.0″ for a narrow outlet, cast iron casing, standard packing of soft asbestos and standard seal.

Group III:

A bronze fitted pump having a large diameter shaft (D = 1.687″) of SAE 1045 annealed steel, "B1G" bearings, bronze impeller with diameters ranging from 7″ to 9.0″ for a wide outlet and from 7″ to 9.5″ for a narrow outlet, cast iron casing, standard packing of soft asbestos and standard seal.

Group IV:

A bronze fitted pump having a large diameter shaft (D = 1.687″) of SAE 1045 annealed steel, "B1G" bearings, bronze impeller with diameters ranging from 9.0″ to 11.0″ for a wide outlet and from 9.5″ to 11.0″ for a narrow outlet, cast iron casing, standard packing of soft asbestos and standard seal.

Group V:

Assembled with a large diameter tapered shaft (D = 1.687″) of Ni-Cu alloy, "B2G2" bearings, wide outlet Ni-Cu alloy impeller with a diameter of 10-1/8″, gun metal casing, Fleximetallic packing and gun metal seal cage.

Group VI:

Assembled with a large diameter tapered shaft (D = 1.687") of steel, "B2G2" bearings, wide outlet Ni-Cu alloy impeller with diameters ranging from 7" to 8.47", gun metal casing, Fleximetallic packing and gun metal seal cage.

Group VII:

A bronze fitted pump having a large diameter shaft (D = 1.687") of SAE 1045 steel, "B1G" bearings, bronze impeller with diameters ranging from 7" to 9.0" for a wide outlet and from 7" to 9.5" for a narrow outlet, cast iron casing, standard packing of soft asbestos, standard wearing rings with a snap ring bearing retainer.

Group VIII:

A bronze fitted pump having a large diameter shaft (D = 1.687") of SAE 1045 steel, "B1G" bearings, bronze impeller with diameters ranging from 9.0" to 11.0" for a wide outlet and from 9.5" to 11.0" for a narrow outlet, cast iron casing, standard packing of soft asbestos, standard wearing rings with a snap ring bearing retainer.

COMBINED PUMP STRESS ANALYSIS

As can be seen by the description of the final groupings Groups I and II, III and IV, and VII and VIII are similar. The difference is that the first group in each case has a smaller range of impeller diameters. The division of impeller diameters is 9.0" for wide (15/16" opening width) outlet impeller, 9.5" for the narrow (3/4" opening width) outlet impellers. This division was arrived at by employing a general stress analysis of the pump groups. The results are used to determine certain failure rate modifiers for the pump reliability prediction. It should be emphasized that the analysis presented herein is not meant to be a rigorous fibre stress analysis. Instead comparative equations are introduced to obtain a measure of the general pump stress levels involved, and rank the pump groups accordingly.

Stress analysis involved determining the relative effect of increasing the impeller diameter, changing from wide to narrow impeller widths, and changing discharge pressures; on the stress level of the two sizes of shafts and the two bearing configurations which remained in the final groups.

The governing criteria of shaft failure is the maximum shear stress imposed upon the material. The equation for this stress is:

$$\text{Maximum Shear Stress} = 1/2 \left[\left(K_m \frac{M_c}{I}\right)^2 + 4\left(K_T \frac{T_c}{J}\right)^2 \right]^{1/2}$$

where K_m and K_T are variable and impact load factors and were obtained from Kent's Mechanical Engineers Handbook (19). The Moments of Inertia I and J were determined from the shaft dimensions.

The bending moment, M_c, was determined by the product of one-half the force acting on the impeller discharge and one-half the distance between bearings. The discharge pressures and consequently the force of the impeller discharge is obtained from Fig. 8 for narrow impellers and Fig. 9 for wide impellers.

The torque T_c, imposed on the shaft can also be found from Figs. 8 and 9 for the combination of impeller diameter and width under consideration by converting the brake horsepower requirement to torque.

Fig. 10 shows the relative shaft stress for all the groups. A factor of 2/3 times the stress curve for a large steel shaft gives the stress curve for monel since it has a yield strength 50% higher than that of steel.

In determining bearing stress both the radial and thrust loads were found. The

Curves show approximately the characteristics when pumping clear water with specific gravity of 1.0. No guarantee is made except for the rated point.

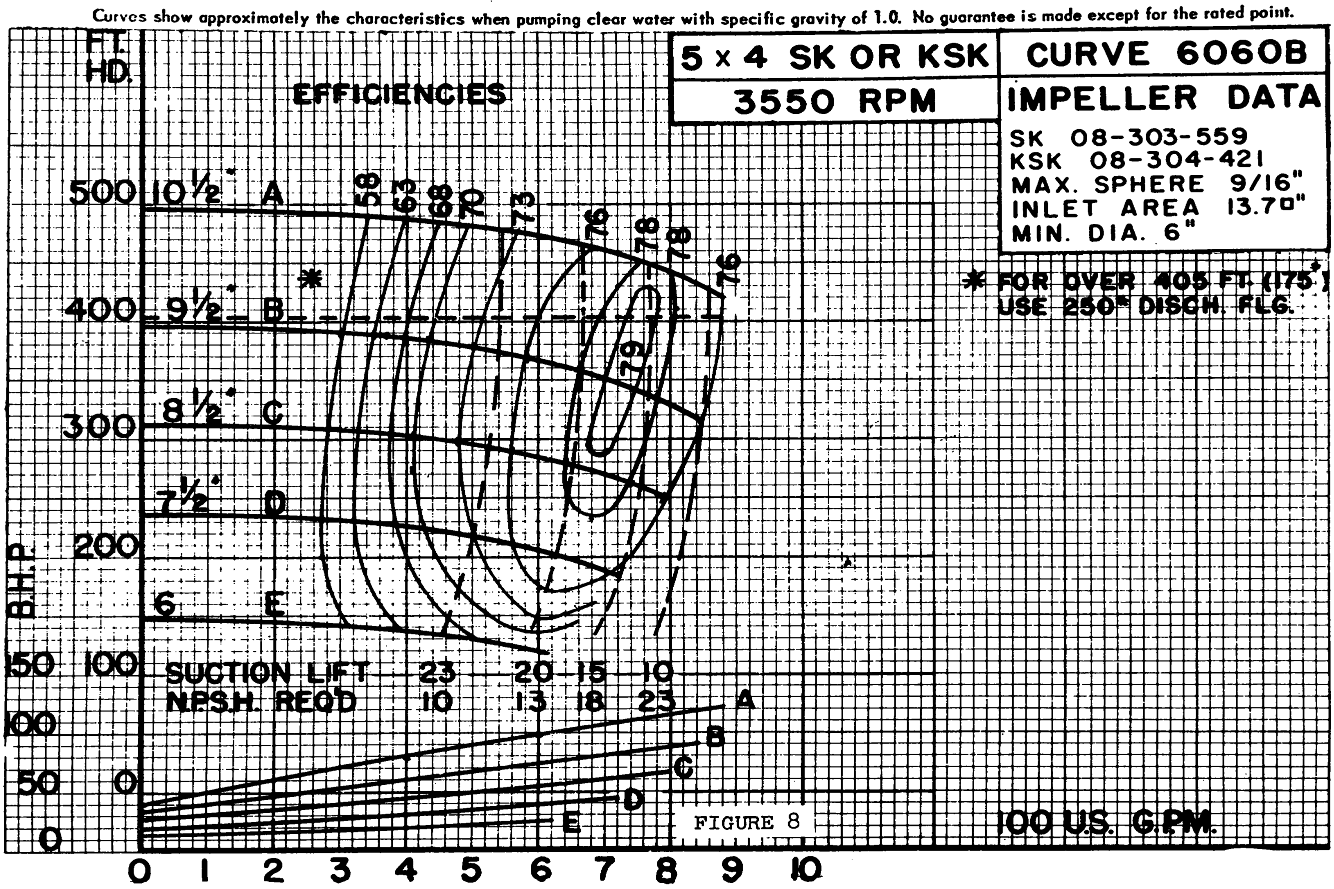


Figure 8.

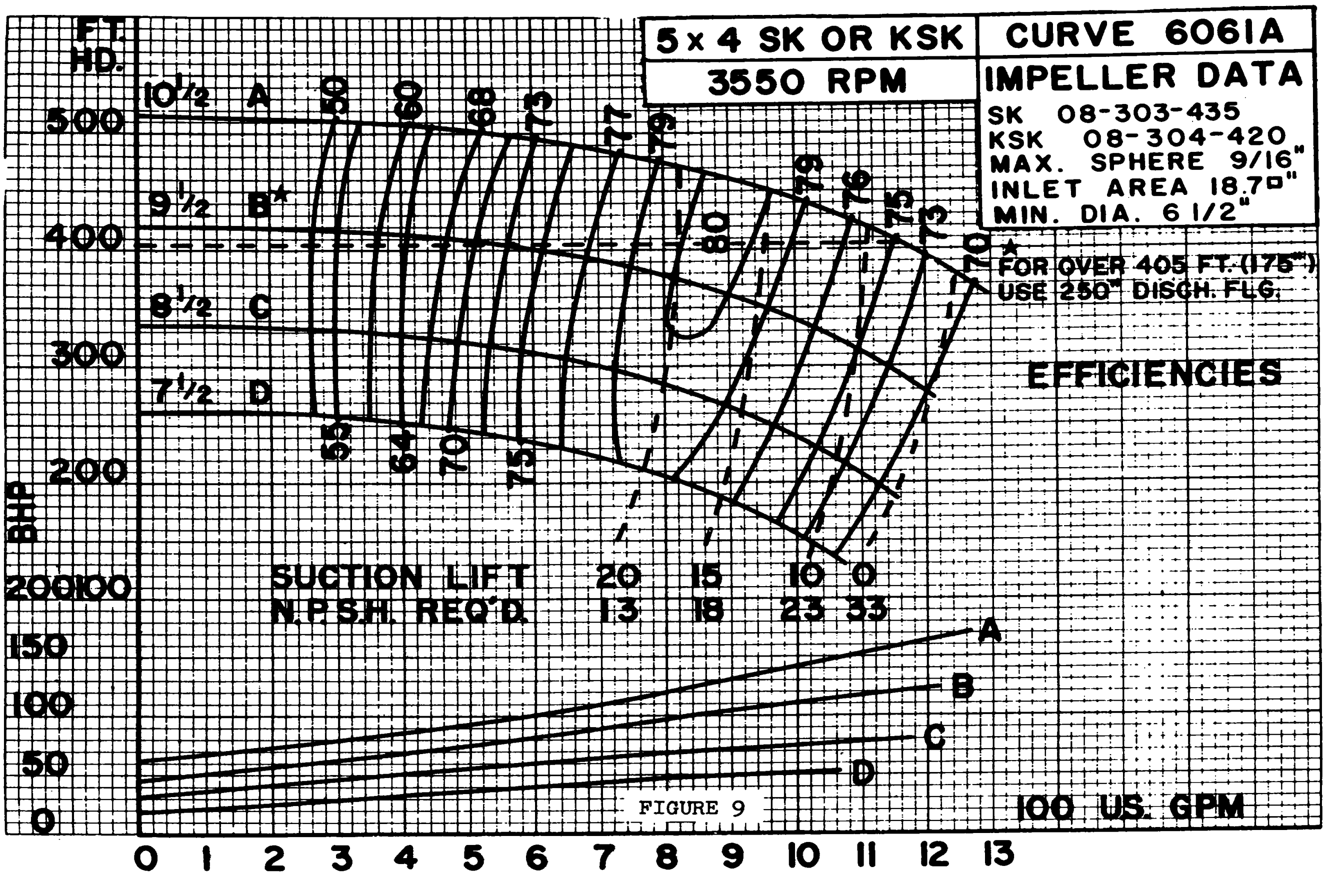


Figure 9.

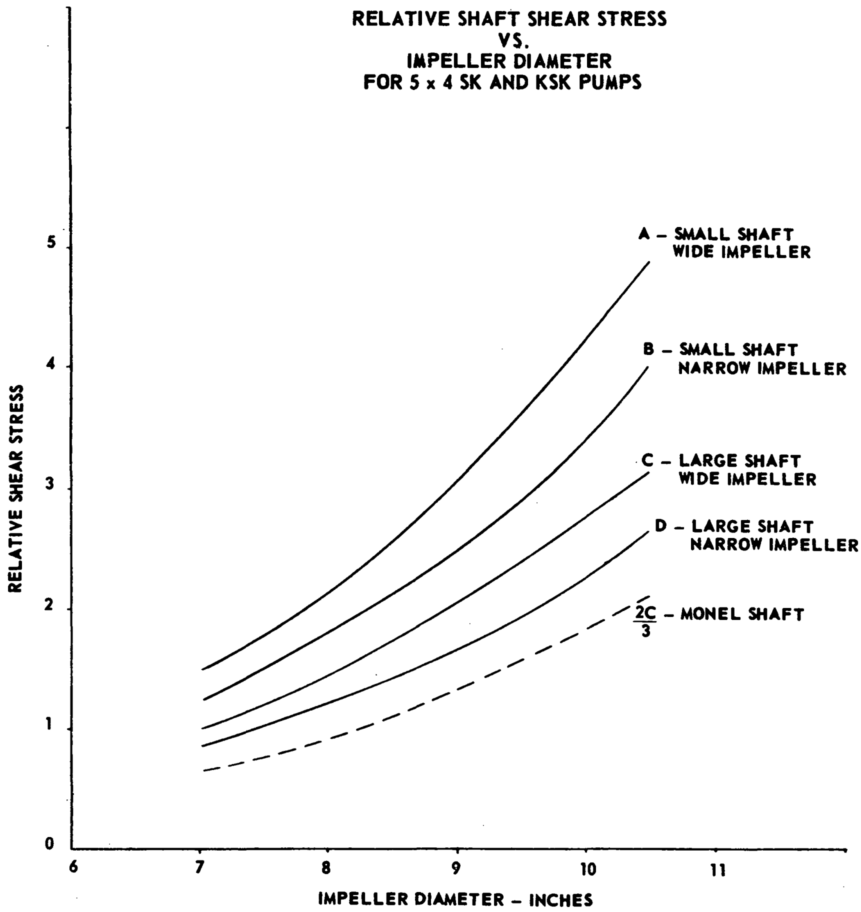


Figure 10.

radial load, L_R, is a function of the impeller width, W, and diameter, D, as may be seen from the following:

$$L_R = K_1 HA$$

where K_1 is a constant characteristic of a pump and its specific speed,
H is the pump head, and
A is the projected area.

Since H is proportional to the impeller diameter squared, D^2, and A is equal to impeller diameter, D, times the impeller width, W, the following ratio results for two similar pumps.

$$\text{Relative Radial Load} = \frac{W_1 D_1^3 N_2}{W_2 D_2^3 N_1}$$

where N is the number of bearings on each end of the shaft.

Similarly, the thrust load on the bearing is determined for the number of rows of balls in the bearing, N, by

$$\text{Thrust Load} = K_2 H \ (D^2 - d^2) \ (\frac{1}{N})$$

where d = shaft diameter

Substituting for H and forming a ratio we obtain an equation applicable to two similar pumps,

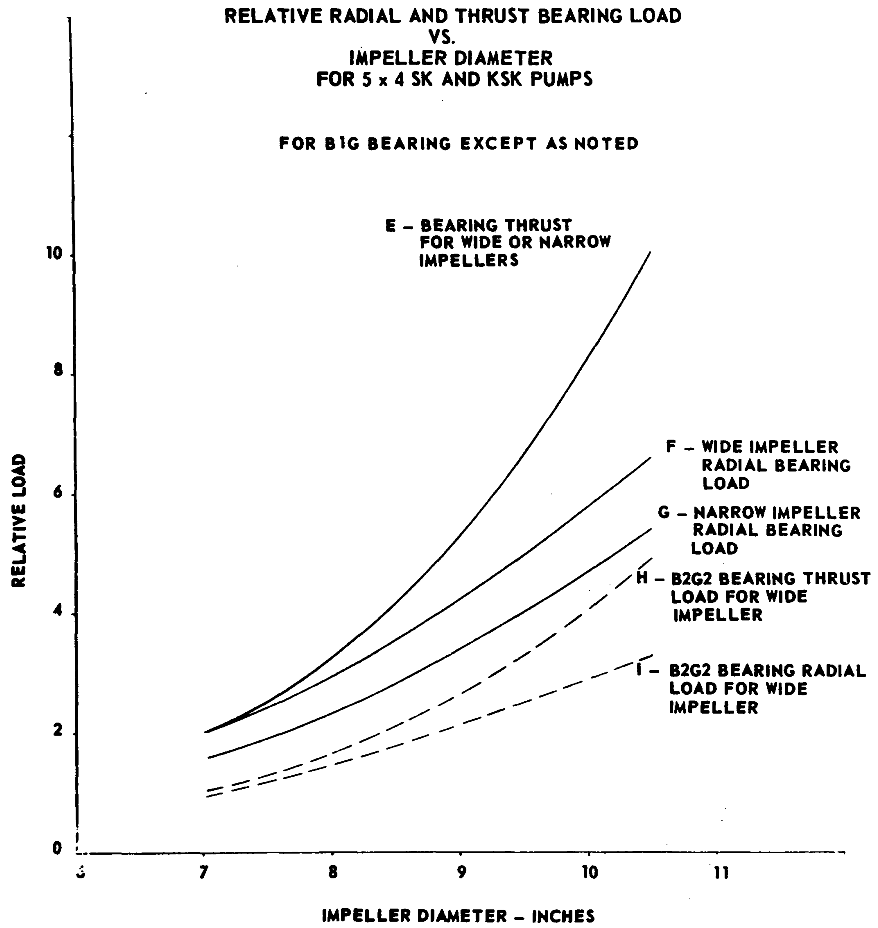


Figure 11.

$$\text{Relative Thrust Load} = \frac{(D_1^2 - d_1^2)D_1^2 N_2}{(D_2^2 - d_2^2)D_2^2 N_1}$$

The computed relative bearing radial end thrust loads are given in Fig. 11 for all groups.

The combined pump stress, shaft and bearing, as a function of impeller diameter was found by adding the appropriate curves in Figs. 10 and 11. Since the failure rates of the bearings and shaft would be added together in obtaining the pump failure rate, and furthermore, since failure rate is a function of stress, or load in the case of bearings, the following equations are obtained as a measure of the combined pump stress:

$$A + F + E = J$$

$$C + F + E = K$$

$$B + G + E = L$$

$$D + G + E = M$$

$$2C/3 + I + H = N$$

$$C + I + H = P$$

where the letters to the left of the equality sign refer to the curves in Figs. 10 and 11, and the letters on the right to the curves in Fig. 12. In Fig. 12 it can be seen

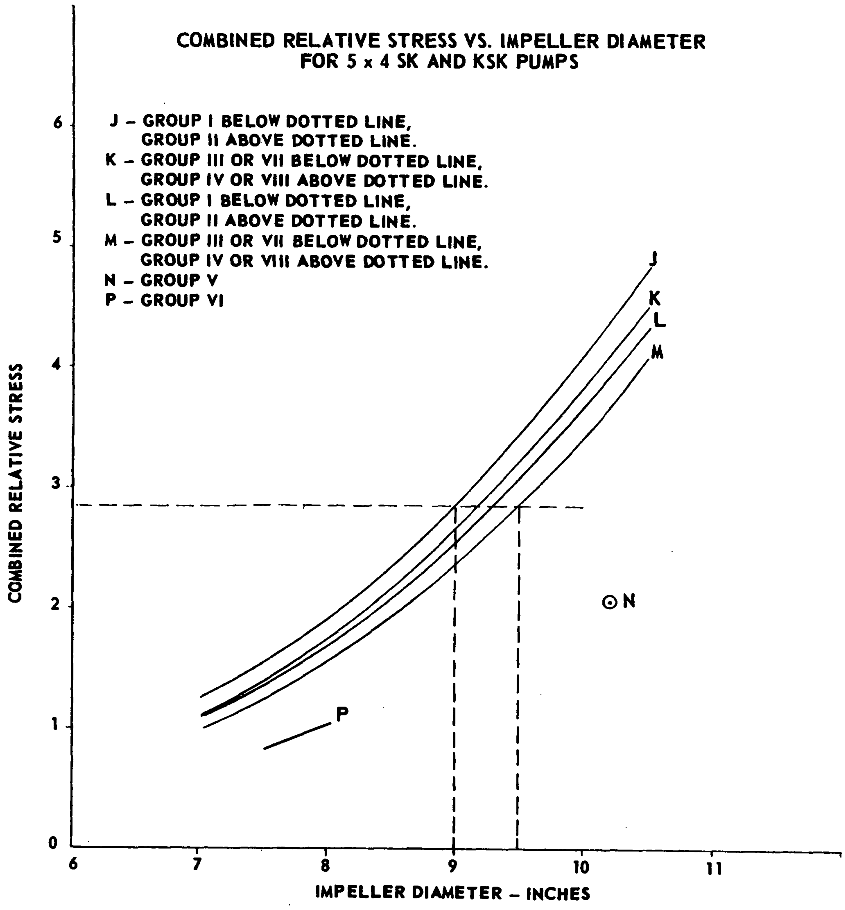


Figure 12.

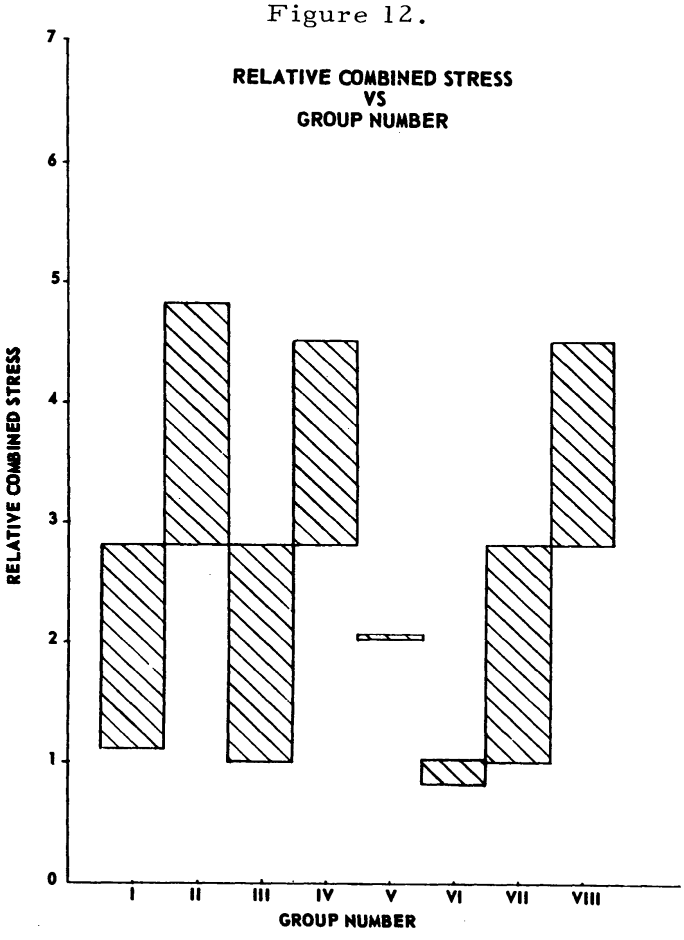


Figure 13.

that the relative stress level for pumps with "B1G" bearings and wide impellers is divided approximately in two at an impeller diameter of 9.0", and at 9.5" impeller diameter for pumps with "B1G" bearings and narrow impellers diameters. This is indicated by the horizontal dotted line. It was not necessary to divide the remaining pump groups up since their combined stresses were distributed over narrow impeller ranges.

The limiting stress values for each group in Fig. 12 are shown by bars in Fig. 13.

PREDICTED PUMP RELIABILITY

Prediction of the reliability of each pump group was required so that (1) a cross check on the actual field failure rates could be made to determine if unusual situations had significantly affected the data, and (2) modifiers could be developed to enable normalization of actual field data to a comparable environment.

The predictions were made under the assumption that the pumps would exhibit a constant failure rate, that all pump elements were in series reliabilitywise and that the part failure rates were independent of each other.

TABLE VI

GENERIC FAILURE RATES FOR 5x4 SK AND KSK PUMP COMPONENTS

Table VI-a: Failure rates of components common to all groups

	Generic Failure Rate fr/10^6 hr.		
Component Name	Max.	Mean	Min.
Shaft	0.62	0.35	0.15
Shaft Sleeves (2)*	0.60	0.30	0.04
Casing Bushing (2)	0.16	0.10	0.04
Shaft Nut (2)	0.0152	0.0084	0.0048
Casing Ring (2)	0.16	0.10	0.04
Straight Key	0.28	0.14	0.08
Step Key	0.18	0.09	0.05
Grease	0.016	0.010	0.004
Permatex	0.000	0.000	0.000
"O" ring (2)	0.06	0.04	0.02
Impeller	0.24	0.15	0.06
Casing	0.910	0.400	0.016
Gland (4 halves)	0.20	0.125	0.05
Gland Bolts-½" (4)	0 061	0.034	0.019
Seal Cage (4 halves)	0.08	0.05	0.02
Gasket (3)	0.675	0.414	0.150
Packing (2)	1.12	0.70	0.25
Cap Screw (12)	0.0455	0.0250	0.0145
Lock Washer (12)	0.0455	0.0250	0.0145
Pipe Plug (2)	0.000	0.000	0.000
Hex. Set Screw (2)	0.0152	0.0084	0.0048
Crank Case Sealer	0.016	0.010	0.004
Taper Dowel (2)	0.015	0.008	0.005
Bearing Cap (2)	0.6066	0.2666	0.0106
TOTAL	6.1210	3.3544	1.0472

*The number in parentheses indicates the number required. The failure rates given in the three columns are multiplied by the number.

The basis of this reliability prediction is the part generic failure rate. It is defined as the inherent failures per million hours of operation that will occur on a part or unit which is operated in a laboratory environment. The generic failure rate applies to the conditions of no externally applied vibration or shock. It also assumes, in the case of pumps, that pure water is being pumped, at continuous operation, within the most efficient operating range of the pump and that the water and surrounding air are at ambient temperatures. The generic conditions also assume perfect alignment of the pump and its driving motor so that the motor is not contributing to any stress risers within the pump.

A listing of pump components and their generic failure rates is given in Table VI. These generic rates, slightly adjusted according to our knowledge of the parts and their use in these pumps, were obtained from Earles (21). Nonexisting generic failure rates were developed by modifying those of a similar part by engineering judgment. Generally, three or four engineers were asked to come up with failure rates for parts on which no data was available; these were then weighed according to the level of experience of the engineer and averaged to establish the accepted failure rate appearing in Table VI. These generic part failure rates were summed for all components of a pump in each group to obtain the generic pump failure rate. These values are tabulated in Table VI and Table XII. The totals in Table VI were arrived at by first summing up the failure rates of all components common to all pumps. Then the sum of the failure rates of the parts peculiar to each group were added to the value for the common parts to obtain the generic failure rate of each group.

After the generic failure rates were thus established, the predicted field failure rates were developed. Two factors were developed which modify the generic

Table VI-b: Generic failure rates of additional components required for pumps in groups I, II, III, IV.

Component Name	Generic Failure Rate fr/10^6 hr Max.	Mean	Min.
Ball Bearing Adapter (2)	0.242	0.174	0.007
Adapter Cap (2)	0.082	0.025	0.001
Spacer Sleeve (2)	0.16	0.10	0.04
Bearing End Plate (2)	0.280	0.176	0.070
Ball Bearing #6206 or 6207	3.080	1.570	0.062
Ball Bearing #6305 or 6306	2.620	1.340	0.053
Oil Hole Cover (2)	0.000	0.000	0.000
Alemite Collar (2)	0.000	0.000	0.000
Valve Stem (2)	0.540	0.336	0.104
Straight Dowel (2)	0.015	0.008	0.005
Aircock Tee Handle	0.140	0.084	0.026
S.F. Hex. Nut (4)	0.030	0.016	0.010
Drive Screw (2)	0.000	0.000	0.000
Lock Washer (2)	0.0080	0.0040	0.0025
Lock Nut (2)	0.0080	0.0040	0.0025
Alemite Fitting (2)	0.089	0.055	0.022
Taper Dowel (4)	0.03	0.016	0.010
Cap Screw (21)	0.1575	0.0840	0.0525
TOTAL	7.485	3.9920	0.4675
Total Generic Failure Rates of Common Components (Table VI-a)	6.1210	3.3544	1.0472
Total for Groups I, II, III, and IV	13.6060	7.3464	1.5147

Table VI-c: Generic failure rates of additional components required for pumps in groups V and VI

Component Name	Generic Failure Rate fr/10^6 hr Max.	Mean	Min.
Ball Bearing Adapter (12)	0.242	0.174	0.007
Adapter Cap (2)	0.082	0.025	0.001
Bearing End Plate (2)	0.280	0.176	0.070
Ball Bearing C.E.	2.8500	1.4500	0.0575
Ball Bearing O.E.	2.1700	1.1125	0.0440
Oil Hole Cover (2)	0.000	0.000	0.000
Alemite Collar (2)	0.000	0.000	0.000
Valve Stem (2)	0.540	0.336	0.104
Straight Dowel (2)	0.015	0.008	0.005
Aircock Tee Handle	0.140	0.084	0.026
S.F. Hex.Nut (4)	0.0132	0.003	0.0042
Drive Screw (2)	0.000	0.000	0.000
Lock Washer (2)	0.0080	0.0040	0.0025
Lock Nut (2)	0.0080	0.0040	0.0025
Straight Hydraulic Fitting (2)	0.089	0.055	0.022
Taper Dowel (4)	0.03	0.016	0.01
Stud (21)	0.0798	0.0441	0.0252
S.F. Hex.Nut (21)	0.0798	0.0441	0.0252
Angle Valve (2)	0.000	0.000	0.000
Coupling Lock Nut	0.0076	0.0042	0.0024
TOTAL	6.6344	3.5399	0.4082
Total Generic Failure Rates of Common Components (Table VI-a)	6.1210	3.3544	1.0472
TOTAL for Groups V and VI	12.7554	6.8943	1.4554

Table VI-d: Generic Failure Rates or additional components required for pumps in group VII and VIII

Component Name	Generic Failure Rate fr/10^6 hr Max.	Mean	Min.
Bearing Housing (2)	0.6066	0.2666	0.0106
Bearing Cover (2)	0.16	0.10	0.04
Deflector (2)	0.16	0.10	0.04
Ball Bearing #6207	3.080	1.570	0.062
Ball Bearing #6306	2.620	1.340	0.053
Snap Ring (2)	0.18	0.09	0.05
Drive Screw (2)	0.000	0.000	0.000
"O" Ring (4)	0.12	0.08	0.04
Cap Screw (21)	0.1575	0.0840	0.0525
TOTAL	7.0841	3.6306	0.3481
Total Generic Failure Rates of Common Components (Table VI-a)	6.1210	3.3544	1.0472
TOTAL FOR GROUPS VII and VIII	13.2051	1.9850	1.3953

failure rate to meet the environmental conditions which the field pump experiences. The Application Factor, K_A and the Operating Mode Factor, K_{OP}, take into consideration the internal and external environments of each pump.

In addition, a factor is required for each pump group which will indicate the over-all level of reliability, design improvements, or state of development of the pump which was not accounted for in the part failure rates. This factor is called the Design Group Factor, K_G.

Application Factor

The Application Factor was developed from engineering knowledge of the effect of the environment on the materials used in the pump groupings. Table VII shows application factors for all the application environments the pumps encountered. The base value of $K_A = 1$ is for the pump pumping pure water. These application factors were applied to each pump as follows:

A non-corrosive pump operating in brine at 16-35% variation would have a $K_A = 3.5 \times 1.1 = 3.85$. Table VIII gives the average application factor for each one of the eight pump groupings.

Operating Factor

Operating environments such as vibration, acceleration, and shock affect the failure rate of the packing, bearings and other pump parts. Hence, to predict the actual field failure rates the Operating Factor — K_{OP} — was developed to obtain the failure rate in the actual operating environments. This is done by multiplying the respective generic failure rates by K_{OP}.

Earles (21) has given an "S curve" (Fig. 7, p. 391 in reference 21) showing mean K_{OP} factors for different equipment installations. But he did not show the deviation about the mean K_{OP} for different degrees of severity in operating environments.

The mean K_{OP} and its standard deviation was determined by using Earles' data which appear in Fig. 14. It was assumed that the logarithms of the failure rate data presented in this figure would give approximately a normal distribution. Therefore, a normal distribution curve for each one of the three installation environments, laboratory, ground, and shipboard, was fitted to these data points. For these distributions the following quantities were obtained.

Laboratory: Mean K_{OP} = 0.928
Standard deviation (σ) = 5.0
Logarithm σ = 0.7
3 σ limit (upper) = 135.2
3 σ limit (lower) = 0.01176

Ground: Mean K_{OP} = 9.45
Standard deviation (σ) = 3.5
Logarithm σ = 0.548
3 σ limit (upper) = 415.9
3 σ limit (lower) = 0.214

Shipboard: Mean K_{OP} = 18.8
Standard deviation (σ) = 3.0
Logarithm σ = 0.48
3 σ limit (upper) = 512.7
3 σ limit (lower) = 0.677

TABLE VII

APPLICATION FACTORS FOR 5x4 SK and KSK PUMPS

			K_A
1.	Pure water	0-4% impurities	1.0
		5-9% "	1.1
		10-14% "	1.2
2.	Pure water with significant impurities	15-20% "	1.3
3.	Salt water non-corrosive material		2.75
4.	Salt water corrosive material		4.0
5.	Salt water non-corrosive material with impurities		3.5
6.	Brine, corrosive material		5.0
7.	Brine, non-corrosive material		3.5
8.	Operation at 0-15% variation from rated		1.0
9.	Operation at 16-35% variation from rated		1.1
10.	Operation at 36-100% variation from rated		1.2
11.	Operation near shutoff	Never	1.0
		Occasionally	1.1
		Frequently	1.2
12.	Temperature (a) Packing	0-91°F	1.0
		91-190°F	1.2
		over 190°F	1.4
	(b) Bearings	0-90°F	1.0
		91-190°F	1.15
		over 190°F	1.25
	(c) The fluid temperature encountered in the use and application of these pumps would have a negligible effect on the failure rate of the other components.		
13.	Specific gravity	Up to .9	.9
		.9 to 1.1	1.0
		over 1.1	1.1
14.	Viscosity	Under 150 SSU	1.0
		150-300 SSU	1.10
		over 300 SSU	1.15
15.	Discharge pressure	0 to and including 150 PSI	1.0
		150 - 250 PSI	1.2

TABLE VIII

AVERAGE GROUP K_A & K_{OP} FACTORS FOR 5X4 SK AND KSK PUMPS

GROUP NO.	K_A	K_{OP}
I	1.23	6.20
II	1.17	7.00
III	1.11	5.9
IV	1.12	6.42
V	4.375	14.00
VI	1.28	16.00
VII	1.13	7.00
VIII	1.52	6.85

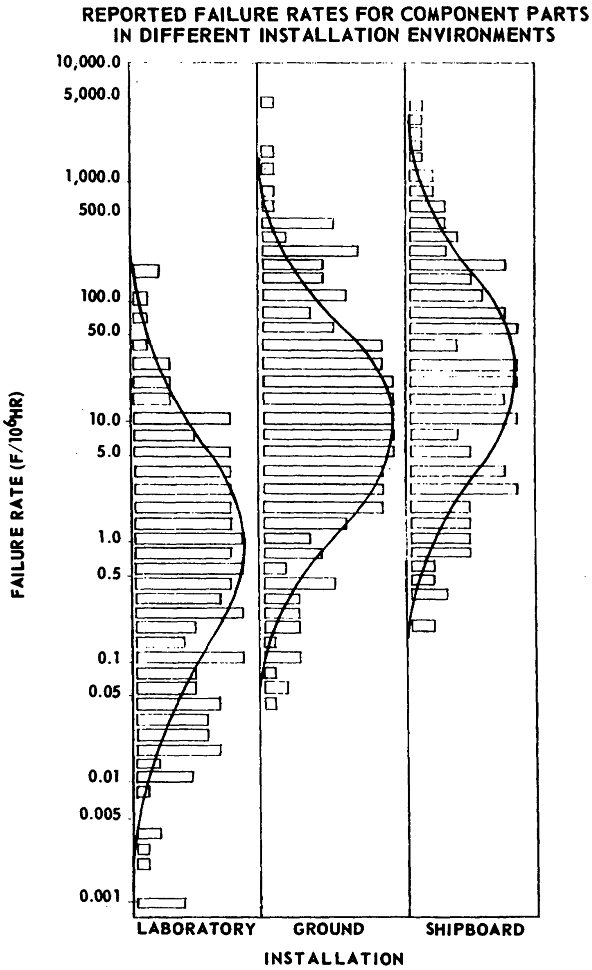


Figure 14.

From the foregoing it may be seen that the ratio of the mean ground failure rate to the mean laboratory failure rate is 10.2, whereas the ratio of the shipboard to the laboratory mean failure rate is 20.2. These values verify the Operating Factors given by Earles (21).

The operating environments found from the field data were studied carefully and it was decided that the variation in K_{OP} could not be between $\pm$ 3 σ or $\pm$ 2 σ limits but would most likely lie within $\pm$ 1 σ limits based on the following discussion.

Data presented in Fig. 14 was obtained from 30 different projects and 500 different components. These 500 components include components of various types of equipment including electronics, electrical, mechanical, hydraulic, etc. With this wide spectrum of equipment types the calculated $\pm$ 3 σ limits given above were obtained. However, when one type of equipment such as centrifugal pump is considered, the magnitude of the deviation range would be much smaller. It is estimated that $\pm$ 1 σ deviation of the total equipment population in Fig. 14 in the ground environments would essentially cover a $\pm$ 3 σ deviation for pumps alone in Ground environments.

Knowing that the range of variation for the Operating Factor would be in the same proportion as the variation of the failure rates in each installation environment, then the extreme limits of "K_{OP}" for Ground and Shipboard environments may be calculated as follows:

$$\text{Ground:} \quad \frac{\text{Min. } \lambda}{\text{Mean } \lambda} = \frac{\bar{x} - \sigma}{\bar{x}} = \frac{9.45 - 3.5}{9.45} = 0.63$$

$$\frac{\text{Max. } \lambda}{\text{Mean } \lambda} = \frac{\bar{x} + \sigma}{\bar{x}} = \frac{9.45 + 3.5}{9.45} = 1.37$$

From our experience the "mean K_{OP}" for these pumps in Ground environments is 7.0.

Hence, Min. K_{OP} = 0.63 x 7
= 4.41 <u>Say 4.5</u>

Max. K_{OP} = 1.37 x 7
= 9.59 <u>Say 9.5</u>

$$\text{Shipboard:} \quad \frac{\text{Min. } \lambda}{\text{Mean } \lambda} = \frac{\bar{x} - \sigma}{\bar{x}} = \frac{18.8 - 3.0}{18.8} = 0.84$$

$$\frac{\text{Max. } \lambda}{\text{Mean } \lambda} = \frac{\bar{x} + \sigma}{\bar{x}} = \frac{19.8 + 3.0}{19.8} = 1.16$$

$$\text{Mean } K_{OP} = \frac{20}{10} \times 7 = 14.0$$

(Mean values of "K_{OP}" estimated by Earles are: Ground = 10, Shipboard = 20)

Hence, Min. K_{OP} = 0.84 x 14
= 11.76 <u>Say 12.0</u>

Max. K_{OP} = 1.16 x 14
= 16.24 <u>Say 16.0</u>

After establishing the range of K_{OP} for the ground and shipboard environments, the K_{OP} values applicable to the pumps operating in the environments given in Table III were arrived at using engineering judgment as to the relative severity of these environments. These K_{OP} values are given in Table IX. As some of the groups contain pumps that do not have the same operating environment, the respective K_{OP} values of the pumps in each group were averaged and this value used as the one applicable to that group as a whole. The group K_{OP} factors are given in Table VIII.

<u>Design Group Factor</u>

The failure rate of a component changes with improvements in design and with changes in material, e.g., a shaft of larger diameter would have lower failure rate than that of a small diameter shaft under similar application and operating environments. Also, a shaft of higher strength material would have a lower failure rate than the shaft of the same diameter but of lower strength material. Because of this adjustment in failure rate should be made to predict the actual field failure rate from the predicted generic failure rate. Hence, a factor called the Design Group Factor, K_G, was developed.

To develop the Design Group Factor the eight pump groups were classified into six design classes that show the comparative improvement in design and material with respect to one another. The following items were found playing significant roles in the general stress level of the pumps:

TABLE IX

5x4 SK & KSK PUMP OPERATING ENVIRONMENTS AND KOP FACTORS IN INCREASING ORDER OF SEVERITY

Classification	Ground Environment	K_{op}
1.a	Condensate, Feedwater and Booster pumps	4.5
b	Fire pumps	
2.	General Service pumps in utility companies	5.5
3.a	General service pumps in other industry such as cement, glass, etc.	7.0
b	General service pumps in paper industry	
4.a	Irrigation pumps	8.0
b	Stream barker service pumps	
5.a	Hydraulic mining service	9.5
b	Oil company service	
	Shipboard Environment	
1	Fuel pumps (Navy)	12.0
2.a	Fire pumps (Navy)	14.0
b	Tanker fire and Butterworth service pumps	
3	Catapult water brake pumps (Navy)	16.0

TABLE X

PUMP GROUPS CLASSIFICATION INTO DESIGN CLASSES

AND

THE DESIGN GROUP FACTOR, K_G

Design Class, i	Group No.	E_i-Ratio of λ of a Class to λ of Class 0	K_G
0	VI	1.00	0.633
1	V	1.29	0.816
2	III	1.58	1.000
3	I, IV, VII	1.87	1.183
4	VIII	2.16	1.369
5	II	2.45	1.55

TABLE XI

PART FAILURE RATE RATIOS FOR SK AND KSK PUMPS

Name of Component	Max. Generic Failure Rate / Min. Generic Failure Rate	Mean Generic Failure Rate / Mean Generic Failure Rate
Shaft	0.62/0.15 = 4.13	0.35/0.15 = 2.33
Packing	1.12/0.25 = 4.50	0.7/0.25 = 2.80
Keys	0.28/0.08 = 3.50	0.14/0.08 = 1.75
Gaskets	0.225/0.05 = 4.50	0.138/0.05 = 2.76
Impeller	0.24/0.06 = 4.00	0.15/0.06 = 2.50
Gland	0.20/0.04 = 4.00	0.125/0.05 = 2.50
Casing Ring	0.16/0.04 = 4.00	0.10/0.04 = 2.50
Average Ratio	28.63/7 = 4.09	17.14/7 = 2.45

1. The shaft diameter was changed from 1.468″ to 1.687″.
2. Higher strength materials like NiCu alloy were used for the shaft.
3. Two different types of bearings were introduced.
4. A variety of impeller diameters was used.
5. Two impeller widths were used.
6. The date of manufacture of the group and the corresponding effect of modernized technique in manufacture were different among some groups.
7. Some parts such as shaft sleeves and impellers were made from higher strength materials in some groups.
8. The Navy pumps were manufactured to more rigid specifications than commercial pumps.
9. A redesign was made specifically for the purpose of reducing the cost of the basic line of pumps, resulting in the pumps in Groups VII and VIII.

Based on these factors and the results of the combined stresses shown in Fig. 13, a ranking of the eight groups, based on relative general stress was established, as shown in Fig. 15.

Considerations of the general stress result in the following change from the combined stress values which are shown for each group in Fig. 13. Only the pumps in Groups V and VI are in the same position as in Fig. 13. Groups I and II were moved up two stress levels in the abscissa or stress axis because these pumps

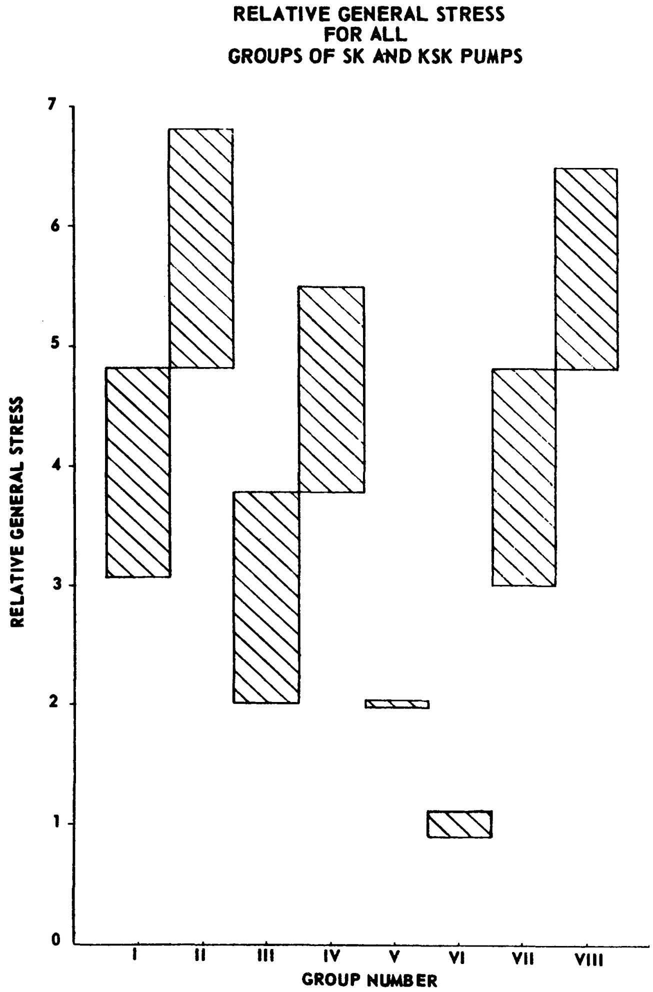


Figure 15.

were manufactured earliest and are not in general made up of as many high strength parts as the pumps in Groups V and VI. The pumps in Groups III and IV are better designed than the ones in Groups I and II but not as advanced as pumps in Groups V, VI, so their relative general stress lies between these groups, as shown in Fig. 15. Similarly, since the pumps in Groups VII and VIII were redesigned for initial cost cutting purposes and are not as well designed as the ones in Groups III and IV, their relative general stress is above that of these groups. The general stress axis in Fig. 15 is divided into six design classes. Table X shows the groups that fall into each design class.

It is mentioned above that each significant change in design affects the failure rate of the pump. Thus, any improvement in the design of any component of the pump which causes a change in its failure rate establishes a definite range between the failure rates of the two designs. As the classification of eight groups into six design classes is based on the improvements in the design of the pump, there is a range between failure rates of the six design classes.

Each component part of a system has mean, maximum, and minimum failure rate. If the mean is considered to be typical of average design, then the maximum failure rate corresponds to a below average design and the minimum failure rate to a better than average design. Thus any change in failure rate due to improvements in design could be represented by one of the following two ratios:

1. Maximum generic failure rate to minimum generic failure rate, and
2. Mean generic failure rate to minimum generic failure rate.

Each of the two ratios has a definite meaning. The first ratio represents the improvement from early design to well developed design, and the second ratio represents the improvement from average developed design to well developed design. These two ratios were calculated for major components and are given in Table XI.

Allis-Chalmers has been manufacturing pumps for several decades and during this period the design has been reviewed many times to introduce new developments to improve pump designs. These improvements are exhibited by the pump groupings. Because of the extensive past history of these pumps, the ratio of maximum generic failure rate to minimum generic failure rate was thought to be too high to use as a representative ratio of the failure rates between classes 0 and 5. Hence, it was decided to use the ratio 2.45 (see Table XI), the ratio of mean generic failure rate to minimum generic failure rate. Thus λ class $5/\lambda$ class $0 = 2.45$, and the ratio of the failure rate of all the design classes to the failure rate of design class 0 can be determined by assuming a linear relationship. These ratios, called E_i are given in Table X.

The ratios in Table X are relative multipliers showing the increase in failure rate for different pump designs. Now, absolute multiplying factors need to be developed. A pump which is representative of today's state of design and development, and is commensurate with industry practice should be assigned a $K_G = 1$. The pumps in Group III are the most logical selection. Hence the K for Group III is 1. The Navy pumps are specially designed pumps and their designs are better than the design of the pump in Group III. Thus, the K_G for the Navy pump groups will be less than 1. The reverse is the case for the remaining groups.

Therefore, the factor K_G for all groups is determined as follows:

$$K_G \text{ for Design Class 2} = \frac{E_1}{E_2} = \frac{1.29}{1.58} = 0.816$$

The K_G for all other groups was calculated similarly and is given in Table XII.

TABLE XII

PREDICTED GENERIC AND FIELD FAILURE RATES FOR 5x4 SK AND KSK PUMPS

GROUP NO.	Predicted Generic Failure Rates Min.	Mean	Max.	Application Factor K_A	Operating Factor K_{OP}	Design Group Factor K_G	Predicted Field Failure Rates Min.	Mean	Max.
I	1.51	7.34	13.61	1.23	6.20	1.18	13.6	66.1	122.5
II	1.51	7.34	13.61	1.17	7.00	1.55	19.2	93.2	172.9
III	1.51	7.34	13.61	1.11	5.90	1.00	9.9	48.1	89.2
IV	1.51	7.34	13.61	1.12	6.42	1.18	12.8	62.1	115.2
V	1.45	6.90	12.75	4.375	14.00	0.82	72.8	346.5	640.4
VI	1.45	6.90	12.75	1.28	16.00	0.63	18.7	89.0	164.5
VII	1.39	7.00	13.20	1.13	7.00	1.18	13.0	65.3	123.2
VIII	1.39	7.00	13.20	1.52	6.85	1.37	20.0	100.6	189.7

TABLE XIII

FIELD FAILURES, TOTAL HOURS OF OPERATION, AND FIELD FAILURE RATE FOR EACH GROUP

GROUP NO.	NO. OF FAILURES	HOURS	FIELD FAILURE RATE fr/10^6 Hour
I	11	345,548	31.8
II	14	427,654	32.8
III	2	258,005	11.6
IV	7	298,918	23.4
V	See Table XIV		115.0
VI	2	67,200	29.7
VII	4	166,428	24.2
VIII	3	90,022	33.4

TABLE XIV

PUMP FAILURES AND FAILURE RATE FOR GROUP V

Operating Hours Interval	Hours Accumulated By All Pumps During The Interval (T)	No. of Failures (f)	Failure Rate fr/10^6 Hours
0-1,000	80,109	22	274.8
1,000-2,000	58,572	6	102.3
2,000-3,000	50,605	6	118.5
3,000-4,000	47,659	9	188.9
4,000-5,000	37,829	4	105.8
5,000-6,000	33,000	4	121.1
6,000-7,000	32,949	5	151.8
7,000-8,000	31,583	3	95.0
8,000-9,000	30,308	5	165.0
9,000-10,000	18,887	2	106.0
10,000-11,000	15,654	1	64.0
11,000-12,000	15,000	1	66.7
12,000-13,000	14,113	1	70.8
13,000-14,000	13,263	1	75.5
14,000-15,000	12,552	1	79.6
15,000-16,000	12,000	2	166.8

Sample Calculation:

Interval: 0-1,000

$$\text{fr}/10^6 \text{ hrs. } (\lambda) = \frac{\text{No. of Failures } (f)}{\text{Hours accumulated by all pumps during the interval } (T)} = \frac{22}{80,109} = 274.8$$

Predicted Pump Field Failure Rates

The predicted pump field failure rates were calculated by taking the product of the respective group generic failure rates, application factor (K_A), the operating factor (K_{OP}), and the design group factor (K_G). These factors and the predicted failure rates for each group are all tabulated per group in Table XII.

ACTUAL PUMP RELIABILITY

Actual pump reliability was determined by the analysis of the performance data obtained from the sources discussed in "Data Sources and Acquisition." The basis for analyzing this data is the identification of a reliability failure. Criteria for this identification are discussed next.

Criteria for Field Failure Classification

A relevant reliability pump failure has occurred when a pump ceases to supply the required output or stops for any reason, excluding scheduled operational stops, scheduled maintenance stops or any reason outside the pump, i.e., power failure, damage to pump from outside sources, or the pump being a secondary failure. In addition, for the purpose of this study, excessive noise, leakage and/or vibration constitutes a failure.

All relevant pump failures were classified as wearout and random. Any time dependent failures were classified as wearout and included wear and corrosion. Any time independent failures, in this case, the rest of the relevant failures, were considered as random failures.

The reason for this failure division is that data received from the customer was not detailed enough. Principal source of data was the Centrifugal Pump Reliability Report and Machinery History Cards. In general they were filled out well enough to determine whether or not a repair or replacement was the result of an unscheduled pump stoppage. The Centrifugal Pump Reliability Reports and the Machinery History Cards contained several factors which often gave clues to what occurred, such as the language used, the time period between repairs and the type and number of parts replaced.

Parts which wore out at uniform operating intervals, even though of short duration, were not counted as relevant failures, particularly if they occurred on the same ship or installation. In general, a broken part was counted as a relevant failure.

Failures attributed to the drain pipes and their valves on the Navy pumps were not counted since these are parts extraneous to the basic pump.

Wearout of the casing and impeller rings and shaft sleeves are not ordinarily detectable during pump operation. Therefore, wearout will generally be detected only during a maintenance inspection or when another part is being replaced. Such a wearout was not considered to have caused an unscheduled pump outage. However, breakage of these parts was counted as a relevant failure.

Manufacturing errors due to poor workmanship were counted as relevant failures. Field failures which occurred due to incorrect maintenance practices or workmanship were not counted as relevant failures if they were of a repeating type because these were not the result of pump unreliability. Non-repeating failures which were caused by poor workmanship were considered relevant.

Spare parts orders were used as indicative of a pump failure only after the following factors were considered.

1. Time between pump installation and spare parts ordered.
2. Type of parts ordered (parts that are normally replaced as part of maintenance program or others).

3. Number of spare parts ordered.
4. Order specified parts for a breakdown.
5. Orders were so spaced that the interval between orders was a great deal less than the expected wearout life of the components or reasonable preventive maintenance intervals.

By using the above failure criteria, all failure data sources were examined and the reliability failures counted. Failures reported by BuShips, Code 706A on the Report of Equipment Failure, were checked against those counted by examining the Carrier's Machinery History Cards, and all were accounted for. The failures thus obtained for each group are tabulated in Table XIII.

Determination of Hours of Operation

In general the customer reported this by checking the closest value of the average pump operating hours per day on the Centrifugal Pump Reliability Report. However, several customers provided actual hours and also hours between repairs and replacements, as well as checking a box on the questionnaire. From this latter information the following relationship between the checked average operating hours per day and the actual operating hours per year were determined.

4 hrs/day equivalent to 1250 hrs/yr.
8 hrs/day equivalent to 2500 hrs/yr.
16 hrs/day equivalent to 5000 hrs/yr.
24 hrs/day equivalent to 7500 hrs/yr.

Using the above values and the date the pump was put into operation, as stated by the customer, the total pump hours of operation were determined. The results are given in Table XIII for each pump group.

Calculation of Failure Rates

From the failures and hours of operation, the failure rates for all groups except Group V were calculated from:

$$\text{Group Failure Rate} = \frac{\text{Total Failures Occurring for the Group}}{\text{Total Accumulated Hours of Operation by All Pumps in the Group}}$$

The results are given in Table XIII. The failure rate of Group V was determined as follows:

Using the data provided by the Navy for Group V, a reliability "bathtub" curve was constructed. The hours of pump operation accumulated before a failure occurred were not available for all pumps, however. In such cases, they were estimated using the equation given below to determine the operating hours per month. The number of months of service were calculated from the date the pump was placed into service up to October 1, 1962.

$$\frac{\text{Accumulated hours of operation up to Oct. 1, 1962}}{\text{Number of months in service}} = \text{Hours of operation per month of service}$$

Accumulated hours of operation and months of operation before failure were taken from the Centrifugal Pump Reliability Report and/or Machinery History Cards. The product of the hours of operation per month and the months operated before failure gave hours of operation up to the failure.

Maximum total hours of operation accumulated on individual pumps were less than 16,000 hours. This span was divided into 16 equal intervals.

The hours of pump operation and failures which occurred during each interval were determined and are given in Table XIV. The failure rates were calculated using the following equation:

$$\text{Failures}/10^6 \text{ hours} = \frac{\text{Number of failures during the interval}}{\text{Millions of hours of operation accumulated by all pumps during the interval}}$$

Group V – Navy Pump "Bathtub" Curve

A graph of failure rate vs. hours of operation, was plotted giving the reliability bathtub curve of Fig. 16. Here the assumption was made that there was immediate replacement after a failure and downtime is negligible. Some pumps, though not failed, have operated only part way through an interval because they have not been in service long. This was taken into account in arriving at the total operating hours for the pumps in each interval. Each failure rate point was plotted at the mean of the interval. The graph given as Curve 1, Fig. 16, is the actual curve obtained by using field failure rates in Table XIV. Curve 2, on Fig. 16, is a uniform mean of failure rates of all intervals excluding those of two extremes. Curve 3 is a step mean. The step mean consists of two parts: (1) between 1,250 to 9,250 hours; and (2) between 9,250 to 14,250 hours. Each portion has its own mean.

These curves may be interpreted as follows:

Curve 1: Between hours 0-1,250 there is a sharp fall in failure rate which may be interpreted as a period of early failures. Then, between hours 1,250-2,750 the failure rate is more or less constant representing random occurring failures during this period. The curve then shows a sharp rise indicating frequent wearout failures of short life components. The rise in failure rate between hours 4,500-6,500 may be due to the wearout failures of those short life components which were replaced during the previous period of frequent wearout. If wornout components are not removed simultaneously, but gradually as they failed, the curve will be considerably flattened, as shown. The curve shows that there is again a steep rise in failure rate at the point where the previous bell shaped curve ends. This forms another bell shaped curve between hours 7,500-8,500. This indicates frequent wearout failure of long life components

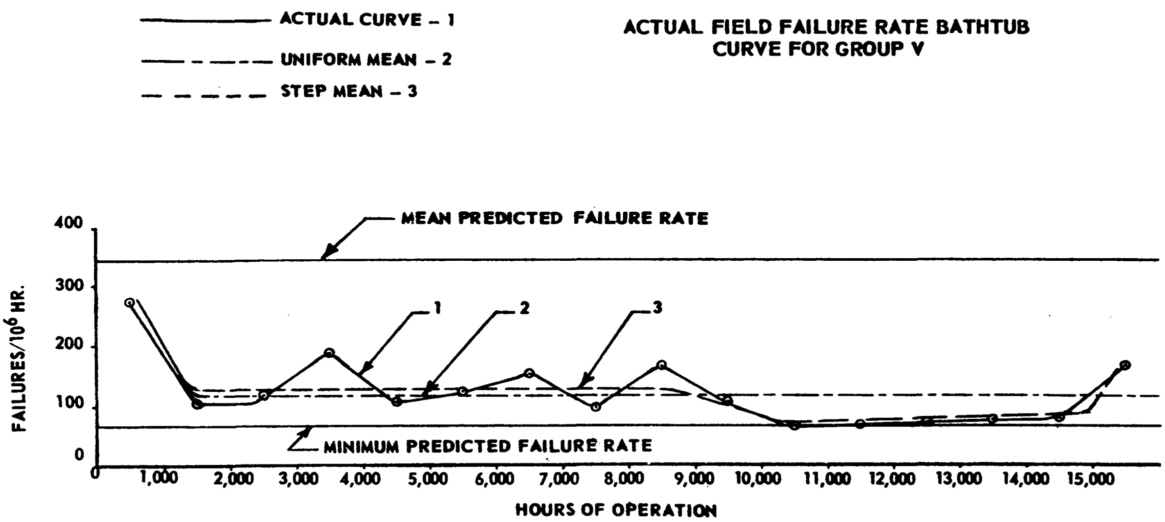


Figure 16.

and also of the short life components which were replaced between hours 2,750-6,500. Failure frequency between hours 10,000-14,500 is constant and indicates that the failures are occurring randomly. The failure reports indicate that the major overhaul was done on most pumps at about 9,000 hours. This may also explain the low failure rate after this period. At the end of this period the curve again rises indicating frequent wear-out failures.

Curve 2: It has been already mentioned that the repair and replacement parts reports obtained from the U.S. Navy were not complete. Consequently, the failures were classified according to the rules discussed previously. Thus, the failure rates given in Table XIV are only an estimate of the true failure rates. These rates are subject to statistical error which may be large or small, depending upon the volume of data and its accuracy. By calculating one uniform mean failure rate, these errors are redistributed and a curve is obtained amenable to easier analysis.

Curve 3: Curve 3 was drawn to increase the accuracy of estimating space part provisioning over that of Curve 2 and to approximate points closer by dividing the whole span into two equivalent "bathtub" curves.

Curve 3 could be indicative of two major facts: (1) The ship's operating environment is changing. Perhaps, since these pumps were placed aboard the ship when it was being constructed, the curve indicates that much of the early life of the ship is spent in harbors and as it ages it spends more time at sea. If so, the fire pumps during their early life would be pumping sea water with considerable amounts of sand, because the intakes for the fire pumps are on the bottom of the hull, which would definitely increase the pump's failure rate. As a greater percentage of time is spent at sea, the pump's failure rate would decrease because of the lower content of sand in the sea water. (2) The maintenance practices of the ship's crew improves with time. It is possible that the crews are somewhat inexperienced when the ships are initially put to sea and their maintenance procedures and quality of work is low. However, as time goes by they improve and pumps experience lower failure rates because of the reduction in failures due to misalignment, incorrect torqueing, etc. This assumes, of course, that there is no frequent rotation of crew which brings in a relatively inexperienced crew.

The pump failure rates will be used here as a measure of the pumps' reliability. This is done so because, as reliability is a function of mission time, as well as of failure rate, and the mission time may be different for each group of pumps, it eliminates the variability of mission time. However, knowing the failure rate and selecting an applicable mission time the pumps' reliability may be calculated when the product exhibits a relatively constant failure rate characteristic.

COMPARISON OF PREDICTED VERSUS ACTUAL FIELD FAILURE RATES OF THE PUMPS

To evaluate the pump field failure rate prediction technique developed in this study, Fig. 17 was prepared where the predicted values are shown alongside the actual field failure rates for all eight pump groups. The predicted values came from Table XII and the actual values from Table XIII and Fig. 16.

The maximum, mean and minimum pump failure rates, given in Table XII, are related in general to the state-of-the-art in the following manner:

The maximum failure rate corresponds to a pump which is in an early state of development, the mean failure rate to an average developed pump, and the minimum to a pump with an advanced design. By examination of Fig. 17 it can be seen that

ACTUAL AND PREDICTED GROUP FAILURE RATE HISTOGRAM

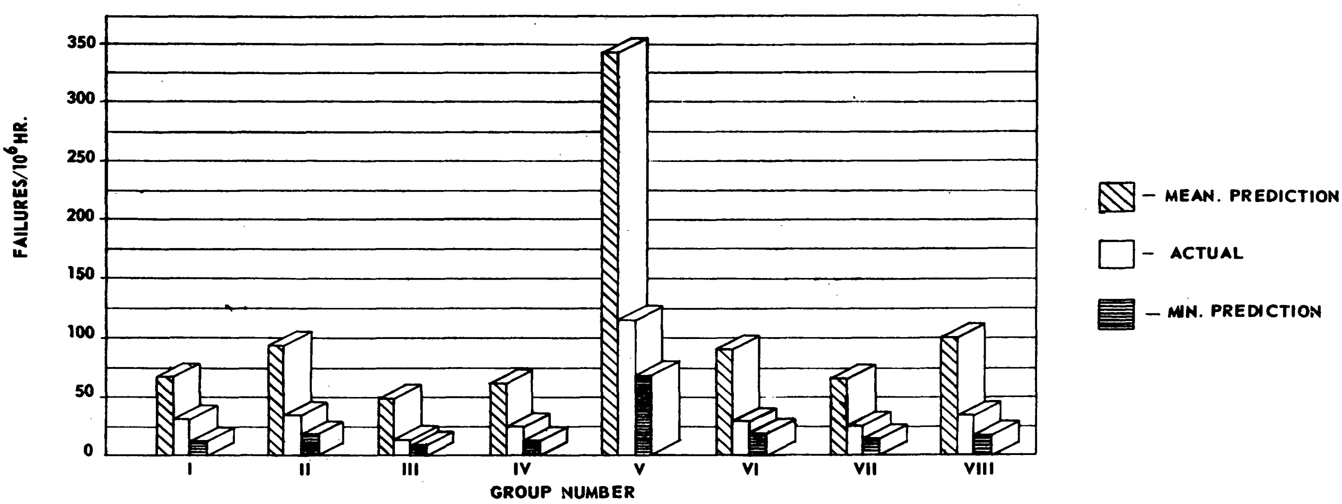


Figure 17.

the 5 x 4 SK and KSK pumps are all well designed pumps for the environments they are operating in because the actual pump failure rates are quite close to the minimum predicted. This is not surprising since these pumps are the evolution of a pump which was initially designed in 1913. However, today a design must be developed much quicker and the key to our ability to speed up design is an active reliability program with data feedback and a corrective action program.

Fig. 17 bears out the fact that the combinations of the generic failure rate, application factor (K_A), operating factor, (K_{OP}), and the design group factor (K_G) were chosen well since the actual failure rate is quite close to the minimum predicted. In addition it may be seen that the ranking of actual failure rates was predicted almost perfectly. From this it can be concluded that failure rate prediction can be a very valuable and extremely useful tool.

It also is apparent that the Navy pumps correspond to the advanced state-of-the-art. This was predicted while developing the Design Group Factor (K_G).

It is obvious from Fig. 17 that there is a wide band between the mean and minimum predicted field failure rates. Up to this time little work has been done on the reliability of mechanical systems and because of this the generic failure rates available for mechanical components may not be precise. Also, generic failure rates were not available for all mechanical components of the pumps and hence were estimated by using proper engineering judgment. Any error in the estimations will be reflected in the predictions. As time goes on and more work is done on the reliability of mechanical systems, failure rates for more mechanical components will become available, which will help improve prediction techniques.

Predicted failure rates and the use of the multiplying factors, K_A, K_{OP}, and K_G were based on the assumption that the pump would exhibit a constant failure rate. As shown by Fig. 16, this assumption is a close approximation where the overlapping wearout distributions of the individual components sum up to form a relatively constant failure rate curve (22). Therefore, based on Fig. 16 a constant failure rate can be assumed for all pumps in this study.

MANUFACTURER'S TOTAL COST

To show the reliability versus cost picture in a logical manner, it is necessary to reduce the cost figures to a base year to present unbiased comparisons. This eliminates the fluctuation caused by inflation, wages and market changes. The pumps under study experienced cost fluctuations because they were manufactured over a ten year span. If all the pumps had been manufactured one year, say 1954, the correction of cost to a base year would not have been required, consequently a "cost index" established, having 1954 as the base year, covered the period of 1953 to 1962. It was arrived at by plotting the specification cost of the pumps versus year manufactured and drawing an average curve through the points. Through the use of this "cost index" all costs have been reduced to a 1954 cost basis.

All costs in this study are relative cost reduced to the base year of 1954.

Determination of the Direct Product Cost

The direct product cost is defined as the direct cost of material and labor plus the manufacturing burden at predetermined burden rates, and may be referred to as specification cost.

The direct material cost is the cost of principal items of material required to make a product. Charges for material are made to the product at the time the material is issued through the use of material requisition tickets shown in Fig. 18. The direct labor cost is the cost of labor which is charged directly to the product. The document for this charge is the labor ticket shown in Fig. 19. The sum of charges against the product during its passage through the factory are accumulated on the form given in Fig. 20.

The manufacturing burden includes the following costs:

(a) The labor of personnel engaged in activities such as supervision, inspection, timestudy, etc.

(b) Indirect labor, such as handling of materials and supplies, electricians, janitors, trainees, standby or waiting time.

(c) Indirect materials, such as lubricants, paints, abrasives, welding and brazing wire.

(d) Maintenance and repairs.

(e) General expenses such as testing of materials and supplies, transferring of capital equipment, workmen compensation costs.

(f) Defective workmanship, material and errors.

(g) Allocated expenses such as water, light, heat and power, Social Security, insurance and vacation.

The sum of the direct material and labor costs plus the manufacturing burden give the part cost. The sum of the costs for all components gives the direct product or specification cost for a pump.

A list of the component parts of a bare pump for each group was submitted to the Accounting Department and the following is the procedure carried out to determine the product cost.

Orders for the pumps under study were selected from a complete list of customers' orders. This was compiled by the product sales department. Because of the Company's record retention policy, orders dated 1954 and after are available. The related detail material and labor ticket for orders dated 1957 and after, with the exception of government orders (1954) are also available.

Compilation of the direct product cost data required the shipping order, order specifications, and the tabulated cost report.

For flexibility in arranging and compiling costs through the use of Data Processing equipment available, a Part Cost card was prepared for each component

ALLIS-CHALMERS MATERIAL REQUISITION TICKET

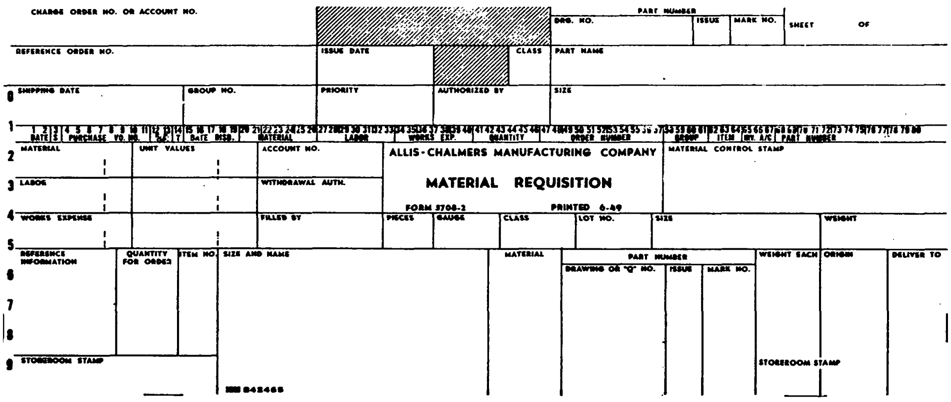


Figure 18.

ALLIS-CHALMERS LABOR TICKET

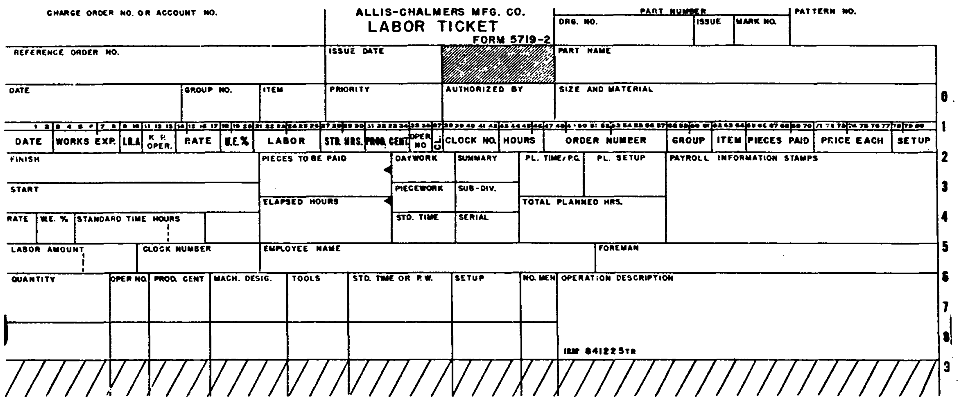


Figure 19.

ALLIS—CHALMERS MFG. CO.

COST REPORT

FORM 7586—1

QUANTITY							PART NUMBER					DATE	
GROUP	ITEM	PROD'N. CENTER	OPER.	CL.	QUANTITY	ELAPSED HOURS	MATERIAL		LABOR	BURDEN		SET—UP	ORDER NUMBER AND/OR SPEC. COST
							CODE	AMOUNT		MANUFACTURING	MATERIAL		

Figure 20.

part, including assembly costs. Data transcribed to the Part Cost card included the following:

Manufacturing date (from order specification).
Actual labor hours (make items and assembly labor)
Part number.
Order number.
Quantity.
Direct material cost.
Indirect labor cost.
Indirect manufacturing burden.

To the above data was added a group and item number to correspond with the group classification and part number.

If components within an order deviated from the standard bare pump unit, a cost for a standard component was submitted for the irregular component.

The Part Cost cards were keypunched and the unit cost calculated on a 1401 EDP data processing unit. The Part Cost cards were then mechanically sorted by part number and a listing by part number was tabulated for cost comparison purposes. The listing was reviewed for irregularities and cost fluctuations, and also

TABLE XV

ACTUAL MANUFACTURER'S RELATIVE COSTS FOR 5x4 SK AND KSK PUMPS

			Pump Groups							
No.	Manufacturer's Costs*	Units	I	II	III	IV	V	VI	VII	VIII
1	Spec. Cost (Matrl., labor, burden)	Relative to Base	3.78	3.78	4.22	4.22	12.26	11.93	3.57	3.57
2	R&D Eng. (For Improvement)	Spec. %	1.8	1.8	1.8	1.8	1.8	1.8	1.8	1.8
3	Net Eng. Expense (Product Section)	Spec. %	5.9	5.9	5.9	5.9	6.2	6.2	5.9	5.9
4	Eng. Change During Mfg.	Spec. %	.4	.4	.4	.4	.4	.4	.4	.4
5	New Patterns (Repair + Storage)	Spec. %	2.9	2.0	2.0	2.0	1.0	1.0	2.0	2.0
6	Small Tools (Drills, Fixtures, etc.)	Spec. %	.5	.5	.5	.5	.4	.4	.5	.5
7	Adjusted Mfg. Burden (Correction)	Spec. %	3.9	3.9	4.6	4.6	9.8	8.8	3.6	3.6
8	Shipping Expense (Crate + Labor)	Spec. %	3.0	3.0	3.0	3.0	4.0	4.0	3.0	3.0
9	Miscellaneous (Variance Costs)	Spec. %	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
10	Total % of Spec. Cost	in %	19.0	19.0	19.7	19.7	25.1	24.1	18.7	18.7
11	Total Relative Costs Added to Spec. Cost	Relative To Base	0.72	0.72	0.83	0.83	3.07	2.88	0.67	0.67
12	Manufacturer's Total Relative before shipment costs	Relative To Base	4.50	4.50	5.05	5.05	15.33	14.81	4.24	4.24
13	After Shipment Costs (No-Charge or Goodwill and Warranty Costs)	Spec. %	1.6	2.1	2.6	0.5	0	0	0.2	3.8
14	Manufacturer's Total Relative After Shipment Costs	Relative To Base	0.061	0.080	0.067	0.021	0	0	0.007	0.136
15	Manufacturer's Total Cost	Relative To Base	4.561	4.580	4.117	4.071	15.33	14.81	4.247	4.377
16	Manufacturer's Selling Price**	Relative To Base	4.87	4.89	5.47	5.42	16.40	15.85	4.53	4.68

*Do not include costs which do not affect, or are not affected by, reliability.
**With 7% Fee as accepted by the Department of Defense.

for quantity irregularities such as spare parts supplied with the bare pump components for Navy pumps. The specification cost was found by summing the unit cost on Part Cost cards for each pump. The specification costs given in Table XV for each group is the average cost of all pumps in the group.

Another factor which often comes into the specification cost of a low volume product is whether it is a "stock" or a "make" item. "Stock" items are made in relatively large quantities at one time, thereby gaining the "mass production" cost advantage, whereas "make" items are manufactured only as required which increases cost because of additional machine setups, etc.

For the pumps in this study the commercial pumps are primarily "stock" items and the Navy pumps are "make" items. However, by comparing the cost of Navy pumps ordered individually against those ordered in groups of 14 at a time for Group V pumps and 8 at a time for Group VI pumps, it was found that a quantity order resulted in considerable cost decrease. However, since parts on the commercial pumps are rarely made for stock in quantities larger than thirty (30) items, even less for major parts, it was concluded that the Navy and commercial pumps costs were equally benefitted by the "mass production" factor. Consequently no quantity cost correction factor was required.

Other Manufacturing Costs

Certain manufacturing costs are often not accounted for in the direct product costs. Whether or not they are depends on the customer and the order. Generally, the costs spelled out below are accumulated for the entire Pump Section. Detailed charges against one size and type of pump, like the 5 x 4 SK, are not available. Instead the particular cost for all pumps sold, of all models and sizes, is known. In order to isolate these costs for each group of pumps in this study, a detailed investigation of each cost was made. The values determined as a percent of the specification cost are given in Table XV.

Engineering Expense

Engineering expense is the cost of the engineering required at the time of the order. For the Navy pumps the expense had been charged against the order and was available. Engineering expense for the commercial pumps under study was not charged against the orders. By consulting with the engineers who worked on the orders, an estimate of the cost was arrived at. The actual expenses for the Navy pumps and the average expense for all pumps served as guides. For commercial pumps a value of 5.9% of the direct product cost was found and for Navy pumps a value of 6.2%.

Research and Development

To determine the research and development cost for product improvement engineers responsible for the various pump designs were approached with this problem. Also, pump cost specialists were consulted and an estimate was obtained for all groups. The basis of the estimate was the average value expended for all pumps in 1954. An estimate of 1.8% of the direct product cost for each group was arrived at.

Engineering Changes

Engineering changes during manufacture are largely due to changes in customer's requirements.

Estimates of this cost were obtained from engineers and personnel in manufacturing. Using the 1954 average for all pumps as a guide, it was decided that a value of 0.44% of the direct product cost for all groups was realistic.

New Patterns

The costs for new patterns, flasks and chills, with their repair and storage costs, were higher for the Navy pumps than for the commercial pumps. It was estimated that for Navy pumps the cost was 1.0% of the direct product cost. For commercial pumps the expense was 2% of the direct product cost.

Small Tools Expense

From a survey of tooling requirements for the 5 x 4 SK pumps, it was determined that both the Navy and commercial pumps could be built using equal expenditures for tooling. Brass is used in the Navy pumps for many parts, whereas cast iron is used in the commercial pumps. Since brass is more detrimental to tooling than cast iron the tool repair costs for the Navy pumps are higher. The expenses were estimated as 0.40% and 9.50% of the direct product cost for the Navy and commercial pumps, respectively.

Adjustment of Manufacturing Burden

Adjustment of manufacturing burden is required since the burden is applied at predetermined rates on direct labor dollars. The over or under applied burden must be considered as an additional cost to the product.

It was determined by the Allis-Chalmers Works Accounting Department that for pumps this adjustment averages +5.97% of the direct product cost for pumps.

To use an average value for all pumps means that the value will be overestimated for low cost pumps and underestimated for higher priced pumps. Therefore, to determine the percentage to add to the product cost, the average manufacturing burden was determined for each group. Then the 5.97% was corrected for each group in proportion to the amount a group's manufacturing burden deviated from the average. In this manner the average percent adjustment for all groups is 5.97% but the adjustment for each group is different.

Shipping Expense

Shipping expenses involved in the preparation of a pump for transporting it to the customer is part of the manufacturer's cost. This cost occurs after the pump has been manufactured and is not included in the specification cost which is to account for the direct material, labor and burden of making the pump.

For the pumps in this study, it costs 4.25 times as much to prepare a Navy pump for shipment as it does for a commercial pump. This amounted to 3% of the specification cost for commercial and 4% for Navy pumps.

Miscellaneous Costs

Miscellaneous costs incurred are such entries as provision for inventory, material received, finished stock variance, etc. These costs are figured at 1.5% of the specification cost.

Other Costs

Selling and administrative expense are approximately the same dollar expenditure for all pumps. The cost of selling a pump to a commercial or Navy customer would be approximately equal. Therefore, this expense is not shown in Table XVI since it will not change the optimum of reliability but will only move the curve upward on the cost axis.

Transportation cost on the shipment to the customer is incurred by the manufacturer. These costs may be for rail or truck transportation of the pump. Since the pumps in this study are approximately the same weight, the cost would be a constant for any single destination. The cost of transporting the pump could affect reliability if the method used was poor and the pump was damaged in transporting it. Failures may occur as a result of damage during transportation. However, we are assuming correct techniques were used during transportation, hence, this cost is the same for all groups.

Determination of After Shipment Costs

After shipment costs to the manufacturer for the pumps were made up of warranty and good will charges. These costs are accumulated by order and pump serial by the product department. Retention of the detailed costs made them available back to 1956. These costs were tabulated for each of the eight groups. The sum of these after shipment costs are shown as a percent of the specification cost for each group in Table XV.

MANUFACTURER'S TOTAL COST VERSUS PUMP RELIABILITY PICTURE

The manufacturer's total, relative cost is shown plotted versus the actual pump reliability in Curve C, Fig. 21, as the sum of the manufacturer's cost before shipment, Curve A, and the manufacturer's after shipment cost, Curve B. Curve D is the manufacturer's selling price which is arbitrarily 7% greater than the manufacturer's total cost.

The optimum level of reliability for minimum total cost occurs in a range of 250 to 275 fr/10^6 hr. for the manufacturer.

To obtain Fig. 21, the failure rates had to be adjusted to a common application and operation environment to permit comparison. All are adjusted to a shipboard environment pumping sea water.

The factors which are used to correct for environmental conditions are the Application Factor (K_A) and Operating Factor (K_{OP}). Therefore, to determine the failure rate of any group in a shipboard environment the following equation is applicable.

$$\lambda \text{ shipboard} = \frac{\lambda \text{ Group i } (K_A K_{OP}) \text{ Group V}}{(K_A K_{OP}) \text{ Group i}}$$

Results are tabulated in Table XVI. On Fig. 21, in addition to the adjusted group failure rates, the MTBF is also given.

As anticipated, the manufacturer's total cost before shipment does not increase very rapidly at lower levels of reliability. Increase is at a very rapid rate after a certain level is reached. Considering the pumps involved in Fig. 21, mainly commercial units making up the left portion and the Navy units the right, it appears that the sudden increase began when the state-of-the-art was being advanced.

Ordinates for Group III on Fig. 21 are the points in the lower righthand corner. Since the pumps in Group III are very similar to Group IV, only difference being a smaller impeller diameter, it is felt that the failure rate ordinate is in error, and that it is actually slightly less than that of the pumps in Group IV.

For comparative purposes the manufacturer's selling price must be prorated

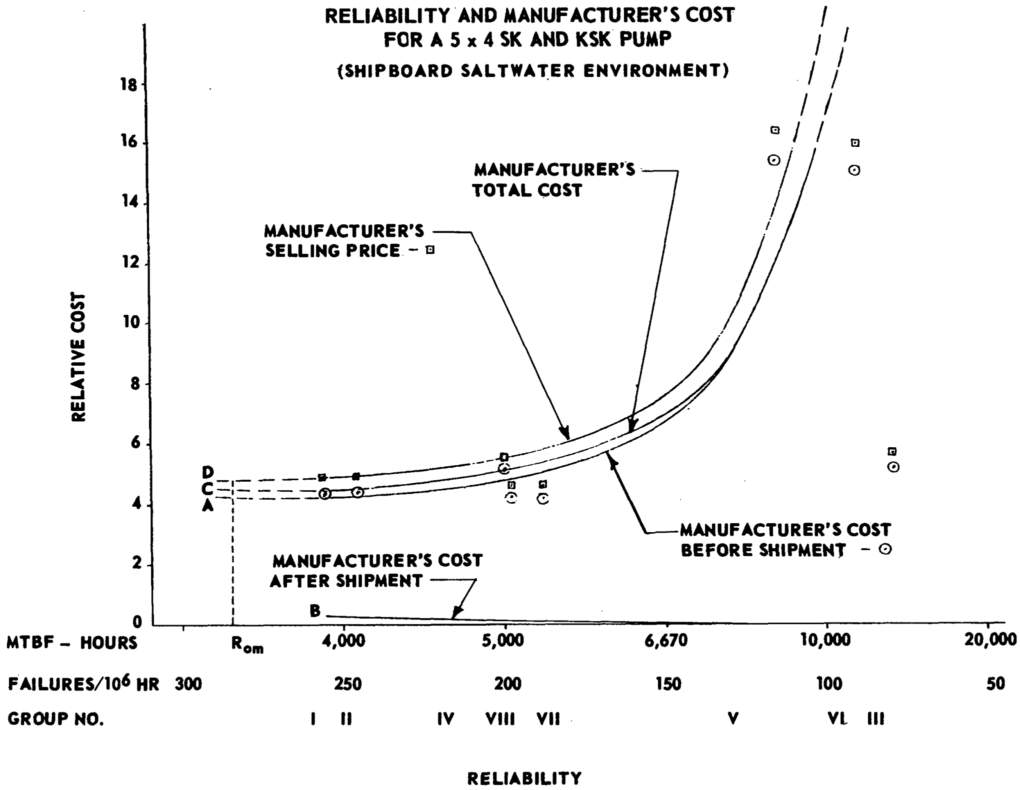


Figure 21.

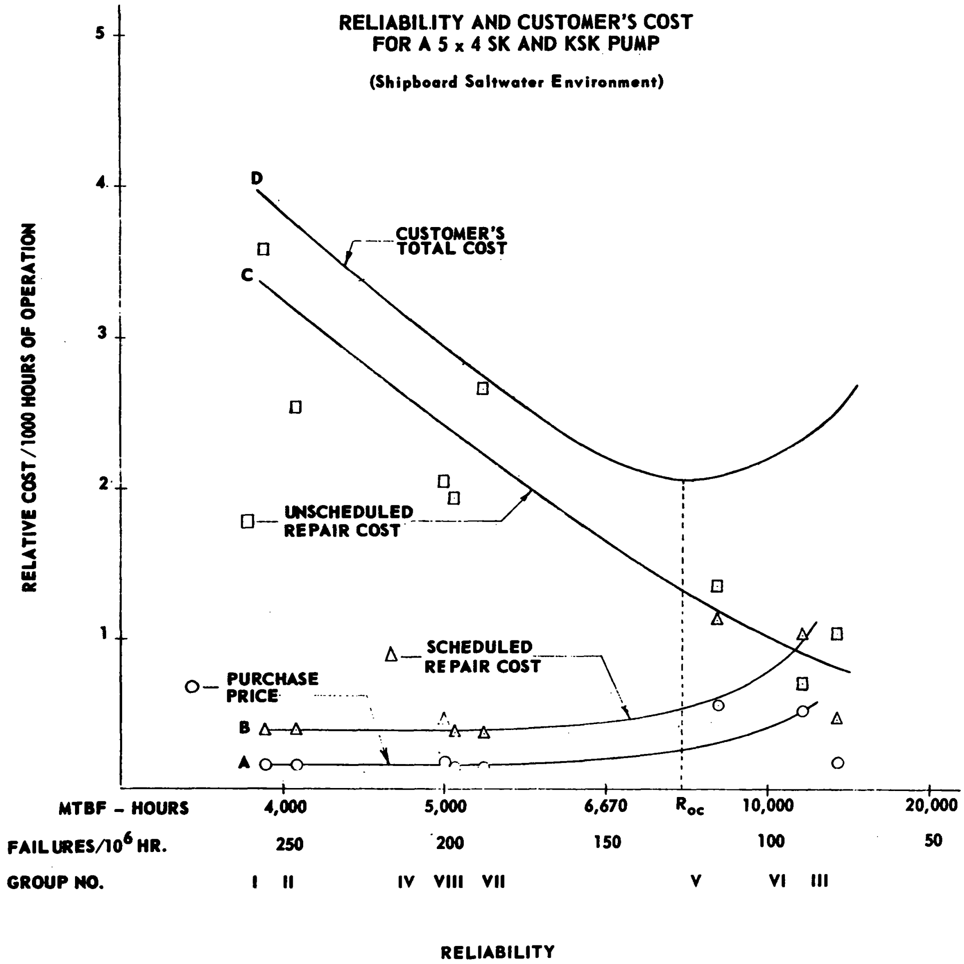


Figure 22.

TABLE XVI

ADJUSTED GROUP FAILURE RATES TO SHIPBOARD SEAWATER ENVIRONMENT

GROUP NO.	$\frac{(K_A\ K_{OP})\text{GROUP } 5}{(K_A\ K_{OP})\text{GROUP } i}$	ADJUSTED FAILURE RATE
I	8.05	256
II	7.50	246
III	9.34	79
IV	8.54	200
V*	1.00	115
VI	2.99	89
VII	7.76	188
VIII	5.90	197

*$K_A\ K_{OP}$)GROUP 5 = 61.3

for equal pump operating periods for the pumps in all groups. A pump does not have a well defined life. Parts can be readily replaced indefinitely and the pump will not reach a worn out condition even though every part in the pump will probably have to be replaced to prevent it. Therefore, the pump's life is the period before its design or application becomes obsolete. Considering the pump applications involved in this study, a life of 30,000 hours was chosen or about 10 years of average operation. (8-hours/day.) In Table XVII the initial cost per 1000 hours of operation is given and the values are also plotted in Curve A of Fig. 22.

For the pumps the after shipment costs are negligible in comparison to the before shipment costs. Since these pumps have such a comprehensive design background, it can be realized that early failures due to faulty design and manufacture are not likely to occur, and warranty costs would be low.

The largest factor in the pumps' before shipment cost is the direct product cost. For the commercial pump groups, Groups I, II, III, IV, VII, and VIII, this cost is made up in the following manner:

TABLE XVII

PRORATED PUMP INITIAL COST

GROUP	INITIAL COST/1000 HOURS
I	0.162
II	0.163
III	0.182
IV	0.181
V	0.547
VI	0.528
VII	0.151
VIII	0.156

Direct Materials	35-45%
Direct Labor	15-20%
Manufacturing Burden (Facilities)	35-45%

For the Navy pump groups, Groups V and VI, the breakdown shows the following pattern:

Direct Materials	50-55%
Direct Labor	12-15%
Manufacturing Burden (Facilities)	30-35%

CUSTOMER'S COSTS

Costs obtained for the customer as outlined in "Customer Reliability versus Total Product Cost Picture" and in "Data Sources Acquisition" are presented here.

The customer's purchase price for the pump is arrived at by adding to the manufacturer's total cost, shown in Table XV, an arbitrary fee of 7%. The customer's purchase price is tabulated for each group in Table XV.

The next cost to be considered is the repair and replacement expense of unscheduled outages or failures. All failures had been identified in advance of the failure rate calculation. Parts replaced during an unscheduled outage, as shown by the Machinery History Card, Centrifugal Pump Reliability Report or Fluid Dynamics Renewal Parts Section Records, were listed for each group. The total number of each part used during unscheduled repairs was found. The cost of each of these parts based on 1954 prices was obtained from the Fluid Dynamics Renewal Parts Section. The total part cost was obtained by summing the product of the part cost and the number of parts used for all parts. The cost of gaskets, bolts and nuts was considered negligible. Replacement of drain valves and the associated piping on the Navy pumps was not considered since they are not part of the pump being studied.

When repairs were made, these costs were estimated by using the detailed part costs as determined by the Accounting Department. For example, if a shaft was built up and remachined, the cost was estimated as equal to the initial manufacturing machining cost. Cost of replacement and repair of parts due to failures is tabulated on Table XVIII, designated as "Unscheduled Replaced Parts."

TABLE XVIII

CUSTOMER'S RELATIVE COSTS FOR 5x4 SK and KSK PUMPS

Cost Item	I	II	III	GROUP NO. IV	V	VI	VII	VIII
Purchase Price*	4.87	4.89	5.47	5.42	16.42	15.85	4.53	4.68
Unscheduled Replaced Parts**	0.0627	0.0398	0.015	0.0223	0.274	0.0219	0.0451	0.0488
Unscheduled Labor**	0.0192	0.0149	0.0049	0.0111	0.053	0.0107	0.0151	0.0141
Scheduled Replaced Parts**	0.0076	0.0190	0.0265	0.0019	1.02	0.8810	0	0.0111
Scheduled Labor**	0.0143	0.0104	0.0065	0.0015	0.094	0.1771	0	0.0094
Unsheduled Downtime Cost **	0.362	0.280	0.0925	0.209	0.995	0.201	0.283	0.264

* With 7% Fee as accepted by the Department of Defense

** Relative cost per pump per 1000 hour of operation

Labor cost for the removal and replacement of parts was determined through the use of a teardown and assembly chart. This chart was constructed using Time-study data obtained for pumps very similar to those under study. In addition time was spent in the pump assembly area observing the assembly of 5 x 4 SK pumps. Personnel in the Service Section of Fluid Dynamics reviewed the teardown and assembly chart and found it consistent with service experience. Using the chart the man hours required to complete all of the unscheduled repairs as indicated by the data forms were computed. By multiplying these by an hourly rate, the "Unscheduled Labor" costs on Table XVIII were found. The labor rate used in this study is \$5.32 per hour. This value is the average wage paid to Allis-Chalmers production employees in 1954 with a 120% burden added. The burden rate was determined as the average burden of several machinists, pipefitters and other maintenance groups within Allis-Chalmers.

By multiplying the unscheduled labor hours by the downtime rate \$100 per hour the "Unscheduled Downtime Cost" was found. This is also tabulated in Table XVIII for all groups. Calculation of the downtime cost in this manner assumes that one man repairs the pump. Then the repair time and downtime due to repair are the same. This approximation is satisfactory for the size of pump under study.

The "Scheduled Replaced Parts" and "Scheduled Labor" costs were determined in the same manner as the "Unscheduled Replaced" and "Unscheduled Labor" costs respectively.

The customer's cost must be compared based on pump operation in the same environment. As was with the failure rates, all the customer's costs are determined as if all the groups were operating on shipboard pumping sea water. Costs to the customer have to be compared based on a finite period of time, i.e., per hour or per year. In this study the time period was taken as 1000 hours. Table XVII shows the purchase price of a pump in each group prorated over 30,000 hours.

The customer's unscheduled costs, labor, parts and downtime can be adjusted to common base in the same manner that the failure rates were, since these costs are a function of the failure rate only.

It is assumed in this study that all the customers have followed the customer's preventive maintenance recommendations for these pumps. And in the case of all the pumps under study the recommendations are the same, therefore T, the period between scheduled repairs can be assumed equal for all the pumps operating in the same environment. These pumps do not vary drastically in the number of parts used, and the parts which wear out frequently are the same. Cost of the parts replaced vary approximately in the same proportion as the purchase price of the pumps.

Consequently, based upon the above reasoning, in the calculation of the preventive maintenance labor cost the same value, that of Group V, was used for all groups for comparative purposes. Cost of preventive maintenance parts was adjusted to the shipboard, sea water environment by taking the cost of preventive maintenance or scheduled replaced parts for Group V and multiplying it by the ratio of each group's purchase price to that of the pumps in Group V.

Other costs for the customer, such as installation, floor space and operating costs (cost of power) do not affect, and are not affected by, reliability. Therefore, they do not have any effect on the optimum reliability-minimum cost picture.

CUSTOMER'S TOTAL COST VERSUS PUMP RELIABILITY PICTURE

The customer's total costs for all groups are tabulated in Table XIX for the pumps, operating in a shipboard sea water environment, per 1000 hours of operation, and are plotted versus failure rate and MTBF. See Fig. 22.

The total cost picture shows that the optimum reliability is in the range of 125-145 fr/10^6 hours.

TABLE XIX

ADJUSTED CUSTOMER'S RELATIVE COSTS* FOR 5x4 SK and KSK PUMPS

Cost Item	GROUP NO. I	II	III	IV	V	VI	VII	VIII
1. Purchase Price	0.162	0.163	0.182	0.181	0.547	0.528	0.151	0.156
2. Unscheduled Replaced Parts	0.505	0.300	0.140	0.191	0.274	0.065	0.350	0.289
3. Unscheduled Labor	0.155	0.112	0.046	0.095	0.053	0.032	0.117	0.083
4. Unscheduled Downtime Cost	2.92	2.11	0.865	1.78	0.995	0.602	2.20	1.56
Total Unscheduled Repair Costs	3.580	2.522	1.051	2.066	1.322	0.699	2.667	1.932
5. Scheduled Replaced Parts	0.302	0.304	0.340	0.339	1.02	0.96	0.282	0.291
6. Scheduled Labor	0.094	0.094	0.094	0.094	0.094	0.094	0.094	0.094
Total Scheduled Repair Cost	0.396	0.398	0.434	0.433	1.114	1.054	0.376	0.385
Total Customer Cost	4.138	3.083	1.667	2.680	2.983	2.281	3.194	2.473

* Relative cost per 1000 hours of pump operation

In Fig. 22, the pumps' purchase price per 1000 hours of operation increases with reliability and is only a fraction of the customer's total cost. The predominating cost is the unscheduled repair cost. Major contributing component in it is the unscheduled downtime cost. As seen in Table XIX the unscheduled and scheduled repair costs are approximately the same in most cases.

If the unscheduled downtime cost is not considered, as it may not be for Navy applications, then the optimum level shifts to the left to a failure rate of 165-175 fr/10^6 hours. This can be seen in Fig. 23.

Of the three major cost components shown in Fig. 22, the initial purchase price is the lowest. This curve is based on pump lives of 30,000 hours. If the life chosen is too long, the optimum point will slightly shift to the left. Pump designs with higher failure rates and lower initial cost will then be optimum. It is more likely though that the lives of these pumps were chosen too short, since aboard one aircraft carrier they have already accumulated 15,000 hours in approximately eight years. Also some commercial units have accumulated more than the 30,000 hours in less than ten years of operation. Pump life could be defined in other ways, such as when the casing or impeller is worn out, or the time to first failure. It is felt that using a life of 30,000 hours would most closely fit the life of these pumps from the customer's point of view.

The scheduled repair costs are higher than the prorated initial cost of the pump. For Group V, Navy pumps, the dollars spent are nearly twice that of the prorated initial cost per 1000 hours of operation. For both groups of Navy pumps, Groups V and VI, the cost of scheduled repairs is several times the cost of unscheduled repairs, without including downtime cost. This could be the result of the following two factors: (1) Unscheduled downtime of these pumps is not tolerable and perhaps the $100 per hour downtime cost used in this study is a good estimate for the Navy. (2) The pumps are over-maintained to provide experience for maintenance crews.

In other words, unscheduled downtime cost is one of the major factors which make frequent scheduled repairs of the pumps necessary.

Until the Navy and commercial customers place values, in terms of dollars, manhours, or relative to other equipment, on the unscheduled downtime of equipment, it will be difficult for manufacturers to produce equipment with optimum reliability. Also, the manufacturer will be stymied in attempting to provide

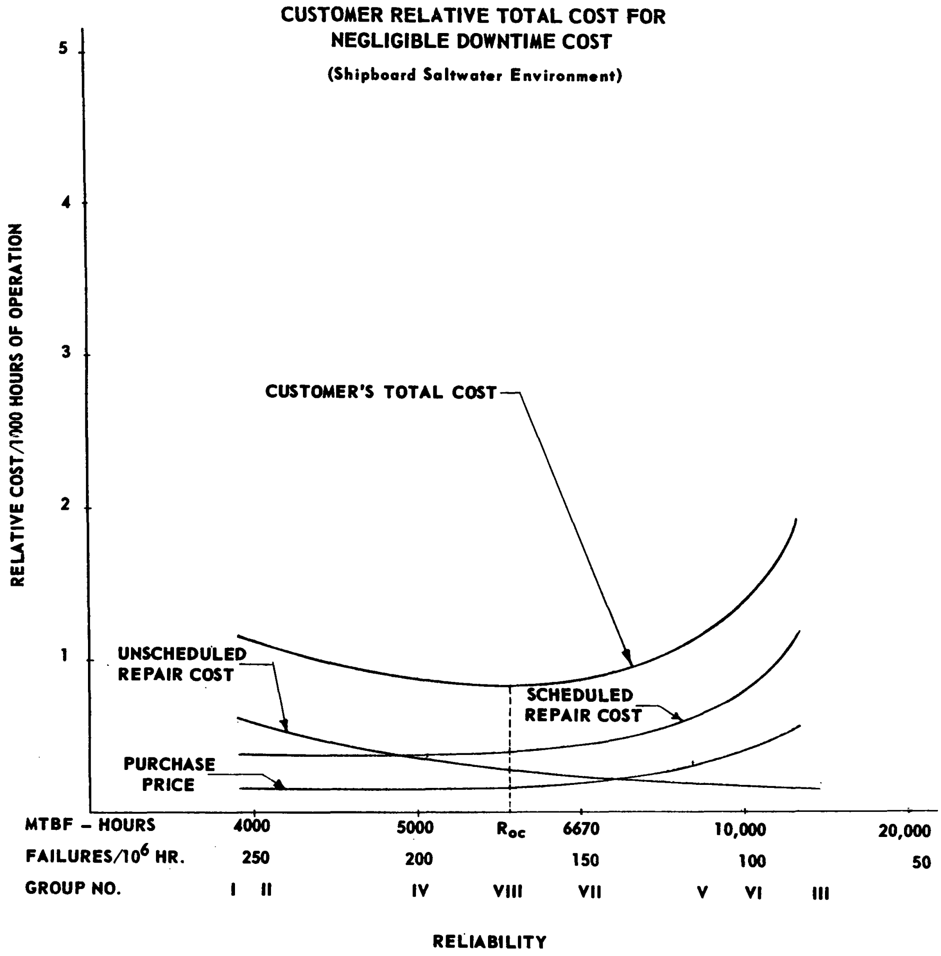


Figure 23.

engineering help on problems in redundant equipment, spare provisioning, maintenance requirements and manning needs.

It is fully realized that the seriousness of equipment downtime varies with the operational mode of the equipment and the ship. This is a multivariable problem which can be handled with probability and engineering analysis.

The major cost on Fig. 22 is the unscheduled repair cost. The primary component in this cost is the unscheduled downtime cost. If this value is zero then the customer's cost picture is changed. (See Fig. 23.) With this change the optimum level of reliability shifts to the left. The optimum value from Fig. 23 is in the range of 165 to 175 fr/10^6 hr. Unless a pump is doing a menial task, it is unlikely that the unscheduled downtime cost will be negligible. The following four considerations suggest possible incurred costs:

1. If standby equipment is kept in readiness, then the cost of the standby equipment is chargeable as unscheduled downtime cost.
2. If other equipment is kept from operating, a loss is involved.
3. If manpower is put on waiting, productivity is lost.
4. If a service being performed by the equipment stops, again productivity is lost.

The degree of importance in each case obviously may be different and varies depending upon the equipment.

CONCLUSIONS

The results of this study permit the following conclusions:

1. Fig. 22 shows that there is an optimum reliability level at which the total cost of these pumps to the customer and the manufacturer is minimum. This level is a function of the pump price, parts cost, random and wearout failure rates, part and pump life, environment, and maintenance practices.

2. For the customer the optimum pump failure rate is in the range of 125-145 fr/10^6 hr., with a mean of 135 fr/10^6 hr. For the manufacturer it is in the range of 250-275 fr/10^6 hr. with at about 263 fr/10^6 hr.

3. The optimum pump failure rate of 135 fr/10^6 hr. is very close to the failure rate for Navy pumps of 115 fr/10^6 hr. (Group V.)

4. Fig. 22 shows that the support costs for these pumps is several times the purchase price. Even if the pumps are not penalized for unscheduled downtime costs, the support cost would still be several times the initial cost. However, for land installation for use with clear water, the support cost would be only a small fraction of the initial cost as may be seen in Table XVIII. This points out the great significance of the pump application and operating environment on its reliability and cost, as a result of which the optimum level of reliability will change. The manufacturer should therefore obtain an exact description of the environment in which the pump is going to operate before recommending the optimum pump for that application.

5. It became apparent during this study that neither the customer nor the manufacturer kept adequate, easily retrievable records of either costs or pump performance data. Efforts are being made in this area, but most are in their infancy. The Navy's Machinery History Card, when properly filled in, produces very useful data, but often the very important hours of operating are not given. The Navy's failure reporting program is of an exceptional nature in concept but has not been successful in getting the failures reported. The commercial customers often do not keep detailed enough records of how, when and why of problems.

6. The manufacturer and customer should not only optimize reliability with respect to reliability, but also consider the optimum preventive maintenance practice. This is a multi-variable problem on which more work needs to be done.

7. The random failure rate of the pumps studied is negligible, and the failure rates are governed more by the mean wearout life of the components (17). Therefore, in order to increase the reliability of any of the pumps the individual part mean lives must be increased. It is important to note that although the failure rate of Group V is approximately constant, the failure rates of the individual parts are not.

8. Fig. 21 corresponds to the righthand portion of the cost curves given in Fig. 1. The pumps are all designed at low enough stress levels and, furthermore, the manufacturing processes involved are well enough developed that extremely few early failures or random failures occur.

9. Fig. 22 corresponds more to the lefthand portion of the cost curves presented in Fig. 2. Here the support cost is much greater than the initial price and the optimum reliability for the customer is considerably to the right of that for the manufacturer. The manufacturer, for his own benefit in maintaining good customer relations, should design the pump at the customer's optimum to

minimize the customer's support costs. As it may be seen by the proximity of the Group V, Navy pump failure rate to that of the optimum in Fig. 22, Allis-Chalmers already has accomplished this.

10. The cost of scheduled and unscheduled repairs for each group is approximately twice the pump purchase price on a cost per 1000 hours of operation basis. The unscheduled repair and scheduled labor costs can be reduced by using parts with longer lives, however these parts will definitely cost more; and therefore the purchase price and scheduled repair parts cost will increase.

11. Fig. 22 indicates that a total cost savings of 45% can be made by spending only about 40% more at the time of the purchase. This is arrived at by comparing the total cost and purchase price values of Group I pump with those of the optimum pump at R_{OC}.

RECOMMENDATIONS TO THE MANUFACTURER

The efforts, procedures, data acquisition techniques, the quality of data obtained, methods of analysis of this data, and the results of this study bring forth the following recommendations to the manufacturer:

1. A concerted effort needs to be expended in improving the following documents:

   (a) A number of Product Reliability Forms should be developed for each major product line. These forms should be complete and when properly filled out, should provide information from which early life, useful life and wearout failure rates can be calculated, and customer's support costs can be obtained.

   (b) The accounting forms should enable the accumulation of the specific cost items that make up the manufacturer's before and after shipment costs on the basis of a specific product in a department rather than on a product department basis where more than one and varied products are involved.

2. All necessary steps should be taken to motivate all disciplines involved to objectively compile the required reliability and cost data.

3. The failure rates and all costs should be calculated at frequent enough intervals to enable their monitoring and the establishment of the optimum level of failure rate for minimum total cost.

4. An increasing effort should be expended to attain and maintain the optimum level of reliability for a specific product.

5. An integrated reliability program should be implemented, to make everyone that deals with a product from birth to death conscious of the existence of an optimum reliability goal for each product and to educate them in the science of reliability so that they can design and build the optimum target reliability into the product.

6. Preventive and repair maintenance schedules should be scientifically worked out by the manufacturer based on the product's reliability bathtub curve so that these maintenance costs, which most frequently are far greater than the purchase price of the product, are minimized over its life.

7. A product's operation and maintenance manual should be prepared with the optimum reliability in mind. The maintenance schedule should include the groups of components that should be replaced at each scheduled maintenance.

8. When bidding on a request for a proposal, an extraordinary effort should be expended to quote on the customer's total cost, as well as on the initial cost

basis. The manufacturer should emphasize in his proposal that he has expended the effort of developing the customer's total product cost, and that the customer should base his selection of the successful bidder on this total cost basis.

9. Enough significant failure rate data should be obtained to determine for which components development and component selection money should be expended. These components would have relatively high failure rates or be responsible for a major portion of the support costs.

10. Fig. 22 shows that the manufacturer is supplying a pump to the Navy that has a failure rate (115 fr/10^6 hr.) very close to the Navy's optimum or 135 fr/10^6 hr.

11. The manufacturer should provide design improvements that would minimize misalignment among the rotating and stationary pump assemblies and the drive motor. Customer data show that a large proportion of early failures are due to such misalignment.

12. Special tools should be developed for the user so that pump bearings can be assembled by Navy maintenance personnel with minimum of cocking in their seat. Furthermore, an identification should be provided so that these bearings would not be inserted wrong face in into their respective bores. Many bearing failures, and most of them after only 300 to 400 hours of operation, are due to improper assembly practices.

13. The manufacturer should use, and to great advantage, the pump field failure rate prediction technique developed in this study for new pump designs to be introduced in the future. Through this technique, the engineering of a product may be done in advance of hardware availability, thus permitting an early trade off analysis to optimize component design and selection.

14. The Navy pump bathtub curve of Fig. 16 should be used to determine the spare part requirements and provisioning schedules, as the area under the curve is equal to the failures for which spare parts are required. Using techniques of pump failure rate apportionment to components, coupled with a consequential failure analysis, the spares required for each pump may be determined.

15. Fig. 16 indicates erratic preventive maintenance practices. This may be minimized by preparing a comprehensive life long preventive maintenance manual for the Navy. A close study of the Machinery History Cards should reveal the best preventive maintenance practice for each pump type on an optimized basis. Reference (17) should be used to accomplish this. This should reduce spare part requirements, the number of preventive maintenance actions and maintenance crews.

16. The bathtub curve for Group V indicates a high early failure rate. Since these failures have not been reported as the result of faulty workmanship or material on the part of the manufacturer, they must have been the result of incorrect installation, maintenance or an abnormal environment such as pumping sand through the pump. Because of this, the manufacturer should maintain close liaison with the customer to isolate and solve this problem.

RECOMMENDATIONS TO THE CUSTOMER

1. The customer should in the near future, require that costs in all proposals be submitted on the customer's total cost basis.

2. The customer should request that all proposals contain total cost versus useful life reliability versus total maintenance cost curves based on several preventive maintenance schedules, for total cost optimization.
3. The customer should learn how to incorporate the optimum reliability level into its technical specifications, how to seek it and how to monitor it.
4. The customer should vigorously pursue a practice of fully documenting all of the pertinent reliability, cost and maintenance data during the life of a product and of making available this data to the manufacturer for his analysis and action.
5. The customer should incorporate in his procurement document clauses for rewarding the manufacturer upon attainment of the optimum reliability goals and for penalizing him on default.
6. The bathtub curve of Fig. 16 and a study of the entries on the Machinery History Cards indicates that much erratic maintenance is being performed on the Navy pumps. It is urged that the manufacturer's maintenance recommendations be dutifully followed, as much as is feasible, to minimize the maintenance cost and reduce maintenance crew requirements.
7. The maintenance crew should be better trained because the excessive frequency of maintenance performed indicates that misalignments and wrong component assembly practices during maintenance abound. Shaft breakages, too numerous bearing replacements, undue shaft sleeve replacements due to wearout can be minimized by better trained maintenance crews.
8. Fig. 22 shows that the Navy is being supplied by the manufacturer a pump very close to the optimum. It is recommended that the Navy continue procuring such pumps having a failure rate within the range of 125 and 145 fr/10^6 hr.
9. The Navy's Machinery History Cards are well conceived, however they are not being completely filled out. The most important bit of information, namely, hours in the life of the pump when a particular maintenance is performed, is very frequently missing. The date the pump was put into operation should be entered, as well as the exact observation that led the crew to decide to perform the particular maintenance.
10. A concerted effort should be expended to get back a greater proportion of the failures reported on the BuShips Failure Reports. Presently, only between 10 and 20% reporting is being achieved. These reports should be matched with entries on the Machinery History Cards for completeness and for cross-check on the efficiency of the failure reporting system used.
11. The Navy, as well as other customers, should use preventive and repair maintenance records to formulate optimum corrective action practices and feed back this information to the manufacturer for his perusal and preparation of operation and maintenance manuals.
12. The customer should motivate his personnel dealing with the product into observing all reliability practices recommended by the manufacturer and into using all specified reliability documentation media faithfully.
13. The customer should promote the development of bathtub curves and should use same, so that he optimizes his spare part procurement and storage requirements.
14. As the customer's support costs for these pumps are several times the purchase price, the customer should exercise stricter control over the application environment of these pumps because a slight increase in the severity of this

environment, such as sucking sand into the pumps while at port, increases the maintenance or support cost sharply.

RECOMMENDATIONS ON THE IMPROVEMENT OF THE QUALITY OF RELIABILITY AND COST DATA

Special efforts should be made to improve the quality of reliability and cost data. These should include the following:

1. Motivate every engineering, manufacturing, and cost center to compile complete reliability and cost data. The details of such data and specific problems were discussed in the respective cost sections.
2. All pertinent forms for the acquisition and processing of this data should be redeveloped and well integrated.
3. The use of such forms has to be put on an almost compulsory basis.
4. Special questions should be provided for certain classes of customers, such as the different segments of commercial and military customers.
5. The information sought should be recorded on specific forms developed for specific information, i.e., failure data on one form, spare parts on another, scheduled maintenance cost on another, etc. This would facilitate the collection of the required information.
6. The explanation of an entry on the form should be directly beneath the question asked. This explanation will vary, depending on the class of customer being questioned.
7. The customer should be made aware of how the information being requested is to be used. He should be given an opportunity to answer a question in his own way if this is permissible. As an example, the customer may want to state the cost in dollars or hours.
8. The operating environment should be more clearly explained by the customer. To get this information from him, the form will have to be more specific and apply directly to the industry in question.
9. The customer should be encouraged to send copies of his records along with the completed form. If the customer is assured that the records will be confidential, he may be willing to cooperate. These records will contain information overlooked by the customer as being applicable to reliability.
10. A checklist type of reply appears to be the most productive. In this study the most useful information was obtained where checkoff blocks were used.
11. Tolerances or ranges should be requested on specific data sought to give the customer the opportunity to check off the range most applicable to his case.
12. It would be productive to send the customer film strips, slides, posters, or literature prior to his filling out the form. These modes of communication would set the stage for better and more useful information.
13. The customer should be encouraged to write comments.
14. The questions should be numbered for electric machine card key punching and processing.
15. Repeat questions should be provided worded to differently provide a check on the previous information entered.

It should be realized that it is a never ending challenge to both the manufacturer and the customer to obtain, compile, properly document, and analyze reliability and cost data. The importance of this challenge should never be minimized. After all, progress is brought by analysis of facts which can best be presented in data form.

RECOMMENDATIONS FOR FUTURE STUDIES

1. Funds should be made available to conduct a similar study on other DOD products to determine how the optimum useful life and wearout life reliabilities shift.
2. Studies on cost optimization for the totality of useful life reliability, wearout life reliability, total inventory cost, (cost of product procurement and of possession) maintainability, availability and safety should be undertaken.
3. Studies to develop multivariable optimization techniques should be undertaken.
4. Funds should be made available to develop further the methodology of predicting mechanical system reliabilities.
5. Studies should be conducted to develop techniques for designing a specified failure rate directly into a product and its components.
6. More studies should be conducted to develop bathtub curves for mechanical subsystems and systems to see whether the prevailing useful life reliability emphasis has basis for the majority of such mechanical subsystems and systems.
7. Studies should be conducted to develop reliability checklists for engineering, manufacturing, quality control, sales, service, maintainability, and customer data feedback to help attain and maintain the optimum reliability level.
8. Optimum spare part provisioning techniques should be developed based on reliability bathtub curves.
9. Component and equipment design techniques should be developed whereby the "One-Horse Shay" concept is approached, if not attained, thus minimizing very costly preventive maintenance.
10. Studies to develop more effective and efficient failure reporting, data feedback, and corrective action procedures should be undertaken. The field failure and performance data should be properly identified and classified, failure rates calculated at regular intervals, coded and stored for easy retrieval. The time required to complete studies, such as this, may be drastically reduced in this manner.
11. Studies should be conducted to determine the effect of startup and shutdown on equipment reliability. This would help in the optimization of the total maintenance costs and in the design of components.
12. More studies should be conducted to determine the effects of various application and operating environments on component and system failure rates, to increase the accuracy of reliability predictions.

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DISCUSSION

HAUSCHILDT (Bureau of Ships): I wonder if you could give us a little information as to the difference between the Navy experience and the commercial experience in this regard. Is there a difference?

KECECIOGLU: Well, maybe I should ask you a question, what do you mean by experience?

HAUSCHILDT: You get certain data from the Navy and the data from commercial.

KECECIOGLU: Well, all this data appears in the paper. There are five commercial and two Navy pump groups. The two data points to the right are for the two Navy group pumps and the ones then to the left are the commercial pumps. The Navy pumps definitely have significantly higher reliabilities, as the figure shows. Does this answer your question?

STORY (Stal-Laval): You had a curve there that showed total customer cost. How do you determine this? Does this take into account the cost of related problems that come from the failure of the pump, or is this strictly a total cost of replacing the part?

KECECIOGLU: These are the total costs and include failure costs and downtime costs.

STORY: That would seem to be a rather difficult thing to arrive at — getting a customer to tell you what the actual cost is.

KECECIOGLU: It is very difficult to arrive at. I needn't go into the mechanics of getting this data and how many customers we had to sample; how many personal visits we had to make, how many district office managers we had to send to the customer to get the data from, time being limited. Quite a lot of it is in the paper. Suffice it to say that five months of intensive effort, of many district office personnel is involved. Does this answer your question? The problem is difficult, but the data is there if the proper effort is expended to get it and then analyze it properly.

STORY: How reliable do you think that curve is?

KECECIOGLU: Well, as reliable as you want it to be. The large population involved makes our data quite reliable.

HIRSCHKOWITZ (U.S. Merchant Marine Academy): In line with this question just asked, how would a merchant ship operator assess his downtime costs in the event that the failure of something that would involve a full time shut down occur? This is the frustrating part of the whole discussion.

KECECIOGLU: You are absolutely right, but it can be assessed as we had to compare these pumps. Now Roy why don't you answer how this cost was arrived at for the gentleman who asked the question. Mr. Hughes is the co-author.

HUGHES (Allis-Chalmers): The costs for downtime were arrived at by questioning several customers. We obtained quite a range of results — now these are commercial customers. We didn't ask the Navy because I don't think the Navy can give us an answer on what their down costs are. Commercial customers gave a range; from about $5.00 per hour to $1,500 per hour. If the cost per hour of downtime is too high, you should install a redundant pump, you have two pumps. If the cost is $500 an hour you can buy another pump with just a few hours downtime.

If the entire downtime cost is removed from this picture, the optimum point will shift one way or the other slightly. The downtime cost can be doubled and again it will shift the optimum point slightly, but it will still be in the range shown.

FRANKEL (M.I.T.): Just to complicate the matter a little bit more, I believe there is an important point with regard to customer costs. I'm afraid, actually to be correct, we have to put in a fourth coordinate called degradation of performance. This may not be important with regard to pumps, but it is of major importance in internal combustion engines, as Professor Lipson will probably agree, and although most people are probably reluctant to go into optimization studies of four coordinate systems, it can be accomplished among other methods by dynamic programming.

With regard to the other comment that the failure rate will actually converge on the 1/M rate in mechanical systems . . .

KECECIOGLU: But there is a trend towards that as shown in the figure of the "bathtub" curve.

FRANKEL: There is a trend, but I would like to make this qualifying statement that this trend will probably only be true if we assume after every overhaul, implying the renewal of many parts, an as-good-as-new system which, I believe, most of us will disagree with. Apart from it, it has to be better than as-good-as-new; it has to be as good as after the initial or infantile failure period.

BERMAN (Westinghouse): It seems your whole curve is based on these two high points, they are supposedly high cost Navy pumps for reliability. I wonder if you would elaborate on whether this high Navy cost is due to the fact that they are cast steel where the industrial low cost pumps are cast iron, so that really the cost difference isn't due to reliability but more to the Navy's combat shock requirement.

KECECIOGLU: Some of this is definitely true, yes. It is not the sole factor however. Dick Haugen (Allis-Chalmers man) would you care to comment on this?

HAUGEN (Allis-Chalmers): Basically the difference between the pumps used commercially and those used by the Navy are the stringent specifications required by the Navy. The Navy specifications, in general, require more inspection and more handling. Stock components of a pump cannot be used on a Navy pump because of the stringent requirements. This means that each component is made special without benefit of mass production. These all result in increased cost for Navy pumps.

KECECIOGLU: Including materials...

HAUGEN: The materials used in the construction of Navy pumps are especially suited for the salt water environment they are exposed to. The design is slightly different, giving a different level of reliability on the Navy pumps. The Navy pump employs two bearings back-to-back on both the inboard and the outboard end of the shaft. The shaft is of a different material and has a longer bearing span than the commercially used pumps which only use a single row bearing.

KECECIOGLU: Incidently, in these curves all pump groups have been reduced to a common application and operation environment to be comparable.

Commercial pumps, for example, obviously most of them were not aboard ship. Therefore they have all been corrected to the common environment of shipboard pumping sea water, as would be in Navy pump use.

BENFORD (Univ. of Michigan): I shall confine my remarks to a single criticism of Dr. Kececioglu's methodology, namely his neglect of the time value of money.

Money in the hand today is more valuable than the promise of a like amount next year. Therefore, future maintenance costs should be discounted before adding to invested costs. Furthermore, maintenance costs will normally increase over the years so that an optimum life study is an inherent, necessary first step.

Discount factors depend upon the rate of interest at which the organization in question values its money. For the government, this would be only about 3 percent. For a typical private firm it would be more like 20 percent (before taxes). We thus have the interesting complication that the government has one level of optimum reliability while private industry has another. The government's would be slightly lower than that proposed by Dr. Kececioglu; industry's would be considerably lower.

In conclusion, I would recommend that Dr. Kececioglu flavor his offering with a sprinkling of engineering economy.

KECECIOGLU: We concur with Professor Benford that the time value of money has not been included explicitly in our paper. However, it was difficult to determine the true discount for the manufacturer because of the wide variability of the time over which the discount accrued, and the difficulty of ascribing an applicable proportion of the discount to the individual cost items presented in Tables XV and XVIII and to those items not affected by reliability.

This does not mean that the interest cost was neglected. Such costs are distributed among the Allis-Chalmers product departments and then redistributed by the product department over its major cost components. Therefore, the time value of money is implicitly included. From the user's standpoint such costs make up overhead or burden rates and these were applied to the maintenance labor costs (120%).

Suppose however we arbitrarily superimpose the time value of money proposed by Professor Benford on Figs. 21 and 22. An optimum shape curve as shown has the property such that any constant percentage increase over the whole curve does not change the optimum value of the abscissa (reliability) but only raises the entire curve. This would be the case for the manufacturer. However, if different percentage increases are applied to each of the component curves and pump groups making up the optimum curve then the above is not valid. Such a condition probably does exist for the user since the discount factors applied to the money for the pump procurement may be different than that for money spent for repair parts at a later date. Furthermore these costs would be different for different users. However, the differences would not be too substantial and the effects on the curves would not be too significant.

We agree with Professor Benford that optimum life studies are of great importance. However, the term "life" should be discussed. A centrifugal pump and many other repairable products do not have a well defined life. Often they are repaired as needed; some of their components never completely "wear out," and eventually the product is scrapped because better, more efficient and economical units become available, or because the need for the unit disappears.

The centrifugal pump maintenance cost does increase during certain periods, however, in the long run these costs remain relatively constant or average out quite well. The shape of the bathtub curve, Fig. 16, tends to suggest this. Indeed, the variance in the preventive maintenance practice, the reliability of parts used, and the changes in labor and material costs due to changes in the cost of living, wage rates, overhead rates, affect the maintenance costs. These have been averaged by us over the selected 30 year life of these pumps.

It is true that the optimum point does change depending on the user not only because of interest and the above costs but also because the application, environment and operating conditions of the product change. All of these affect the reliability and the maintenance cost considerably. However, they are extremely difficult to isolate. We believe our averaging method has not significantly distorted the optimum failure rate and therby the optimum reliability levels we set out to determine.

In conclusion we believe that reliability-cost optimization is one of the pillars of engineering economy.

RAILROAD RELIABILITY AND MAINTENANCE PROCEDURES RELATED TO LOCOMOTIVE PROPULSION ENGINES

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Twenty-seven years of examining the maze of interrelated factors which influence the maintenance problems of railroad diesel engines leads to the view that our principal problem today is in getting the maintenance work done in the right way at the right time.

There was a time when the problems of metallurgy, design, fuels, lubricants, filtration, cooling water treatment, etc., were paramount and actually overshadowed the problem of how the maintenance work was done. But scientific advancement in these areas has led to a technology which so greatly improved the reliability of engines, as built, that the importance of the knowledge, skill and reliability of the maintainer is now the principal limiting factor in engine reliability.

The history of the railroad diesel has been one of limiting factors — first one, then another, as problems were solved. This is undoubtedly true of all things where advancement is being made. Speaking from memory, and starting with the first nonarticulated main line diesel locomotive engines, and the first (prototype) diesel switcher, the limiting factors in reliability occurred and were removed in the following order. Fuel oil was the first thing which took diesel engines out of service because of fuel system deposits and fuel system corrosion. When that limitation was removed piston rings were found to be the next limiting factor. Improvements in oil control rings, in compression ring breakage, scuffing and wear rates involved intensive experimentation and resulted in rings which outlasted the bearings. So the campaign for reliability turned to bearings and then to oil filters and air filters and so on thru all the parts of the engine, and then back thru most of these parts again and again as each in turn was made more reliable than the other. This involved also the fuels and the lubricants in the endless quest for improved reliability and more favorable maintenance costs. This spiral of improvement continues today.

The amount of work which went into this effort, and the range of talent which was utilized, is little recognized outside the groups which participated. In addition to their own unilateral research and development, the engine builders working with the engine users, including the Military, and with the fuel and lubricant suppliers, brought the talents of many companies, universities and private research organizations to bear on these maintenance problems as they arose. One of the most lucrative clearing houses for applied research and exchange of information was the Coordinating Research Council of the American Petroleum Institute and the Society of Automotive Engineers. Without its tremendous contribution, stemming from its ability to freely draw upon the experience of appropriate scientific and technical personnel from practically any field of specialization in this and other countries, the potential reliability of railroad diesel engines would be far less than it is today.

The point I am trying to make is that tremendous effort has been made at high scientific and technical level in the interest of engine reliability and economy in maintenance. The information so obtained and applied by the engine builders, oil companies and engine users now seems to have made the engine maintainer the limiting factor in engine reliability and in maintenance economy. It is my view that in this area we have not fully capitalized in maintenance shops on the technology which is available to us. But on the other hand in retrospect we have come a long way in reschooling steam locomotive maintainers in the extremely complicated job of maintaining the precision machine which we call a diesel engine. The maintenance schools which the engine builders conduct for the engine users supervisors, and the instructions which these supervisors have transmitted to the maintainers have resulted in increasing appreciation of the fine workmanship which is required in the maintenance shop in order to get the best out of this engine. As a result there has been, and will continue to be, steady improvement in maintenance and reliability of railroad diesels. I mention this because the transition from steam to diesel with the same maintenance personnel was and still is somewhat of a problem.

The prime movers of practically all American railroad locomotives are now diesel engines, which are directly connected to electrical generators, which transmit power to the wheels thru electric traction motors which are geared to the axles. A locomotive in railroad parlance consists of one or more "locomotive units," averaging about three, but sometimes considerably more, depending upon the work which is to be done.

It does not seem practical to discuss here all the types of diesel engines in locomotives as a group, nor to attempt to consider them separately. There are four cycle, two cycle, opposed piston, normally aspirated and supercharged engines in railroad service. For practical purposes it may suffice to confine our discussion to the freight locomotive engine, and to illustrate railroad maintenance and reliability of locomotive diesels in general by using one of the common types, assuming that, while there may be specific differences between types, it will serve to illustrate in general the reliability and maintenance procedures for all types of similar horsepower.

Our maintenance practices and reliability of power are influenced to a significant extent by 2 cycle, 16 cylinder, 8-1/2" bore X 10" stroke engines of several different models. They develop from 1610 brake horsepower at 800 rpm, burning 0.394 lbs. of #2 fuel per brake horsepower hour to 2600 brake horsepower engines at 835 rpm, burning 0.357 lbs. of #2 fuel per brake horsepower hour. Fuel consumption at full load is 91.0 gal. per hr. and 131.7 gal. per hr., and at idle 3.5 and 4.0 gal. per hr. respectively.

When in service most of their work is at full load, but there is considerable cycling between idle and full load. When not working they are generally allowed to idle, especially in cold weather in order to maintain uniform engine temperature. Except for the longer idling periods these engines appear to be worked somewhat like Ohio-Mississippi River heavy duty tow boat engines.

RELIABILITY

There are differences between railroad and marine parlance and practice which make it difficult to accurately compare engine reliability and maintenance practices. When a railroad man speaks of reliability and maintenance he is generally thinking of a three or more unit locomotive in the case of reliability and a single locomotive unit in the case of maintenance, and in both cases he is not thinking of the diesel engine alone. To him reliability and maintenance includes the engine, the controls, the electrical equipment, the air compressor, axle journal bearings, wheels, etc. His estimate of reliability is based on the number of locomotive units out of service for some kind of repair and the number of train delays occurring from any malfunction in any of the equipment. He gets alarmed if he has more than 12% of his units out of service at any time for inspection or mechanical and electrical causes.

Electrical and mechanical causes not associated with the diesel engine account for considerably more out of service time than those associated with the engine.

Locomotives must be held at light maintenance shops at specific periods for safety inspections required by law under jurisdiction of the Interstate Commerce Commission. These may account for perhaps 3% of the out of service time of the locomotive unit, leaving perhaps 9% of the down time of the unit chargeable to repairs. How much of this is chargeable to the diesel engine is questionable but I will estimate it at less than half. Ninety-six percent availability for the diesel engine may not be too far off. I do recall that on earlier freight switch and passenger units, where accurate records were kept, we obtained about 98% availability, and on the switch engine we got 98% utilization from it for several years.

A railroad man generally thinks of utilization in the terms of miles run per month. The average freight unit may run 8,000 to 10,000 miles a month, but there are units running 12,000 to 15,000 miles a month which is pretty good freight engine reliability and utilization from the railroad viewpoint where the average speed of thru freight trains ranges between 20 and 35 mph and where, to make such a schedule on runs thru flat and mountainous terrain, requires running 70 mph wherever it is permissible to do so.

MAINTENANCE

This discussion applies only to the diesel engine excluding electrical equipment, air compressors, wheels, etc. Railroad practice differs from marine practice in that there is no one assigned to the engine room while the engine is in service. When a diesel engine is offered by the maintenance shop, following a 15 day inspection, for service to the transportation department it will receive practically no attention in route for hundreds of miles and unless some malfunction occurs to interfere with its operation the only servicing it will receive will be sand, water, oil and fuel, with occasional inspection at terminals until its next scheduled inspection 15 days later. At that time engine intake air filters are changed and an engine lubricating oil sample is sent to the laboratory for analysis. Fifteen days later a 30 day inspection and maintenance will be performed including additional items such as oil filters, etc., and this will be repeated in a still more comprehensive manner at a quarterly and semi-annual inspection.

Railroad maintenance repairs are generally classified as Heavy Maintenance and Light Repairs.

Heavy Maintenance consists of two phases.

(a) *The major one is a complete disassembly* and rebuild of all major components every seven to nine years. Much material will be reconditioned and reused, but the objective is to restore the engine reliability to new condition.

(b) An upgrading rebuild is performed every 30 to 36 months. This involves reringing the engine, replacing pistons and liners as needed, new lower half main bearings and repair of leaks and other work as needed.

Light Repairs consist of

(a) Correction of conditions found at the time of scheduled 15, 30 day and quarterly and semi-annual inspections. These may involve replacement of faulty injectors, repair of leaks, adjustment of engine loading, etc.

(b) *Emergency running repairs* are those which can be performed in the running maintenance shop, such as renewal of prematurely worn piston rings, valve blows, an occasional bearing, oil cooler cleaning, or turbo changer renewal, etc.

Experience is that the lower half of the main bearings need renewal in about three years while the other bearings, including cam shaft and accessory end gear bearings can be expected to run seven to nine years.

Instead of machining crankshaft journals undersize when worn or damaged they are chrom plated back to standard size on some roads. One road having approximately 700 freight engines restores to full journal size about 43 shafts a year and scraps about 7 shafts per year as unreclaimable.

Railroads are constantly searching for means to improve maintenance and reliability of their engines. Some have or are improving their intelligence concerning the condition of the individual engines in their diesel fleets. This appears to be an essential step toward maximum reliability combined with minimum maintenance cost. To do this properly without overdoing it, or on the other hand not doing it comprehensively enough, is difficult. It appears to require very rapid flow of information concerning many things about a single engine. The difficulty is compounded by the number of engines in the fleet, which may be hundreds or thousands, by the distances which are involved between the maintenance points on a railroad and by the constant migration of the engines and lack of engineers in attendance, as in marine service. Where the number of engines runs into the thousands, telegraphic tape to card might be the only efficient way of accumulating and using such information at one central point. Such a system would keep management better appraised of the potential reliability of individual engines, the quality of workmanship at individual maintenance points and deficiencies in parts performance which affect reliability and maintenance cost. It would enable management to selectively maintain engines which need maintenance and to gain extended time between maintenance of engines which do not need it.

Most railroad locomotives are not equipped with odometers, nor their engines with hour meters or any other indication of the amount of work done. Nor are figures customarily available for the amount of fuel burned by individual engines. In the past such engines were maintained on a mileage basis where the mileage was calculated from the trips which it had made. The difficulty of keeping such records led to substitution of maintenance on a calendar basis. Neither took into consideration the amount of work done by the engine. Because of this, and because variations occur in maintenance requirements between engines of the same type in the same service, even when performing the same amount of work, it appears that uniform scheduled maintenance for all engines must lead to over-maintenance of some at high cost for reliability and under-maintenance and unreliability in others.

While there are certain things which must be inspected and maintained on a calendar basis there are many things which contribute to the life and reliability of railroad diesels, such as oil coolers, piston rings, bearings, deposits and lube oil condition which are best left alone as long as they are performing satisfactorily, but which should be fixed immediately at the first sign of malfunction.

One refinement in the prediction of parts distress is in the area of crankcase oil testing where the mass spectrograph has become a routine tool in supplementing the conventional routine crankcase oil analysis procedures. The degree of success depends upon adequate laboratory experience in interpreting the combined results of both spectrographic and conventional oil analysis. Adequate communications both ways between the laboratory and the maintainers in the field is essential. When properly organized, the life of all engine parts, which are affected by the oil, is extended by maintaining good lubricating oil in the individual engines of the fleet. It has a further advantage in that with sufficient skill a number of kinds of engine parts distress can be predicted from characteristics of the oil sample. This permits maintenance at the most economical time without adversely affecting engine reliability.

As users of diesel engines it seems that our major efforts must be directed to the efficient exchange of information both ways between the workman, the maintenance supervisor, the technical personnel and the executive level in the company. In the final analysis it appears that all the instrumentation, automation, testing, tooling, research and technology are inexorably linked with the intelligence, skill and reliability of the man who actually performs the work of maintaining the engine.

The full realization of the potential reliability and economy presently built into locomotive diesel engines appears to depend at this time principally upon our ability to maintain the right part in *exactly* the right way at the right time.

DISCUSSION

FIXMAN (Maritime): Mr. Seniff, could you give us any estimate of the man hours involved in the various kinds of overhauls you were talking about? For example a heavy maintenance job, everyday maintenance, man hours per locomotive — just a rough idea. I'm trying to relate this to marine practice.

SENIFF: I hesitate to quote figures off hand. When are you going to publish these papers Professor? (Reply — two weeks). I can put that in for your benefit if you would like to see it. I think they could. We would have to lump it for all types I think. I can say this, it has been tremendously decreased in the last 15 years. I don't try to quote figures and get a bum figure in the records. I can insert a paragraph for you on that if you like to have it.

MAC FARLANE (United Control): The idea of failure prediction by spectrographic analysis sounds very intriguing. Has this been published in the literature or is it internal in your railroads?

SENIFF: There has been a great deal published about it. You will find references to it in SAE papers. This is the first place I would look. I think you will find references there that will pretty well tell the story.

HIRSCHKOWITZ (U.S. Merchant Marine Academy): I get the impression you have a most sophisticated maintenance program. Could you possibly give us a word picture of the type of equipment and the type of staffing that your maintenance shops would require which I somehow feel contributes greatly to this reliability that we are talking about?

SENIFF: We really should start back with the 15 day inspection, a couple of men might go over a locomotive in two hours for the 15 day inspection, and 30 day might not take much longer. The quarterly will tie a locomotive up for at least 8 hours and maybe 4 men.

Actually you will find maintenance points maintaining a fleet with maybe one man per locomotive, which is spreading it fairly thin if they run into emergency work. There again I could probably cover this pretty well in the paragraph I should insert. I don't like to quote too many figures off hand.

HIRSCHKOWITZ: What type of equipment do they need to have at their disposal?

SENIFF: Do you mean in the way of tools?

HIRSCHKOWITZ: Tools and lab backup and that sort of thing.

SENIFF: Well, they have pretty good lab. backup. Any parts failure that isn't clear cut and clearly understood will be referred to the lab for analysis, the reason given if possible, and if it becomes epidemic, you had better find a reason and a cure. I would say practically all failures are analyzed as to cause and means for correction. Oil is checked locally almost trip wise for viscosity and water to determine whether there is dilution or water leaks, both of which are conducive at least to increased engine wear rates and could result in catastrophic failure if they became excessive. Sometimes you get a bad injector or broken injection line that can become excessive in one trip.

We haven't lost engines on account of fuel leaks or water leaks in years, although that used to be a prolific cause of engine failure. The oil might be checked not less often than every two weeks, more often in some cases if you had suspicious engines. You mark an engine up on the board as a suspicious engine if it has shown some sign of dilution in a previous analysis or some sign of moisture.

Maintenance forces would be alerted to correct that condition, and then you will keep that engine on alert until you are sure that it had been corrected. You save a lot of engine failure that way.

Sorry I can't give you man hours on these things, but insert it if you would like.

HIRSCHKOWITZ: The only thing I wish you would include is special tools.

SENIFF: Well, if you can envision what it would take to maintain a diesel engine, you have just about all the tools you would need in the way of guages for determining wear rates when we want to determine them. Compression instruments to determine ignition pressures, pyremeters if you need them. Frequently if we suspect something is wrong with the engine, we will put it on what they call the load box whereby you can dispose of your electrical energy and operate the engine at any load you wish to operate at, and make a quick check, standing as you would otherwise have to do running.

Outside of ordinary tools you would expect to have to maintain an engine of that size, I can't think a lot of special equipment other than that. It is pretty straight forward in that respect I would say.

HUGHES (Allis-Chalmers): Are the preventive maintenance schedules or these inspection schedules set up from experience mainly? Have you made any studies to see whether you have increased the period between inspections?

SENIFF: Inspection schedules are set up by law. We maintain at these inspection times by necessity what we see needs maintenance at those times. We take advantage of that inspection period to look over the engine and see how the rings are, the compression for instance. If it is a two cycle engine, you can look in and see the rings. If it is a four cycle, you have to take the compression, and while we have pyrometers for these guys to use to check cylinder temperatures, they go along feeling the exhaust stack. They figure that's ample, and they are pretty good at it too. They can tell when they have a defective injector by the feel of the exhaust stack or the appearance of the exhaust in many cases. If they see a locomotive coming into the house and know when they have a defective injection, it comes from experience.

So these inspection periods are set by law, and we take advantage of them as much as we can to do whatever maintenance is necessary. When we get through with one of those, we expect the critter to run until the next one without any trouble, and if it doesn't somebody has to explain for it.

HUGHES: You gave a figure of 12% as the limiting amount of down time you would allow before becoming alarmed. How much of this 12% is taken up by the maintenance intervals required by law?

SENIFF: At least 3% of that down time is actual inspection time. The rest of it will be maintenance work, about 9% will be alloted to maintenance work, and I believe I mentioned there a total of 4% actually on the diesel engine itself. The electrical equipment takes quite a lot of maintenance, comparatively speaking.

BAZOVSKY (Raytheon): Sir, you mentioned if I understood right that with the old steam engine you had an availability of 98%. Was that right?

SENIFF: The availability of the average steam engine was probably 80%.

BAZOVSKY: 80%, and do you achieve a higher reliability, a higher availability with the diesel?

SENIFF: The average, and I'm saying the best steam engine, around 80%. Average on a diesel engine should not be less than 88% or we get worried. I don't know where the railroads got that figure but that is one that is fixed in their minds, and if you get more than 12% of your fleet unavailable you're in trouble.

BAZOVSKY: This is a great improvement against the steam engine?

SENIFF: Yes, indeed.

SULLIVAN (Maritime): Could you go into just a little more detail on the programming of maintenance information that you said some of the companies have begun recently as to exactly how they have gone about getting this information.

SENIFF: Let me just give an illustration and this won't be a whole lot of detail. It may serve to illustrate what they have in mind. The problem is to try to stay in the middle of the road with this thing and not to collect too much information on one hand and not enough on the other. You can drown yourself in information if you get too much of it even though you automate it all.

Let's take a simpler example. If we know how much trouble this engine has been in with water leaks, fuel oil delution and some idea how long she has suffered, we have some idea how our lubricating oil is running, it is a pretty good indication of the health of the engine. So you should feed that data into some system that would retrieve it quickly for you. Here is an engine that has been in good health as far as its blood stream is concerned, so its wear should be lower than an engine that has had a number of instances of trouble.

You need to know how many power assemblies have been pulled out of 16, and for what reason. So you would have a date on power assembly pull and a reason why it was pulled. We used to write it down this way: B for broken rings, W for worn rings, and so on and so forth until we covered all the principal causes of the power assembly renewal. Going through the rest of the component parts of the engine, the bearings and so forth, you know the dates on those and here is an engine that has a nice clean history behind her, and she'll probably run a lot longer than one that has a dirty history behind her, a history of water leaks, dilution which you know has scuffed up things in the engine, increased wear rates, or an engine that is predominently running a lubricating oil dirtier than another engine, you know that engine is the one to watch, it will probably be the first one in your shop.

We used this during World War II to great advantage with a fleet of some 20 or 30 freight engines and 6 passenger locomotives. It was a life and death struggle to keep those darn things in service. Where bearings in those days were supposed to do just fine, the connecting rod bearings for instance, if you got a couple hundred thousand miles out of them.

We had some emergency war-time bearings that they thought a hundred thousand would be pretty good, and a lot of railroads lost about a hundred thousand. We watched this fleet of engines very closely and watched all these details and maintained by those details and didn't allow any engine to go out that had a trace of dilution or water in the crank case, because we knew the bearings were extremely sensitive to the change in viscosity or in water, even the best of bearings up to that time had a life expectancy of 200,000 miles.

We ran some of those locomotives right straight through the war and accumulated almost a million miles, nine hundred and some odd thousand miles. On some of those units just by watching these things and predicting and maintaining by them. Whereas 200,000 would have been considered pretty good in normal times.

Now we lost some of the bearings. We had to renew some of them earlier, but I'm speaking of the better engines, and the whole thing is just in the difference in the nature of the critters. Here was a fleet of engines all delivered about the same time, the same kind of service, the same manufacturer, the same class, all shuttling in the same type of service, the variation in life of the bearings for instance of those engines ranged from 200,000 miles to nearly a million miles.

We predicted when they needed attention by oil analysis, keeping the oil good, and when we saw something happen that was a demerit against that engine. If she got a shot of fuel oil dilution or water dilution, we knew we had done some damage. So somewhere along here we started watching these engines that had been bad performers and managed to carry some through to phenomenal mileage. We've never done that well since, because we've never again followed any of them that closely. But it could be done again if you could get the flow of information in such a form that you could retrieve it in an intelligent manner, and that is what some of us are trying to do.

It will involve more than just the things I've mentioned, all the things that might affect life expectancy of an engine, but I know it does work, because it has worked on a smaller scale. It is just a matter of blowing it up bigger scale and making it work on a fleet of a thousand or two thousand engines, and I hope that is what you fellows are talking about and give us a way of doing it to help us in our approach to this thing.

COYLE (ONR): You mentioned that you idle your engines and don't secure them to keep your temperature pretty well equalized. I was wondering if you ran into excessive carbon build up around your rings due to all this idling?

SENIFF: Yes, it is a bad thing. We don't like to do it, but it is still the best way we find to keep the critters warm and ready to go in the winter time. In talking not long ago with a representative of one of the Canadian railroads on the pros and cons of keeping engines warm in the winter time, he felt after trying many different approaches that it still was the best way to do it. However, it is the toughest thing there is on an engine. A diesel engine doesn't like that nearly as well as it likes to get out and work hard and clean itself up.

KAUFMAN (Naval Boiler & Turbine Lab): When you spoke of bearing corrosion some years ago, what was the cause of the bearing corrosion?

SENIFF: There were several causes. I had better start at the most recent ones and work backwards. One was the loss of lead out of the copper-lead matrix of copper-lead bearings. That is where the spectrograph really shines. We find that we couldn't possibly have any bearing corrosion of a copper-lead bearing without lead showing up just like that in the lubricating oil spectrographic analysis. You couldn't miss bearing corrosion with a spectrograph. It was a perfect approach and everybody thought that the spectrograph was wonderful in those days when we licked the problem and then we wondered what we were going to do with the spectrograph.

It licked the copper-lead bearing corrosion problem by telling us when we had it, when to look for it, and it didn't take too long after we started using that approach to really solve the problem.

Now going further back we had other causes of bearing troubles, and one of the most bitter ones I can think of was misuse of fuel oil additives. This is going so far back, it won't make any difference. This was about 1938. We were inveigled in trying a special fuel oil additive and the number of the lube oil went like that, and the bearings came out at about the same rate. Since then I've always been very cagy like the burned child about using anything when you don't know what it is before you put it in an engine. It was a bitter lesson. We had several bearing corrosions for instance caused simply by oil acidity or some particular acid in the oil. We've also lost bearings simply because of the failure of the oil to lubricate a given type of bearing. So we were very cagy about the kind of oil we put in our railroad diesels. Now I admire the military because they say they don't consider diesel engine a real good engine unless it will use most any kind of lubricating oil, and I kind of back them up on that. I think it is real nice if an engine can be made rugged enough that it is not so choicey on the kind of lubricating oil it would like to have.

THE RELIABILITY OF MODERN AIRCRAFT GAS TURBINES FOR MARINE PROPULSION

DR. J. J. MC MULLEN

A. INTRODUCTION

Although reliability is always listed as a requirement for marine propulsion systems, reliability as a specific measure of operational ability has not been rigorously, scientifically or thoroughly treated in the marine field. Rather, we are inclined to use qualitative and general measures of power plant reliability such as maintenance cost, size of crew required, and time lost in port or at sea due to failure of components. These statistics, roughly related to reliability, are quite common within the marine field because of the ever present need for economic studies. True power plant reliability evaluation has been hampered by the difficulty in obtaining feed-back reports from ships on component outages unless a serious casualty occurs. There has been no continual requirement by manufacturers for feed-back information and, further, the process of providing complete records of information on breakdowns is an added burden which would add to the cost of ship operation. As a result, a complete reliability analysis has seldom, if ever, been concluded for the propulsion system of a merchant ship.

This has not been so in the case of aircraft operation because casualties in connection with aircraft engines make headlines and are a direct matter of life and death. Furthermore, the manufacturers have been very demanding of servicing personnel to provide feed-back reports in order to accumulate data on their machines so as to guard against serious casualties, to increase time between overhauls and to continually improve ratings and specific fuel consumption.

Treating reliability as a ship problem, it is necessary to draw a boundary line around everything connected with the propulsion system. This would include not only the reliability of the main shafting components and main drive, but also the supporting auxiliaries. In addition, however, this boundary line should encompass the fuel system and fuel quality. Figure 1 outlines this boundary line around a very simplified representation of a power plant.

Reliability of the over-all machinery system is of course capable of being easily treated mathematically, as will be discussed in detail later, subject to the accuracy of input data by component. For example, reliability of series components is merely the product of the individual reliabilities of each component in the series system. If each individual component in a series system consisting of five units had a reliability factor of .995, the resultant reliability of the chain of five in-line components is 0.975 and for ten in-line components the over-all reliability is 0.951. This highlights the importance of reducing the number of machinery components.

Obviously, you can increase over-all reliability of a plant by the use of a stand-by unit in parallel with the running unit but the increase is not nearly as great as could be gained by eliminating the need for the unit altogether. In short, the most reliable plant is the one with the minimum of complexity and the fewest number of components.

Before proceeding to a detailed analysis of the reliability of a marine plant utilizing an aircraft gas turbine, attention should be directed to the important general aspects and conclusions on this subject. First, the Aircraft Gas Turbine has a very carefully accumulated history of trouble reporting and therefore its reliability can be precisely stated. The same is true for large electric generators and electric motors because the electrical manufacturers and power companies likewise have kept reliability and outage records of large units. Difficulty of data accumulation develops with the use of small units where there has been a lack of information on troubles with these units. This is not necessarily the fault of the manufacturers but more a lack of communication in reporting back out-of-service problems.

Secondly, it is obvious that power plant reliability, assuming good quality components throughout the plant is exponentially increased as the number of plant components is reduced.

Confidence in the accuracy of reliability of a component is a function of the number of operating hours used on the basis of data. For example, if a family of units has a total history of operation of a million or more hours with a good feedback of trouble experience, the confidence in the resultant reliability factor is high. However, in a new component particularly a custom made component or a developmental component the confidence in the reliability factor must be low until extensive operating history has been obtained. It is for this reason that in the new field of nuclear power, plant components are kept on test continuously so as to keep ahead of the total number of hours of operation actually on installed plants. It is for this reason that the confidence in the reliability of the gas turbine of the aircraft type is high because of the extremely large number of operating hours behind the data.

The degree of reliability is also dependent upon the consistency of fuel supply to the propulsion system because the gas turbine plant engine (and also the steam boiler and the diesel engine) is sensitive to contaminants such as vanadium, sulphur, and sodium as well as total ash content. Therefore, a higher degree of reliability must be assigned to a fuel which meets a precise gas turbine fuel specification than to an unspecified fuel even if treated.

The next point has to do with the reversing methods. There are many choices of reversing that can be considered such as a controllable pitch propeller, electric drive and various mechanical and hydraulic reversing mechanisms. However, only two of the reversing systems have had sufficient experience at high power ranges so that a high degree of confidence can be put in the reliability of the system selected. Hence, in our work with gas turbines we have limited the choice of the electric drive and to the controllable pitch propeller. There should be no question as to the high degree of life expectancy and over-all good operation that electric drive provides because of the T-2 tanker and other marine experience which is widely recognized. As to controllable pitch propeller, there are few in service in this country except in very small horsepowers; however, a large number, over 2000, are in service in foreign flag ships. In fact in the total tonnage afloat a significant percentage of plants do have controllable pitch propellers and an impressive reliability record is accumulating.

On the gas turbine itself, the reliability is also a function of the amount of development, testing and improvement that has gone into the evolution of this component. With the extensive facilities that the aircraft gas turbine manufacturers have maintained for test purposes, there can be little question that no marine component has been so searchingly examined and perfected as has been the gas turbine for aircraft use.

It should be emphasized that the over-all plant can be considered to be more reliable if absolutely no unproven component is used. In the plant described in this presentation, such is the case; all components employed have extensive background of service behind them.

It is important to note that in steam and diesel marine plants the improvements over the years have increased efficiency, decreased cost, decreased weight, and made other improvements but generally with an increase in complexity because such improvements and experience with casualties have introduced safety devices and added components so that plants of today are more complicated. In addition, with the emphasis presently being placed on automation, it is obvious that steam, diesel, and industrial type gas turbines will become even more complex. It is apparent that the aircraft type gas turbine has an outstanding advantage in this regard because, by concept, it is already simple and automated.

The following sections will treat in more detail three pertinent areas contributing to the over-all reliability of the marine propulsion plant. These sections review the effect of fuels on reliability, casualty analysis and reliability of the gas turbine, and the reliability of the over-all power plant.

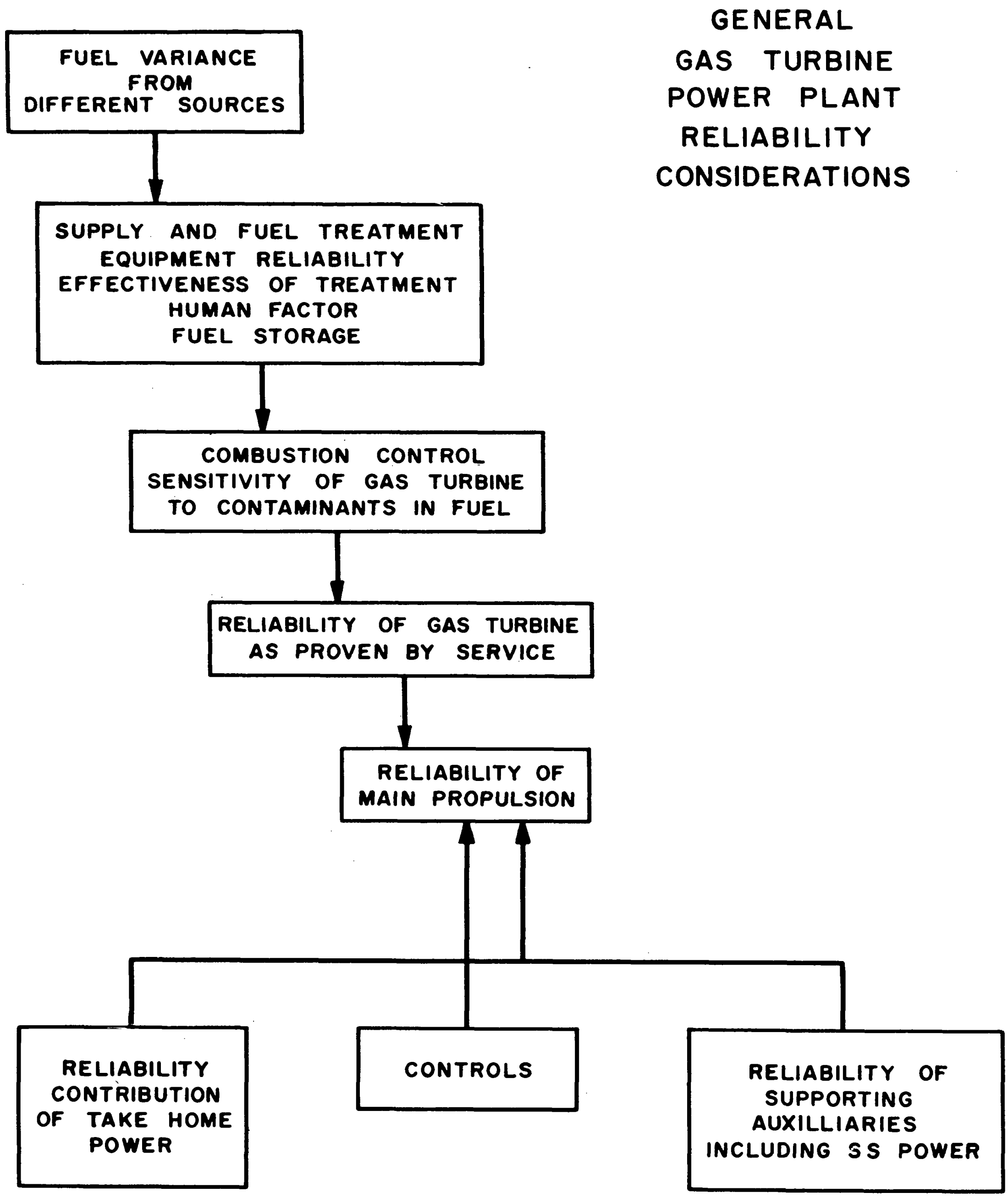


Figure 1.

B. FUELS

Although fuels cannot be considered a component of the power plant to which a reliability factor can be specifically assigned, nevertheless the degree of consistency of fuels and of fuel treatment effectiveness and methods to produce a fuel of certain uniform characteristics has a bearing on a reliability of all types of propulsion. It is true that the higher temperatures involved in a gas turbine make it more sensitive and the effects of low grade fuels are more quickly noticeable.

Sodium, vanadium, sulphur and ash content are important variables affecting the engine performance. These variables determine the amount of corrosion, erosion, and build up of deposits which result in changes to the efficiency of gas turbine engines, the time between overhauls, and the ultimate life of the engine. Unfortunately, the usual residual fuel ash components result in compounds whose melting or softening temperatures are below the turbine operating temperature. These compounds tend to flux out and deposit on the hot turbine metal surfaces with resultant lower efficiencies and excessive metal temperatures.

The Pratt and Whitney Aircraft specification for fuel for aircraft gas turbine for marine use limits sulphur to 1.0% and total ash to 0.005%. While limits on sodium and vanadium are not specified, their control is by the total ash specified and certain other characteristics limited in the specification and by burning tests. If specified they would be held to a low value of ppm.

The problem with fuels begins with the source of materials. If a fuel is purchased to a manufacturer's specification so that it can be burned directly in the engine without treatment the effective reliability of the fuel can be considered to be 1.0. However, if a fuel is bunkered which must be treated in order to satisfy the reliability requirements and the life of the engine, the precise reliability effect of the fuel is difficult to ascertain for the following reasons:

1. These fuels are generally residual fuels. With the development of improved refining processes, larger proportions of the charged crude have been removed as distillate and motor fuel stock, and the trend has been to produce less residual oil but with higher concentrations of contaminants. Instead of a 40% residual oil yield that was common in years past, it is not uncommon for the amount of residual oil obtained to be now as low as 5% on crude oil. The consistency of No. 6 residual fuel varies widely from port to port and often from wells in the same oil field and may have contaminants ranging as follows:

Sulphur	0.6% to nearly 4%
Vanadium	0.0 ppm to 400 ppm
Sodium	1.6 ppm to 290 ppm
Total ash	0.1% to 2%

Actually there is no typical No. 6 residual fuel because it is an unspecified fuel and the above was selected at random from a limited survey report in order to highlight the discrepancy between contaminants and the typical limits of contaminants in fuel to gas turbines necessary for long life and high reliability.

As a matter of fact, most oil companies are reluctant to state the analysis of Bunker "C" fuel oil and since the oil is generally purchased under annual contracts, the steamship operator must be prepared to accept bunkers all over the world with widely varying properties.

2. Heavy fuels like residuals which are bunkered may stratify in the fuel tanks and, therefore, the consistency of the bunkered load varies as it is used.

3. The bunkered fuels may pick up salt water in the fuel oil tanks and scale from pipes, tanks and treatment equipment.

After several years of development toward burning of residual fuels, methods have been developed for treatment of fuel by filtering, washing, emulsifying, mixing, centrifuging and inhibiting through the use of additives. This adds equipment which must be included in the power plant diagram in calculating over-all reliability of the plant.

Therefore, the fuel treatment apparatus must be able to reliably and automatically cope with variations in fuel which depend not only upon the variations when bunkered but also the variations that can occur on board the ship.

Obviously then, the reliability of the gas turbine depends upon the consistency of the oil which is fed into the burners and it follows that a fuel which is purchased to specification has a higher factor of reliability than that which is purchased without specifications to limit the most harmful contaminants, and which must be treated continuously to produce an acceptable fuel.

One point is obvious; the use of higher grade fuels will certainly increase reliability and at the same time, reduce maintenance and repair costs.

C. THE PRATT & WHITNEY AIRCRAFT TYPE FT4 MARINE GAS TURBINE

In most marine applications the main engine is custom built to the ship or is the adaptation of a product line where a dozen or so identical units have been produced. However, in aircraft gas turbines there are thousands of units that have been produced which can be adapted with minor change to marine propulsion. This, like the total number of hours of service, lends confidence in the ability of this plant to continue running without casualty. Specifically in regard to the FT4 main propulsion proposed by the Pratt & Whitney Aircraft study for the Maritime Administration:

1. The design which was used for the basis of the FT4 has over 4,000,000 hours of operating experience in aircraft service. Its long term development and tests are completed. Also it is a simple engine with few moving parts, small and light and requires a very simple supporting auxiliary plant.
2. Being developed for use in an aircraft, it is already designed for remote control and, therefore, requires no development or use of additional controls for application to a ship.
3. The Pratt & Whitney Gas Turbine Power Plant is a plant that operates with no high pressure systems associated with it. It eliminates all of the components and systems associated with the usual steam plant. For example, there are no boilers having a large number of tubes which statistically may result in early failure; there is no high pressure steam system with all its associated valves; no auxiliary steam, reducing valves, pressure regulators, boiler level control, drains, relief valves, no feed and condensate system, no gland seal, no condenser, no large seachests and main circulating pumps, and the lube oil, fuel oil and salt water systems which are required are greatly simplified over that of the standard steam plant. Even compared with the diesel plant the gas turbine is simple because the internal moving parts are far fewer than that of the diesel engine.

   Furthermore, there is inherent reliability in reduced size and complexity of the gas turbine. The gas turbine is analogous to components developed in the electronic field. The use of transistors and other solid state devices, printed circuits and the like have produced much more reliable equipment along with reduction in space occupied and complexity. Similarly, a 15,000 pound gas turbine/free turbine unit replaces some 400 to 500 tons of steam, water and other machinery in getting power to the output flange.

D. BACKGROUND

Although the purpose of this paper is confined to estimating the over-all reliability of a modern aircraft-type gas turbine in a marine propulsion system, it may assist the reader to include a brief description of the power plant.

The basis for the work presented in this paper was a research and development contract granted by the Maritime Administration to Pratt & Whitney Aircraft, Division of United Aircraft Corporation, East Hartford, Connecticut, with Isbrandtsen Company, Inc. and John J. McMullen Associates, Inc. as participants. The study covered integrated marine gas turbine power plants and was confined to preliminary design and investigation of the subject.

In order to maintain a basis of evaluation, Maritime fixed certain of the parameters, one of which included the basic design of the ship. This reference design, known as the ship hull PD108, was an advanced concept of ship design for high speed cargo handling. The preferred machinery location was forward and this was the location which was used by the Pratt & Whitney team in developing all of the facts and information.

Basically, the conclusions of our detailed study was that a single Pratt & Whitney aircraft FT4 gas turbine engine driving a 3600 RPM electric generator directly from the free power turbine output shaft resulted in the most feasible prime mover for the PD108 design.

At the outset of the program, the FT4 had capabilities of delivering 20,000 SHP with a time between overhaul of 2,000 hours. The engine performance was based on use of a distillate fuel such as Navy Diesel or commercial equivalent such as Diesel oil or fuel oil #2. However, no technical reason was or is foreseen to prevent the attainment of at least 4,000 hours between overhaul, higher horsepower and a significant reduction in fuel costs with the use of lower grade, less expensive distillate fuels after sufficient experience has been gained either on the test stand or through operation in a merchant ship. Steps are now being taken to intensify the fuels program.

The selection of electric drive propulsion for the PD108 was consistent with the ship's lines and the objectives of reliability, simplicity of operation and maintenance and maximum cargo cubic capabilities. The low installed weight and compact size of the FT4 propulsion system make possible complete gas turbine engine removal and replacement with a factory serviced unit well within a 24-hour period. The engine is removed by way of a patch over the inlet air passage and no hull cutting is required.

This thoroughly integrated power plant represents a low installed capital cost and over-all operating economy, including fuel, even with the increased fuel rate envisioned by the slightly lower efficiency of the electric drive. The total simplicity of the conceived propulsion system and its attending auxiliaries permits complete operation of all machinery functions from a central control console on the bridge. As shown in this paper, the reliability together with attendant simplicity of the gas turbine exceeds that of any other type of marine propulsion systems.

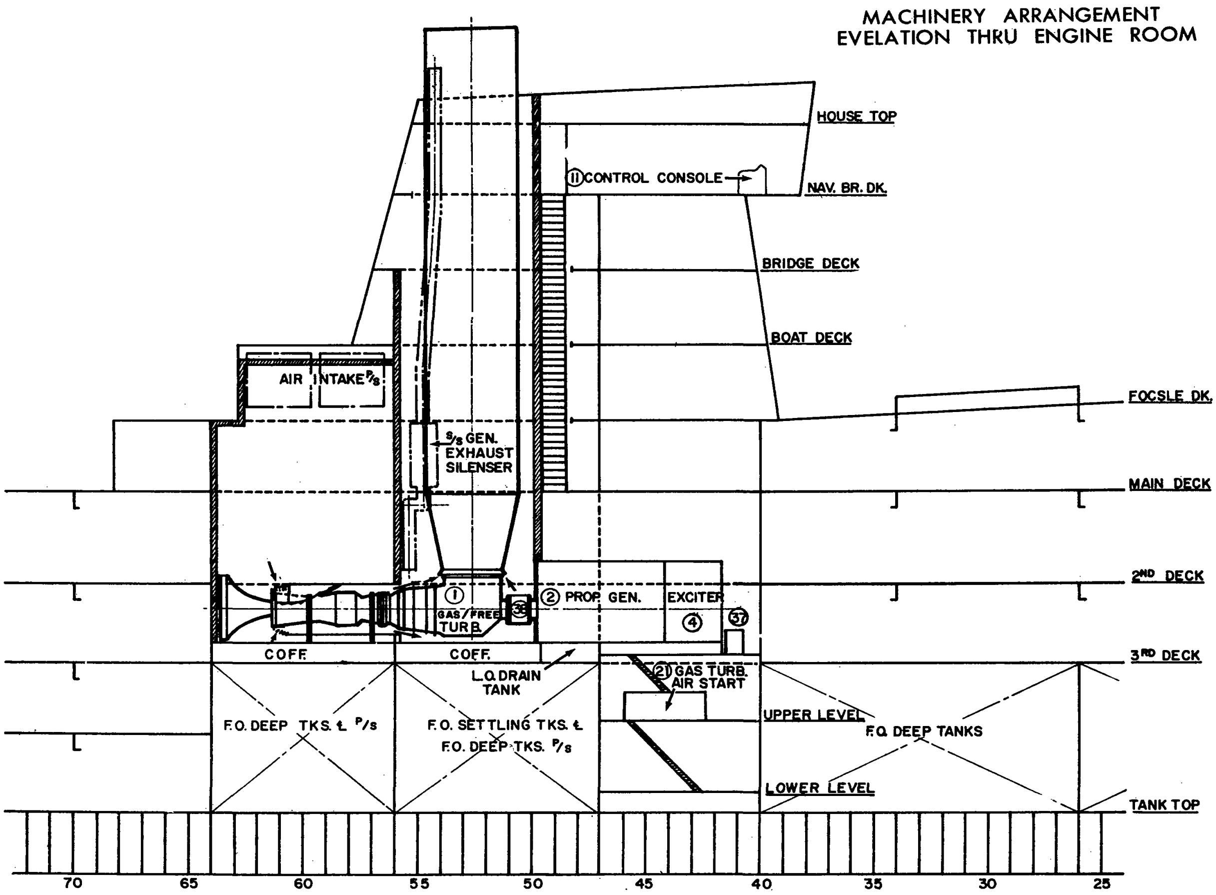
The four layouts which follow show plan and elevation arrangements of the machinery spaces and it is hoped that with this information the reader will be able to understand the basis of the propulsion system which was used to present the following reliability study.

It should be noted that the above has in no way attempted to present all of the technical and economic reasons for the selection of various components, but that this information is available and is presented in "Phase I Study — Integrated Marine Gas Turbine Power Plants" which was presented to the Maritime Administration.

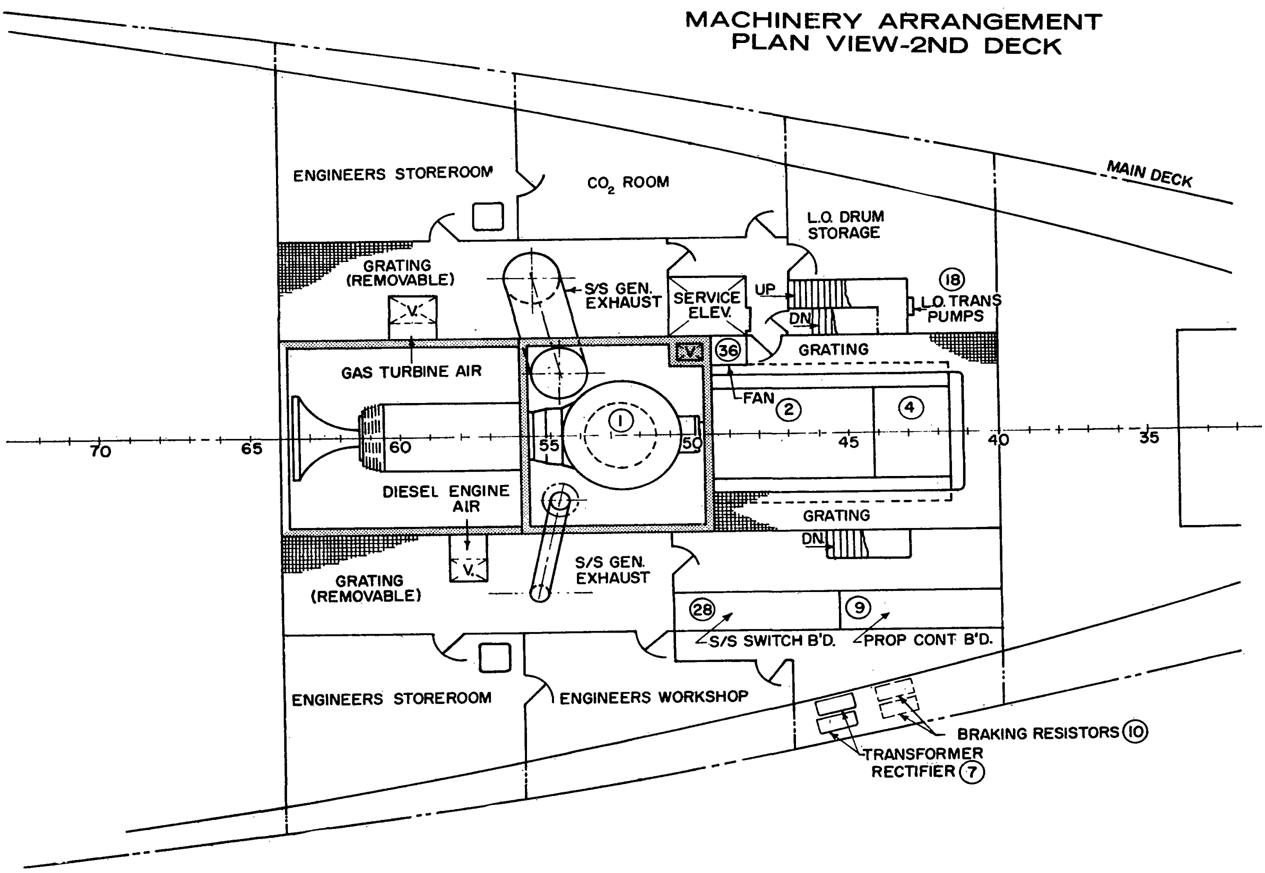
E. RELIABILITY AND CASUALTY ANALYSIS OF GAS TURBINE

In order to provide a clearer understanding of our approach, the following discussion of casualty analysis and reliability of the power plant is presented.

Reliability is the probability that a device will perform at a specified level, in a specified environment for a specified period of time. The true reliability of a power system is never actually known until many of the devices have been operating in the field for a long period of time. For this reason, it has been the practice of Pratt & Whitney Aircraft to catalog failures against all of its engines in field

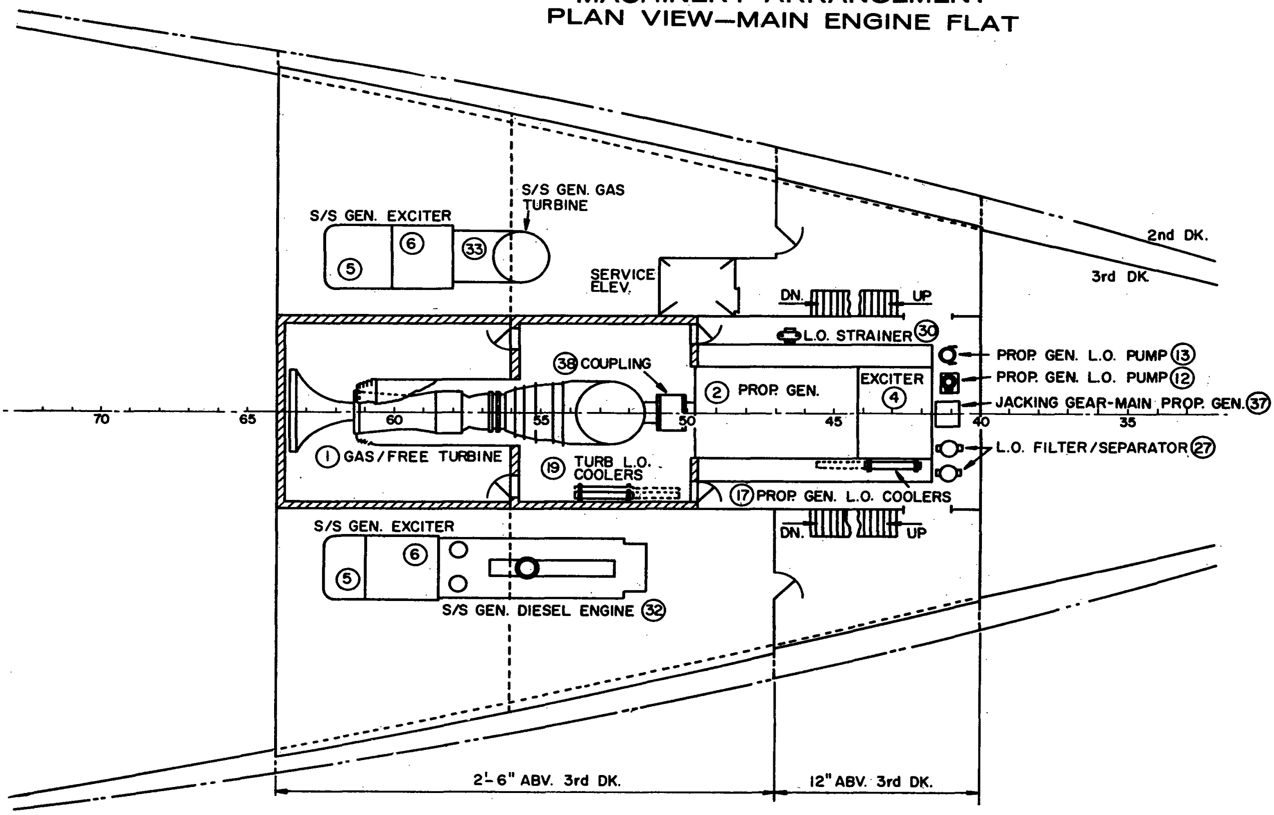


MACHINERY ARRANGEMENT
EVELATION THRU ENGINE ROOM
HOUSE TOP
11 CONTROL CONSOLE
NAV. BR. DK.
BRIDGE DECK
BOAT DECK
FOCSLE DK.
MAIN DECK
2ND DECK
3RD DECK
TANK TOP
AIR INTAKE P/S
S/S GEN. EXHAUST SILENSER
1 GAS/FREE TURB.
2 PROP. GEN.
EXCITER
4
37
COFF.
COFF.
L.O. DRAIN TANK
21 GAS TURB. AIR START
UPPER LEVEL
LOWER LEVEL
F.O. DEEP TKS. ℄ P/s
F.O. SETTLING TKS. ℄
F.O. DEEP TKS. P/s
F.O. DEEP TANKS
70
65
60
55
50
45
40
35
30
25

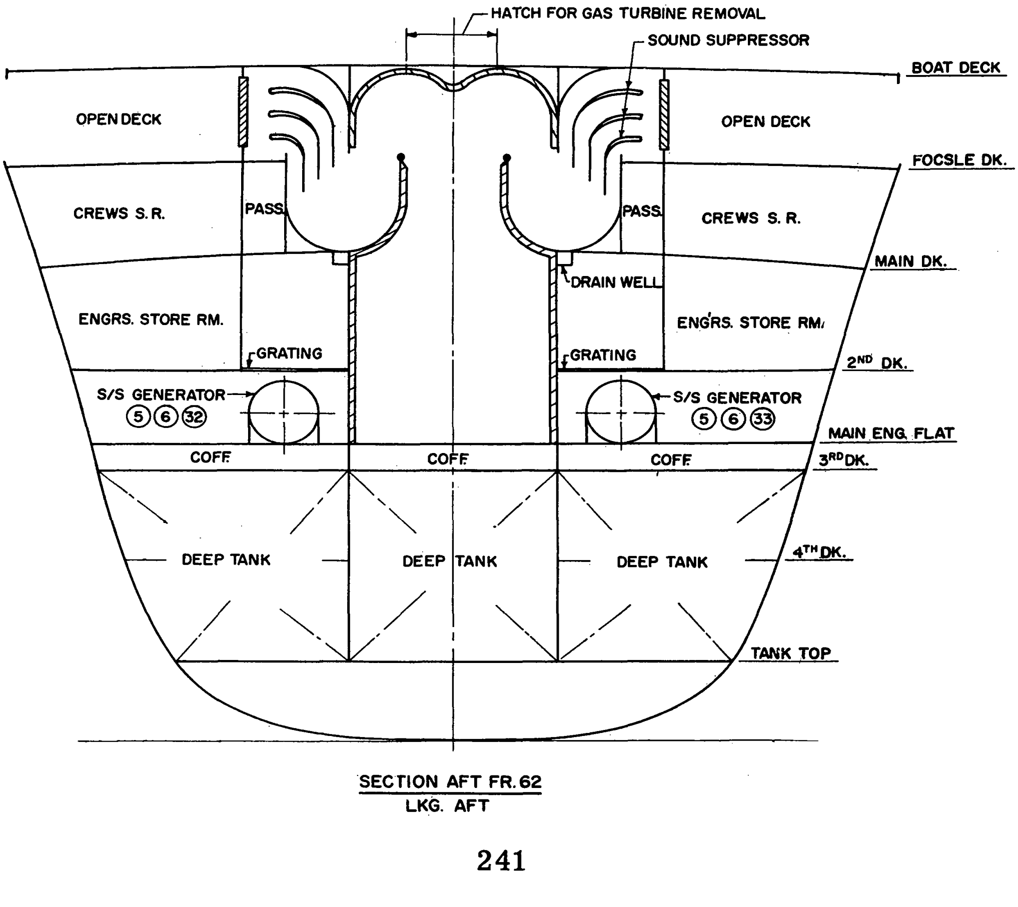


MACHINERY ARRANGEMENT
PLAN VIEW-2ND DECK
MAIN DECK
ENGINEERS STOREROOM
CO2 ROOM
L.O. DRUM STORAGE
GRATING (REMOVABLE)
S/S GEN. EXHAUST
SERVICE ELEV.
UP
DN
18
L.O. TRANS PUMPS
V.
GAS TURBINE AIR
36
GRATING
FAN
2
4
1
70
65
60
55
50
45
40
35
DIESEL ENGINE AIR
GRATING
DN
V.
S/S GEN. EXHAUST
GRATING (REMOVABLE)
28
9
S/S SWITCH B'D.
PROP CONT B'D.
ENGINEERS STOREROOM
ENGINEERS WORKSHOP
BRAKING RESISTORS 10
TRANSFORMER RECTIFIER 7

MACHINERY ARRANGEMENT
PLAN VIEW—MAIN ENGINE FLAT



MACHINERY ARRANGEMENT
SECTION THRU ENGINE ROOM



SECTION AFT FR. 62
LKG. AFT

service. The failure data is statistically analyzed against operating time. This means a reliability audit is maintained against each engine. This information is in the form of:

1. In flight shutdowns.
2. Premature engine removals.
3. Flyovers (component replacement without overhaul).
4. Time between overhaul (T.B.O.)
5. Component failures.
6. Part failures.

The conclusions drawn in this reliability audit and applied to marine and industrial engines are considered conservative since the aircraft service generally imposes more cyclic stresses on the components.

On the basis of this background, a reliability audit was made of inflight engine shutdowns from the start of JT4 commercial jet operation in 1959 through April 30, 1962. During this period, the accumulated operating time was 3,569,478 hours. Approximately one unscheduled shutdown occurred per 25,100 hours of engine operation. A review of these shutdowns, listed on page 251, indicates that over 40% were of a type which could have been repaired aboard a ship. On this basis, approximately 60% of the failures, or one in 42,500 hours of operation, could not be repaired aboard a ship. Should an engine require overhaul, it can be removed and replaced with a factory overhauled unit within 24 hours while the ship is in port.

Reliability is calculated using the following equation:

$$R = e^{\lambda t}$$

Where:
R = reliability
e = 2.73
λ = random probability (failure/hr.)
t = mission time (hr.)

A curve of time between overhaul vs calendar time for the JT4 engine is shown in Figure No. 2.

This reliability audit includes only the gas turbine engine. The design of the free power turbine is both mechanically and aerodynamically conventional. Furthermore, the maximum gas inlet temperature to the power turbine will be approximately 60% of the gas generator turbine temperatures. Because of these facts, Pratt & Whitney Aircraft believes that the reliability of the free power turbine will be better than the gas generator.

Failure Analysis

A failure analysis for each major component of the gas turbine engine is shown on pages 243 through 249. Included in these analyses is a listing of the various components, possible malfunctions, the probability of the malfunction occurring, the effect of such a malfunction or failure, and the proper corrective action.

All of the various headings are self-explanatory with the possible exception of probability. It is felt that a "feel" for the meaning of failure probability can be gained by drawing an analogy with a D.C. motor. If failure possibilities in a D.C. motor were rated on the same basis as the gas turbine engine study, they would be rated as follows:

Infrequent - Failures would be represented by commutation problems (brushes)
Seldom - Failures would be represented by bearing failures.
Rare - Failures would be represented by field coil failure.

GAS TURBINE ENGINE

Failure Analysis

Component	Possible Malfunction and Effect	Probability	Corrective Action
Gas Generators			
Front Compressor			
1. Unbalance	Rough operation - In most cases automatic shutdown.	Rare	Overhaul gas generator to repair and balance front compressor.
2. Blade Failure	Vibration with resulting shut-down.	Rare	Overhaul gas generator.
3. Tachometer drive failure	Loss of speed indication	Rare	Repair at next shutdown.
4. Bearing Failures	Rough operation - Resultant vibration with automatic shut-down, dependent on degree of failure.	Rare	Overhaul gas generator.
5. Seal Failures	Loss of oil, drop in oil pressure, which could result in automatic shutdown dependent on degree of failure.	Rare	Overhaul gas generator.
Rear Compressor			
1. Unbalance	Rough operation - In most cases automatic shutdown.	Rare	Overhaul gas generator.

Component	Possible Malfunction and Effect	Probability	Corrective Action
Rear Compressor			
2. Blade Failure	Vibration with resulting shutdown.	Rare	Overhaul gas generator
3. Tachometer drive failure	Loss of speed indication	Rare	Repair at next shutdown.
4. Bearing failures	Rough operation - Resultant vibration with automatic shutdown dependent on degree of failure.	Rare	Overhaul gas generator.
5. Seal Failures	Loss of oil, drop in oil pressure which could result in automatic shutdown dependent on degree of failure.	Rare	Overhaul gas generator.
6. Magnetic Pickup failure	Inability to start.	Rare	Replace pickup.
Burner Section			
1. Burner can faulure	Poor turbine discharge temperature distribution. Automatic shutdown dependent on degree of failure.	Seldom	Repair or replace burner can.
2. Manifold failure	Poor turbine discharge temperature distribution. Automatic shutdown dependent on degree of failure.	Rare	Repair by replacing manifold.

Component	Possibile Malfunction and Effect	Probability	Corrective Action
3. Fuel Nozzle plugging	Poor turbine discharge temperature distribution. Automatic shutdown dependent on degree of plugging.	Seldom	Repair by removing nozzle-cleaning and replacing.
Turbine Section			
1. Nozzle guide vane failure	Poor temperature distribution. Vibration with automatic shutdown dependent on degree of failure.	Rare	Repair by replacing nozzle guide vanes.
2. Bearing Failures	Rough operation. Resultant vibration with automatic shutdown dependent on degree of failure.	Rare	Overhaul gas generator.
3. Seal Failures	Loss of oil, drop in oil pressure which could result in automatic shutdown dependent on degree of failure.	Rare	Overhaul gas generator.
Anti-icing shut off Valve Failure	Inability to turn on could result in undesirable icing. Inability to turn off could result in slightly lowering of maximum power available due to inlet heating	Rare	Replace or repair.
Anti-icing valve regulator failure	Could result in undesirable icing or excessive power loss due to excessive anti-icing air flow.	Rare	Replace or repair.

Component	Possible Malfunction and Effect	Probability	Corrective Action
Ignition Exciter	Inability to start if both exciters have failed.	Seldom	Replace ignition exciter.
Ignition plug failure	Inability to start if both igniter plugs have failed	Seldom	Replace ignition plug.
Accessory drive bearing failures	Roughness, vibration, instability or possible automatic shutdown dependent on degree of failure.	Rare	Repair
Accessory drive shafting failures	Automatic shutdown.	Rare	Repair
Accessory drive seal failures	Loss of oil, contamination of oil with fuel, loss of oil pressure with resultant automatic shutdown.	Seldom	Repair - If fuel to oil seal leak - change oil.
Accessory drive spline failures	Loss of component drive with automatic shutdown.	Seldom	Repair
Fuel Pump Failure	Automatic shutdown	Seldom	Replace pump - check fuel system downstream for evidence of contamination. If contaminated, replace components as necessary.
Fuel deicer failure	Loss of fuel deicing protection.	Rare	Repair or replace.

Component	Possible Malfunction and Effect	Probability	Corrective Action
Fuel control failure	Shutdown, uncontrolled operation.	Rare	Replace fuel control.
Fuel control shutoff valve actuator failure	Inability to start or shut-down normally. Can shutdown with emergency stop.	Rare	Replace actuator.
Fuel oil cooler failure	Oil leak. Oil contaminated with fuel.	Rare	Replace and change oil.
Emergency shutdown valve failure	Emergency shutdown protection lost providing both emergency shutdown valves failed simul-taneously.	Rare	Repair or replace valve
Pressurizing and dump valve failure	Engine operation limited due to pump relief operation, fuel leakage.	Rare	Replace pressure and dump valve.
Fuel pump filter pressure differential switch failure.	Loss of filter plugging. Engine power output limited	Rare	Replace switch.
Oil tank failure	Oil leakage.	Rare	Replace oil tank.
Oil pump failures	Loss of oil pressure. Bearing contamination.	Rare	Replace and check for contamination.
Oil scavange pump	Oil overheating. Oil contamination.	Rare	Replace and check for contamination.

Component	Possible Malfunction and Effect	Probability	Corrective Action
Control linkage from free turbine to gas generator failure.	Uncontrolled operation.	Rare	Repair
Engine Discharge temperature thermocouple failure.	Loss of temperature indication.	Seldom	Replace
Free turbine to gas generator bellows connection failure.	Loss of power. Exhaust gas discharge into engine compartment.	Rare	Repair
Free turbine nozzle guide vane failure.	Roughness. Vibration with resultant automatic shutdown.	Rare	Overhaul free turbine.
Free turbine rotor blade failure.	Vibration with automatic shutdown.	Rare	Overhaul free turbine
Free turbine accessory drive bearing failure.	Roughness, vibration, instability dependent on degree of failure.	Rare	Repair.
Free turbine accessory drive gearing failure.	Loss of free turbine control automatic shutdown.	Rare	Repair.
Free turbine accessory drive seal failures.	Oil leakage - Low oil pressure with resultant automatic shu down.	Rare	Repair.

Component	Possible Malfunction and Effect	Probability	Corrective Action
Free turbine governor failure.	Loss of free turbine control-automatic or emergency shutdown.	Seldom	Replace governor.
Free turbine magnetic pickup failure.	Automatic shutdown.	Rare	Replace pickup.
Free turbine shaft failure.	Automatic shutdown.	Rare	Overhaul free turbine.
Free turbine coupling failure.	Automatic shutdown	Rare	Replace coupling.
Free turbine exhaust duct failure.	Exhaust gas discharge into engine compartment.	Rare	Repair.
Free turbine mount failure.	Vibration with automatic shutdown.	Rare	Repair and realign free turbine with generator
Gas generator mount failure	Vibration with automatic shutdown.	Rare	Repair and realign gas generator with free turbine.
Engine sequencing system failures	Uncontrolled automatic operation. Manual control	Infrequent	Repair
Air starter failure.	Inability to Start	Rare	Replace the starter

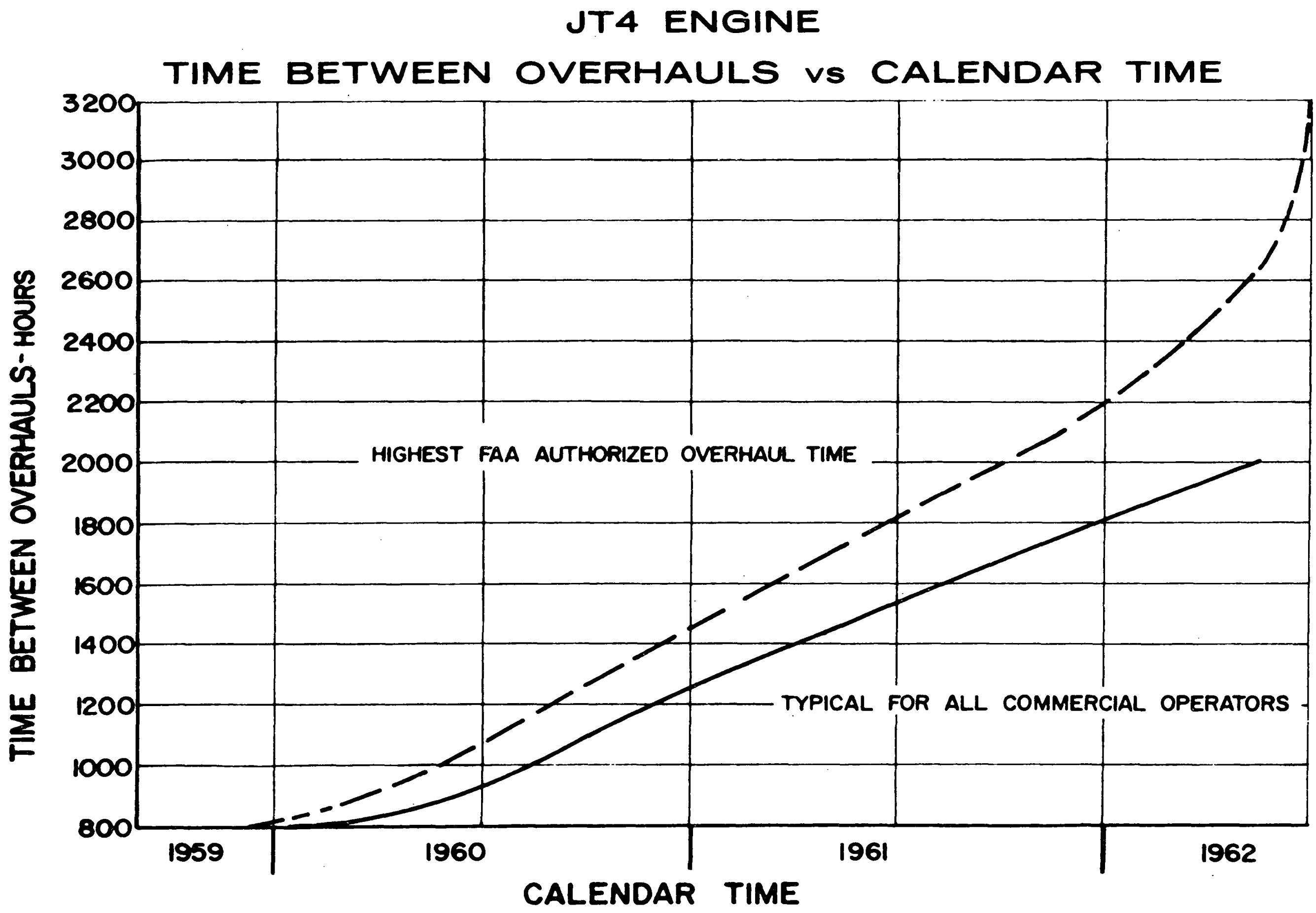


Figure 2.

F. IN-SERVICE PROBABILITY OF ELECTRIC PROPULSION SYSTEM

In-service probability of any component of the propulsion system may be mathematically defined as follows:

$$P = \text{in-service probability} = \frac{\text{hours in operation}}{\text{hours in operation + hours forced out of service}}$$

$$Q = \text{out of service probability} = \frac{\text{hours out of service}}{\text{hours in operation + hours forced out of service}}$$

$P + Q = 1$

The in-service probability of a series of mutually dependent components will then be expressed as the product of all of the individual in-service probabilities as follows:

$$P_s = P_1 \text{ X } P_2 \text{ X } P_3 \ldots\ldots\ldots\ldots\ldots \text{ X } Pn$$

The in-service probability of a sub system of two or more components in parallel, only one of which is required to be in service at any one time, will be expressed as one minus the product of the individual component out-of-service probabilities as follows:

$$P_P = 1 - (Q_1 \text{ X } Q_2 \text{ X } Q_3 \ldots\ldots\ldots\ldots\ldots\ldots \text{ X } Q_n) \quad \text{or}$$

$$P_P = 1 - (1-P_1)(1-P_2)(1-P_3) \ldots\ldots\ldots\ldots\ldots (1-P_n)$$

CAUSES OF JT4 SHUTDOWNS

	Total to Date	Considered Repairable Aboard Ship
Fuel Control Malfunction	14	14
Low Compressor Out Of Balance	3	
1st Stage Turbine Blade Failure	10	
3rd Stage Turbine Blade Failure	32	
No. 2 1/2 Main Bearing Failure	1	
No. 3 Main Bearing Failure	9	
No. 3 Main Bearing Seal Spanner-nut Failure	4	
No. 4 Main Bearing Failure	5	
No. 4 Main Bearing Seal Failure	9	
No. 4 1/2 Main Bearing Failure	1	
6th Stage Compressor Blade Failure	1	
Combustion Chamber Failure	5	5
Compressor Discharge To Fuel Control Tube Failure	7	7
Main Oil Pump Discrepancy	1	1
Fuel Pump Failure	9	9
Press. & Dump Valve Malfunction	1	1
Assembly Error — N_2 Compressor Rotor Hub Tube Lock Ring Missing	1	
Accessory Drive Shaftgear Failure	4	4
Accessory Drive Shaftgear Bearing Failure	3	3
Starter Drive, Coupling Lock Nut Backed Off	1	1
Tachometer Drive Shaftgear Failure	1	1
Accessory Drive Shaftgear Coupling Spline Failure	8	8
No. 5 Main Bearing Failure	2	
Accessory Drive Tower Shaft Failure	1	
Diffuser Case Cracked	1	
5th Stage Compressor Blade Failure	1	
#3 Bearing Seal Failure	2	
Loss of Oil-Cause Not Yet Determined	1	
Fuel Pump to Fuel Control Tube Broken	1	1
Fuel Control Drive Shaftgear Bearing Failure	2	2
Total	141	57

Ideally, all in-service probability figures would be determined from a statistical analysis of long term component operation. However, since this data is available only in isolated cases it has been necessary to estimate the in-service probabilities of certain apparatus.

A summary of component in-service probabilities is listed below. Detail calculations of these in-service probabilities is included.

1. Fuel Oil Service System — P_1 - 0.9995
2. Main Propulsion Gas Generator — P_2 - 0.9960
3. Main Propulsion Free Turbine — P_3 - 0.9950
4. Flexible Coupling — P_4 - 0.9995
5. Main Propulsion Generator — P_5 - 0.9969
6. Main Propulsion Generator Lube Oil System — P_6 - 0.9995
7. Salt Water Service System in Engine Room — P_7 - 0.9995
8. Main Propulsion Motor — P_8 - 0.9969
9. Main Propulsion Motor Cooling System — P_9 - 0.9940
10. Main Propulsion Motor Lube Oil System — P_{10} - 0.9995

11. Salt Water Service System in Motor Room — P_{11} - 0.9995
12. Thrust Bearing — P_{12} - 0.9990
13. Propulsion Controls — P_{13} - 0.9960
14. Ship Service Generators — P_{14} - 0.9969
15. Shafting and Propeller — P_{15} - 0.9965
16. Emergency Propulsion Drive — P_{16} - 0.9970

The in-service probability of the main propulsion system is expressed:

$$P_{mps} = P_1 \text{ X } P_2 \ldots\ldots\ldots\ldots \text{ X } P_{15}$$
$$= 0.9645$$

The in-service probability of the emergency propulsion system is expressed:

$$P_{eps} = P_1 \text{ X } P_7 \text{ X } P_{10} \ldots\ldots\ldots\ldots \text{ X } P_{16}$$
$$= 0.9874$$

The in-service probability for the combination of main and emergency propulsion is expressed:

$$P_c = P_1 \text{ X } P_7 \text{ X } P_{10} \ldots\ldots \text{ X } P_{15} \text{ X } [1-(1-P_2 \text{ X } P_3 \text{ X } P_4 \text{ X } P_5 \text{ X } P_6 \text{ X } P_8 \text{ X } P_9)(1-P_{16})]$$
$$= 0.9902$$

In-Service Probability Calculations

1. Fuel Oil Service System

Assume forced outage each pump - 7 days per 340 day year

Q (each pump) $= \frac{7}{340} = 0.02$

P (each pump) $= 1-.02 = 0.98$
P (each motor) $= 0.998$
P (each controller) $= 0.999$
P (each pump, motor & controller) $= 0.98 \text{ X } 0.998 \text{ X } 0.999 = 0.977$

As pumps are in parallel service. the in-service probability is P_1 * (both pumps, motors & controllers) $= 1 - [(1-0.977)^2] = .9995$

2. Main Propulsion Gas Generator

Pratt & Whitney statistical data: In 3,569,478 hours of operating there was one unscheduled shut down per 25,100 hours of engine operation. Forty percent of these shutdowns were of a type considered repairable at sea and 60 percent would require overhaul of engine.

Repairable breakdowns. 6 hours outage

Unrepairable breakdowns: If vessel on 4000 mile voyage mean nearest distance to either end of voyage is 1000 miles, ship would proceed to nearest port by emergency propulsion at 6.2 knots. New engine would then be installed in 24 hours.

$$\text{Outage} = \frac{1000}{6.2} + 24 = 185 \text{ hours outage}$$

See Figures Nos. 3 and 4.

MAIN PROPULSION RELIABILITY

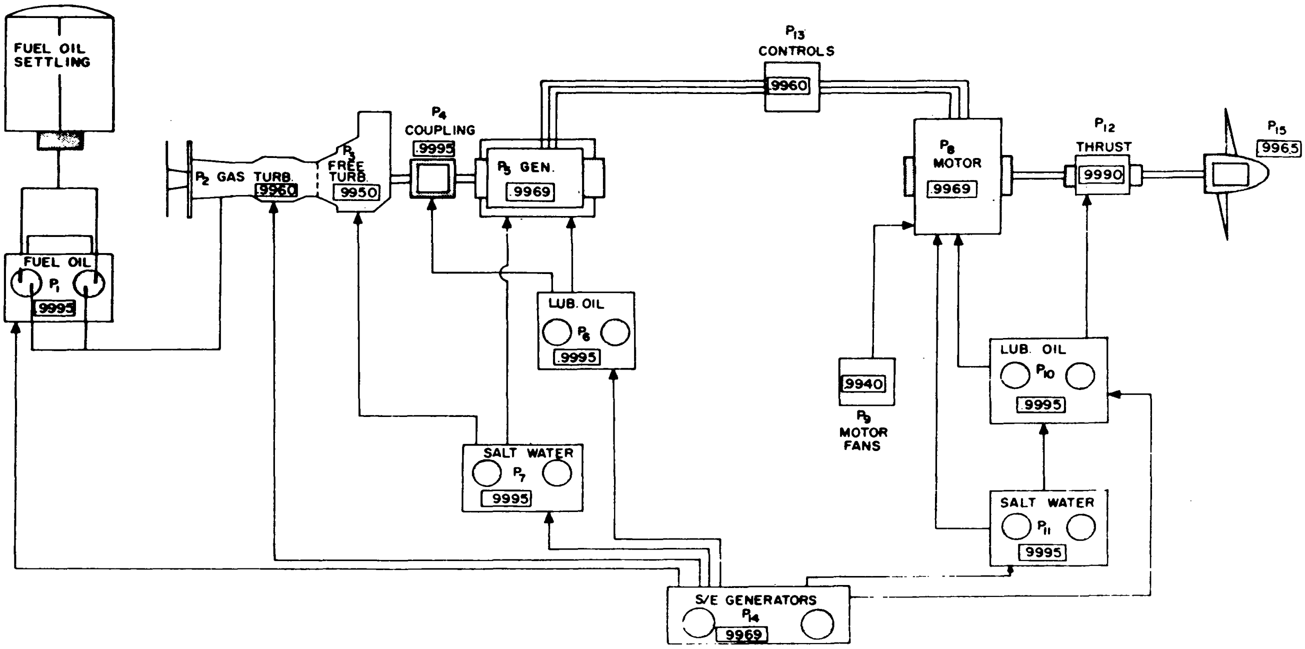


Figure 3.

EMERGENCY PROPULSION RELIABILITY

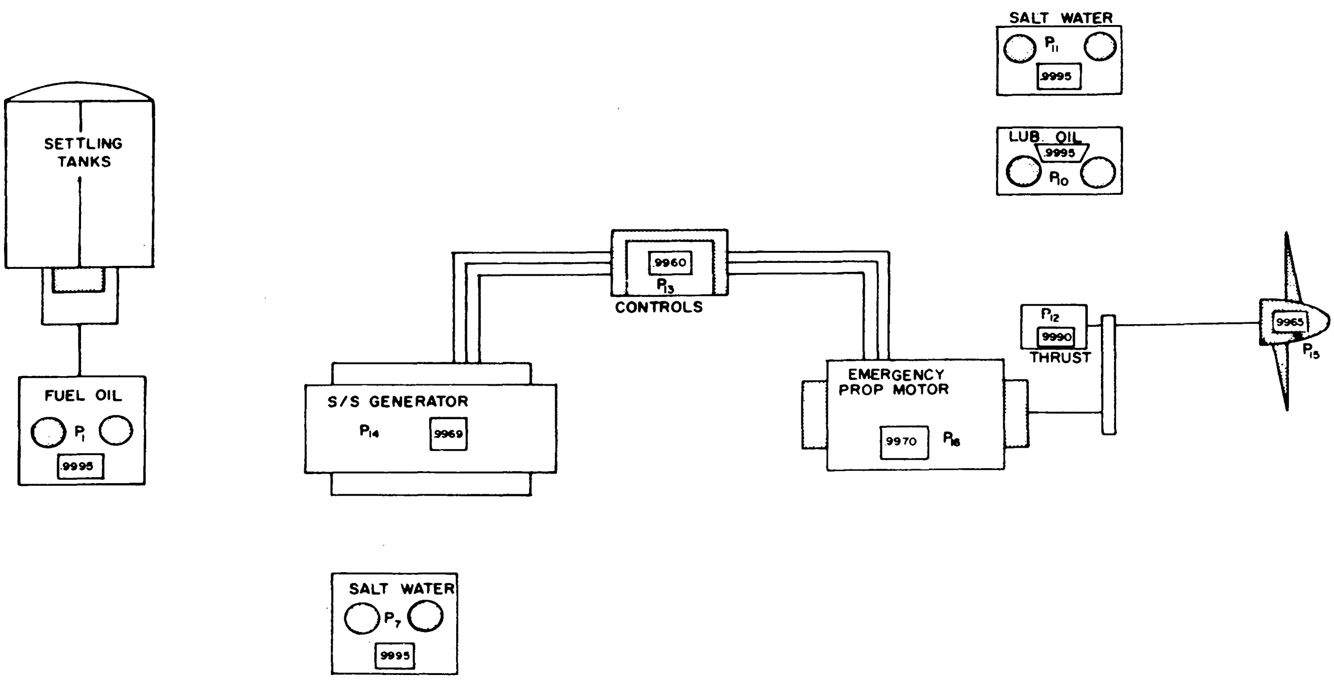


Figure 4.

$$Q_2 = \frac{57 \times 6 + 84 \times 185}{3,569,478} = 0.0044$$

P_2 for marine service = 1-.0044 = 0.996

NOTE: Outage hours due to vessel being towed or proceeding on emergency propulsion are included in gas generator reliability and will therefore not be included in other propulsion reliabilities.

3. Main Propulsion Free Turbine

Assume forced outage = 1 day per 204 days
$Q_3 = 0.005$
$P_3 = 0.995$

4. Flexible Coupling

Assume forced outage = 1 day per 2040 days
$Q_3 = 0.0005$
$P_3 = 0.9995$

5. Main Propulsion Generator

Westinghouse data for similar AC Synchronous machines indicate 272 hours forced out of service in 87600 hours
$Q_4 = 0.0031$
$P_4 = 0.9969$

6. Main Propulsion Generator Lube Oil System

P (shaft driven pump) = 0.98 (see F.O. Service System)
P (motor driven pump) = 0.977
$P_6 = 1- [(1-0.98)(1-0.977)] = 0.9995$

7. Salt Water Service System in Engine Room

P (motor, pump & controller) = 0.977
$P_7 = 0.9995$ (S.W. Service Pump is spared by Fire Pump)

8. Main Propulsion Motor

$P_8 = 0.9969$

9. Main Propulsion Motor Cooling System

P (each of 2 fans) = 0.997 (Westinghouse estimate)
$P_9 = 0.997 \times 0.997 = 0.9940$ (for full power operation)

NOTE: Main Propulsion Motor may be operated at up to 50% motor rating and 80% propeller speed on only one fan. In-service probability of Motor Cooling System at this rating would be 0.997.

10. Main Propulsion Motor Lube Oil System

P (each motor, pump & controller) = 0.977
$P_{10} = 0.9995$ (for parallel operation)

11. Salt Water Service System in Motor Room

P_{11} = 0.9995

12. Thrust Bearing

Assume forced outage = 2 days per 4080 days
Q_{12} = 0.0010
P_{12} = 0.9990

13. Propulsion Controls

P reversing contacts = 0.999
P circuit breakers = 0.999
P drum switch = 0.999
P other controls 0.999
$P_{13} = (0.999)^4 = 0.9960$

14. Ship Service Generators

P (each AC generator) = 0.9985 (Westinghouse estimate)
P (each DC generator) = 0.995 (Westinghouse estimate)
Assume forced outage of diesel engine or gas turbine = 5 days per 100 days

P (each prime mover) = $1-\frac{5}{100} = 0.9500$

P (AC generator, DC generator & prime mover) = .9985 X 0.995 X 0.95 = 0.944
$P_{14} = 1-(1-0.944)^2 = 0.9969$

15. Shafting and Propeller

P_{15} = 0.9965 (from Maritime Administration)

16. Emergency Propulsion Drive

P (motor) = 0.998 (Westinghouse estimate)
Assume forced outage of reduction gears = 2 days per 2040 days
P (reduction gears) = 0.999
P_{16} = 0.998 X 0.999 = .9970

G. SUMMARY

In conclusion, the following points are emphasized which lend confidence in the over-all in-service probability of a marine power plant utilizing a well-proven aircraft gas turbine as the prime mover:

1. The reliability of a power plant is dependent upon the quality of fuel supplied to the engine, which in turn is dependent upon the quality of fuel at the source and the effectiveness of treatment methods to obtain the necessary quality of fuel for long, uninterrupted service.

2. The PWA FT4 marine engine is a simple engine. It has a high reliability factor based upon millions of operating hours experience and upon thorough feed-back of data on unscheduled shutdowns.

3. Use of electric drive gives confidence in a high in-service probability because of the known high reliability and extensive service experience with electric drive.

4. A high in-service probability for the entire propulsion plant is more assured as the number of components comprising the plant is decreased.

5. The reliability of automation of a simple plant such as an aircraft gas turbine is greater than that of a complex steam or diesel plant.

In addition, the marine engineer is dependent on any in-service probability analysis on the validity of reliability values assigned to each component. Until reliability data is available on all components making up the power plant, the marine engineers cannot be specific or have assurance of the calculated value of the in-service probability of the complete power plant. Furthermore, the marine engineer must always be aware in his analysis that confidence in any component reliability is a function of the total hours of operation of a class of components and also is a function of a degree to which failure information is reported back to the manufacturer.

There is no doubt that the use of proven components like the Pratt & Whitney FT4 aircraft type gas turbine and the development of a plant with the fewest supporting auxiliaries is the direct and immediate route to a highly satisfactory, highly reliable and economic main propulsion plant.

ACKNOWLEDGEMENTS

In the preparation of the paper of this nature where there is no standard technique on marine power plants for treating the subject, it is necessary to consult with a large number of individuals and search through extensive literature. In particular, I wish to express appreciation to Pratt & Whitney Aircraft for consulting with me in the preparation of a report to the Maritime Administration on an "Integrated Marine Gas Turbine Power Plant" utilizing aircraft type engines. Further, I wish to acknowledge the valuable background gained from the following papers:

1. ASME — Research Publication	"A Review of Available Information on Corrosion and Deposits in Boilers and Gas Turbines."
2. B. G. A. Skrotzki	"Energy System Economics" Power Special Report — December, 1961.
3. B. O. Buckland D. G. Sanders	"Modified Residual Fuel for Gas Turbines" ASME Paper No. 54-A-246 — December, 1954.
4. E. J. Tangerman	"A Manual of Reliability" Product Engineering — 1960.
5. H. Pfenninger	"Operating Results with Gas Turbines of Large Output" ASME Paper No. 62-WA-188.

In addition, information was taken from a paper by the author entitled, "Problems Associated with the Use of Bunker "C" Fuel" — SNAME, New York — December, 1959.

DISCUSSION

BERMAN (Westinghouse): In determining the gas turbine fuel rate, did you include the electrical transmission losses, and what is the turbine inlet temperature?

MC MULLEN: Yes, it does include electrical transmission losses. Actually we're using Pratt and Whitney J75 unit with a total shaft output, calculated with all the losses in it, of 20,000 shaft horse power.

The turbine inlet temperature for this particular turbine as I recall was 1505°. Which is a fairly conservative rating in relation to what it's exposed to in the aircraft industry. As you know, on take off, these units go up to somewhere around 1700°, I believe.

LARSON (Pratt & Whitney): Dr. McMullen, may I clarify a point in case there are those who are not familiar with the Pratt & Whitney Aircraft gas turbine engine. This consists of a J75 gas generator with an attached Free Turbine. The turbine inlet temperature which is being discussed is that at the gas generator turbine inlet, the turbine driving the compressor. It is not the temperature at the inlet to the free turbine.

MC MULLEN: That is correct. The basic unit of the aircraft unit consists of a compressor, a combustion system and a compressor turbine, and this is the unit that we are taking as a basis for the power output. Then we're adding a free power turbine which will be separate and distinct. The basic reason for going to a concept such as this is the fact that the marine industry in itself could never really afford the cost of developing a unit of this reliability and this output on its own.

After all in the United States we build perhaps 50 and 75 marine power plants per year. Whereas this aircraft type unit is turned out in quantities which approach 30 and 40 per day.

HAUSCHILDT (Bureau of Ships): I was wondering if you were considering any changes in the materials in the gas turbine as a result of the fact that it would operate in a marine atmosphere? Salt Air?

MC MULLEN: The word marinize has been devised to describe what would be done to the unit itself. This would consist internally of practically nothing and externally only as it affects the shell, the materials which are used and the housing of the unit itself.

FRANKEL (MIT): First of all wouldn't this also, this hostile environment, very much effect our 2,200 hour period between major overhauls which, we hope, with lesser service stress tendencies in the marine field up to 8,000 hours.

My second question is just comparing your reliability figures, which I believe, are manufacturing salesman figures, and very optimistic even from their point of view, for comparable systems with those given to us today by Mr. Riddick of Newport News from actual ships in comparable systems for instance. Considering feed pumps and salt water systems and translating the reliability figures into mean time between failures, yours are just about one order of magnitude out.

MC MULLEN: I'll start with your last question first. First of all of course you know we don't have any feed water pumps, nor do we have any salt water system involved in the main propulsion of the ship. So for this reason we've eliminated these completely from consideration from a standpoint of reliability. In so far as the accuracy of the reliability figures are concerned I believe as I recall from this morning's presentation that Mr. Riddick's numbers were not based on any statistical information, but simply upon a record which was kept in a repair yard, Newport News, and extrapolated into reliability information.

Now if you noticed, we reduced the number of components in our plant to 11 items to which we had to assign reliability factors and, with the exception of two of these, the numbers were based upon fairly good statistical information.

For example, there can be little doubt about the percentage assigned to the electrical system, the gas generator and the shafting and the propeller. Then, the percentages assigned to the salt water lube oil cooler were rather acceptable ones.

FRANKEL: It is just the last two ones which I consider very high.

MC MULLEN: You mean on the coolers.

FRANKEL: Yes.

TILLSON (General Electric): Will you comment briefly on how you furnish your excitation power and also some of your other ships services.

MC MULLEN: Basically on furnishing the ships services, as I mentioned, we eliminated the concept of the waste heat boiler for several reasons. First of all, it introduced a pressure loss in the system and thereby reduced the power output of the overall unit.

Secondly, we felt that the introduction of any steam system into the ship would not only complicate the overall system, but also perhaps complicate negotiations for reduced manning of the ships. So we took care of all the heating and other services of the ship with electrical power, and we felt that as a result of the combined effect of reducing the number of people, and reducing the requirement that this was quite acceptable. So we considered that all the ships service power would be electrical. We eliminated evaporators from the ship because we don't have a requirement; the only water we need is potable water in this ship. Thus, the gas turbine permitted the removal of many of the problems and many of the ships service functions which result in additional problems, so we considered all of the ships services to be of electrical power.

Excitation was supplied by running this generator right off of the ship service generator sets for excitation of the main propulsion plant.

SULLIVAN (Maritime): First, I would like to say that your questioning Bob Tillson on the Sergent reminds me more of Eisenhower asking Kennedy what does he think of the qualifications of Dick Nixon.

John, also I thought that was a dandy question where you asked Bob about if he had to pick the two weakest links from a reliability standpoint, which would they be. I know you have numbers on most of your components, but forget the numbers for a moment, which do you think would be the weakest within the aircraft design?

And then a second question: do you think there is a chance that we can burn residual fuel in the future, and if so, what should we be doing about it now?

MC MULLEN: To answer your first question, the weakest link of course in overall reliability is the fuel oil. Or do you wish to confine it to the mechanical end?

SULLIVAN: To the machinery end.

MC MULLEN: Well, let me just say first again, the weakest link of course in reliability is the quality of the fuel oil. I think that in all other reliability studies which are even made with steam and diesel plants that they must include some type of qualification or some type of definition of the fuel oil which is going to be used.

From a mechanical standpoint in the overall system, I would imagine the weakest place in the system might have to do somewhere with the excitation of the main generator, having been run off one of the ship service generators. However, I think having two generators reduces somewhat the loss of excitation.

HIRSCHKOWITZ (U.S. Merchant Marine Academy): One of the things that perhaps I'm misinformed of in your program is that you could easily remove an engine from the ship. Do I get the impression that a fleet operator would have to have some engines in storage to take care of this operation?

MC MULLEN: Not necessarily. At the present time, as far as I know, the policy of Pratt and Whitney, if they sold a steamship operator an engine, would be to guarantee him a backup unit.

BAZOVSKY (Raytheon): I would like to point out a falacy in presenting reliability figures if you don't quote the time to which they apply. You gave us reliability numbers. Now to what operating time do those reliability numbers apply?

MC MULLEN: The operating time on the gas generator was that 3,500,000 odd thousand hours of operation.

BAZOVSKY: No sir, I mean reliability is being defined as probability of survival. You say it is 96% for the overall plant. Is it for an operating period between the two thousand hours?

MC MULLEN: No, this is on an annual basis.

ROGERS (Esso International): John, this morning I have heard several comments about reliability and know when you go to the manufacturers that you cannot get good data on the subject. I've been an operating man in this marine field for about 30 years, and I can't remember once in 30 years, any manufacturer coming to me and asking me how his equipment was working. How in the world can they feed reliability factors to you when they do not know what we are doing with their equipment in Timbuktu or someplace else when they do not call on the user for his experience.

MC MULLEN: Actually this is a good point and I think it is really one of the reasons for this entire session being held if I am not mistaken. I think the theme of this present meeting is to try to attempt to do something about this very problem.

ARNOLD (United Aircraft): To that last question, there has been so much talk in the conference about the fact that data hasn't been available and this has been one of the problems that I thought it might be useful to send in to Professor West the description of the system that the Pratt & Whitney aircraft people do use to gather data. This system is needed because of the very high reliability required for manned aircraft. It may not be entirely practical in the marine industry but it may give people in this field some ideas of how to gather better data. Part of it I might add is obtained by automatic recording.

MC MULLEN: However, the point I wish to make and I think that Mr. Rogers is bringing up is that we have to be careful and not have the cost of this accumulation of information exceed its value. The advantage the aircraft gas turbine has in so far as the marine industry is concerned is again the point I stressed before; namely, it's available to us in its form with this statistical information, because of certain circumstances which occur in the aircraft industry.

FIXMAN (Maritime): The Maritime Administration is mailing a letter to all kinds of research people soliciting letters of interest to undertake a study to try and develop a system whereby we can get some of the data we want from operating cargo ships. For you people in the audience who are interested in this program, just address a letter to the Maritime Administration. We would like to find out who is interested in doing the work.

MC MULLEN: Incidently I didn't answer the second part of Mr. Sullivan's question concerning fuel oil. Basically in so far as the aircraft type of gas turbine is concerned and recognizing the spectrum of the various fuels and so forth, I have no doubt that we will be able to burn marine diesel fuel. I think this will be acceptable and when this occurs the annual increase fuel oil cost on a ship such as the Reference Ship of the Maritime, 20,000 shaft horsepower, would be approximately 40% to 50% higher than they would be if they were burning straight Bunker C.

As far as any future work is concerned, I think there are ways these fuel costs can be reduced. I think that if the need is there, and if the application is determined in the near future, the aircraft gas turbine can get its fuel costs down to a comparable level. Even so, if this fuel cost does remain 40% to 50% higher, I think that from the standpoint of an American Flag operation, with the combined effect of the initial reduction in costs of the construction of the ship and the reduced crewing, the gas turbine is already competitive with that of any other type.

CONTRACTING FOR RELIABILITY

JOHN DE S. COUTINHO
Chief of Reliability Control
Grumman Aircraft Engineering Corporation

The title of this paper implies that you can buy reliability like you buy anything else, say a house or a car. Such of course is not the case. Reliability is not a commodity; it is a product characteristic, like weight or size, and cannot be dis-associated and bought separately from the product.

Nor does the technical concept of buying reliability apply to all types of products. Generally, we do not have much of a reliability problem with products which are already designed and developed, especially those available on a competitive market. If you are dissatisfied with one brand, you switch to another, and the competition forces suppliers to maintain at least a minimum standard of quality within a given price range. If a supplier does not conform, he loses customers and his trade association will get after him.

Reliability becomes a real concern with products which you cannot buy off the shelf, products which are specifically designed and built to order, particularly very complex new weapon systems which are advancing the state of the art.

A real problem arises because of the difficulty in specifying the desired level of reliability in the contract and measuring the attained reliability on acceptance. When you specify that a product shall not weigh more than so many pounds, or shall not take up more than so much space, you can easily measure these characteristics on acceptance. If the product does not meet the specification, it can easily be rejected without much argument, or the supplier can be subjected to an agreed-upon financial penalty.

Reliability is not as easy to specify and to measure on acceptance as weight or size. Furthermore then are two other overriding considerations associated with complex new products and systems: single source procurement and the delivery schedule.

Complex new equipment is very costly and competitive parallel development is generally out of the question. Except in the most unusual circumstances, competition exists only in the proposal stage. Once a supplier has been awarded a contract, he becomes the sole source for the item. We immediately face the serious question of how to motivate a single source supplier to produce a reliable product in a situation where the more reliable is his product, the fewer units he will sell. In the aircraft and engine business, it has been the overriding concern with safety of life which has provided the motivating force to supply a reliable product. Whenever there is single source procurement, and safety of life is not involved, the customer must provide an effective policing action to assure that he gets a reliable product.

The need for such action is magnified by the necessity for maintaining a fixed delivery schedule. Critical new components of new systems, such as new airframes and new engines, are concurrent developments which must be completed on a given date so that they can be brought together systematically. Slippages of delivery dates are of critical importance and cannot be tolerated. If on the scheduled delivery date a single source item not involving safety of life does not have the required reliability, there is very little the customer can do about it. It is too late to use his only club, the ability to transfer all or part of the contract to another supplier.

These problems become all the more important the larger the project. The men in charge of large weapon system developments or space projects are responsible for the expenditure of billions of dollars of public funds, and they need to be assured that the single source product which they are buying will do its job.

Since reliability measurements by themselves as part of the acceptance procedure are largely ineffective, it has become customary to specify a reliability control program during design and development. The supplier agrees to do certain things at specified times which will assure that when the product is finished it will have the desired reliability.

Present contracting procedures have a number of deficiencies. Normally a supplier has a fixed program in mind when he submits a proposal. The customer usually does not have adequate procedures for evaluating the supplier's proposed reliability program, and furthermore under the pressure of having to evaluate a number of proposals in a short period of time, the proposed reliability program never is properly examined and evaluated, and the customer never finds out what the supplier is proposing. However, once the contract is signed, the supplier feels that the program he was thinking of at the time of signing the contract is the legal agreement. This intended program may or may not be a satisfactory program, but now there is little the customer can do about it.

It is therefore essential that the supplier's intentions be fully explored and that there be an adequate meeting of the minds before signing a contract. This requires of the customer that he have formulated satisfactory standards for a reliability program and that he has at his disposal a staff of reliability engineers who are competent to monitor and police the supplier as the program evolves.

At this point it might be worthwhile to determine whether these principles are new, or whether programs of this type exist in other areas. In actual fact, the procedure I have just described is quite standard in other technical areas; probably the most clean-cut and successful such program is the one dealing with the structural strength of the airframe. This area is also taken very seriously because safety of life is involved. Here the contractual design monitoring procedures are well established and uniformly accepted.

For each class of aircraft, the customer has developed a set of specifications which establish the required strength; the design safety factor (usually 1.5); the acceptable materials and the properties that may be assumed for design purposes (MIL-HDBK-5), and the acceptable analytical procedures for computing margins of safety. A positive margin of safety must be shown for each structural element. Whenever the established methods appear inadequate to solve a special problem, a report describing a proposed procedure is submitted to the customer for approval.

Key structural assembly drawings are submitted to the customer for approval before further work can progress, and all manufacturing drawings must be signed by the supplier's stress analyst and the customer's representative before the part can be manufactured at the customer's expense. Test plans to demonstrate compliance with the design requirements are submitted to and approved by the customer before the test is run, and the test itself is witnessed by the customer's representative. A schedule is established for the submittal of analytical reports, key structural drawings, test plans, and test reports; and this schedule is phased with the projected progress of the design and development. Tests are scheduled as early as feasible.

If a required document is not submitted on schedule, the customer's accepting authority may be notified that the final product is contractually in an unacceptable status, and normally the deficiency is quickly corrected.

This description is highly simplified, but it demonstrates a successful approach for monitoring a development program to assure that the final product will meet the design requirements. I would like to remind you that these procedures were not established without a real struggle, and have been accepted practice for only the last 25 years.

The step from structural integrity to reliability of other type devices is only a short one. In structural analysis of static strength there is only one variable, namely load. The margin of safety is determined by dividing two numbers; strength by ultimate applied load. When more than one variable is involved, say load, rpm, temperature, and contamination, and each is capable of causing the unit to fail, the problem becomes complex and we must resort to statistical analysis. However, such statistical techniques are to a great extent already available today. Instead of a simple margin of safety, the index of strength is the probability that the unit will successfully withstand a given level of combined environments for a given length of time; this is our mathematical definition of reliability.

This derivation of the reliability concept from the static margin of safety was proposed by R. Lusser almost 10 years ago and was incorporated into the missile specifications issued by Redstone Arsenal. The approach has been applied to some of our most successful missile programs.

The Redstone specifications require that a device be tested to failure under judicious combinations of systematically increasing environmental stresses. Sufficient tests must be run to determine the distribution and standard deviation for each mode of failure, in order to ascertain the strength margin over the peak anticipated operating environments. Minimum acceptable margins are specified. Each test to failure may not require a new unit, if the failed unit can be repaired and retested; but at least two units should be used.

We are now proposing that this type of testing be extended into the time domain by running a series of tests to failure for at least two points in time: on new devices, and then on units which have survived an operating life cycle at peak operating environments.

These tests to failure can be run relatively early in a development program; in fact the development testing program should be designed to yield as much of the required information as possible, so as to cut down on the number of additional tests needed to satisfy the reliability demonstration requirements.

The purpose of these tests-to-failure is to establish the strength of components under combined environments. Subsequent system tests are then required to establish that the actual peak combined operating environments on the components do not absorb the required margins of safety and reduce the reliability below the required level.

Although this concept of a formal safety margin as an analytical reliability tool is not new, it is still almost unknown to many of our suppliers. In the electronic industry we have just completed the transition from tubes to semiconductors without any appreciable gain in system reliability. Although the semiconductor itself has a high inherent reliability, almost all of the technological advance on the systems level has been exploited to save weight and to reduce space requirements; these are hard, saleable items. The competition for even greater performance is such that whatever safety margins do exist, they are very small. The great effort is to get one or two pieces of equipment to work in the laboratory and to pass the qualification test. When the equipment later fails in service, it is then obviously due to poor quality control or inadequate maintenance. Although quality control and service groups often take the blame, and are active in reliability and have a contribution to make, neither of these two groups can get to the heart of the problem; this is still a design function.

I would also like to point out certain defects in our present system of qualification testing. Normally one unit is required to undergo a specified environmental spectrum for a given period of time without failure. The test environments are

TABLE I - ABBREVIATED HEADINGS OF A GENERAL SPECIFICATION FOR RELIABILITY AND MAINTAINABILITY PROGRAM REQUIREMENTS

3.	Requirements
3.1	General
3.2	Reliability and Maintainability Control Program
3.2.1	Program Management
3.2.1.1	Policies, Instructions and Standards
3.2.1.1.1	Organization and Procedures
3.2.1.1.2	Monitoring Procedures
3.2.1.1.3	Review of Procedures
3.2.1.2	Reliability and Maintainability Control of the Design Phas
3.2.1.2.1	Management Tasks
3.2.1.2.2	Design Review
3.3	Reliability Requirements
3.3.1	Reliability Apportionment
3.3.2	Reliability Estimates
3.3.3	Environmental Effects on Reliability
3.4	Configuration Analyses (Trade-off studies)
3.5	Circuit Analysis
3.6	Applied Loads and Stress Analyses
3.6.1	Applied Loads and Environments
3.6.2	Distribution of Internal Loads
3.6.3	Selection of Materials and Sizes of Structural Members
3.6.4	Calculation of Margin of Safety
3.7	Failure-Effect Analyses
3.7.1	Part Failure-Effect Analyses
3.7.2	Failure Simulation
3.8	Maintainability
3.8.1	Maintainability Analysis
3.8.2	Availability
3.8.3	Repairability
3.8.4	Maintenance Diagrams
3.8.5	Maintainability Estimates
3.9	Approved Parts List
3.9.1	Component Parts Testing
3.9.1.1	Part Production Tests
3.9.1.1.1	Scope of Tests
3.9.1.1.2	Test Plans and Reports
3.9.1.1.3	Test Plan Summary
3.9.1.2	Part Acceptance Tests
3.9.1.2.1	Acceptance Test Plans
3.10	Surveillance of Test Program
3.10.1	Review of Test Plans
3.10.2	Test Reports
3.11	Fabrication and Quality Control Surveillance
3.11.1	Review of Manufacturing Plan
3.11.2	Review of Quality Control Plan
3.11.2	Review of Procurement Plan
3.12	Failure Reporting and Analysis System
3.12.1	Reporting Forms
3.12.2	Analysis of Failures
3.13	Reliability and Maintainability Reports
3.13.1	Subcontractor Internal Reports
3.13.2	Subcontractor External Reports
3.13.2.1	Reliability and Maintainability Control Program Report

TABLE II - SCHEDULE OF REPORT SUBMITTALS

Paragraph No.	Title	Type*	Submittal Date: Initial submittal	Submittal Date: Updating
3.13.2.1	Reliability and Maintainability Program Plan	R	30 days after authorization to proceed	As changes occur
3.13.2.1.1 3.13.2.1.1.1	Monitoring Procedures Report	R	30 days after authorization to proceed	Quarterly
3.13.2.2	Preliminary Reliability Report	N	90 days after authorization to proceed	
3.13.2.3	Reliability Estimate Reports	R	4 months after authorization to proceed	Monthly (ref. par. 3.13.2.14)
3.13.2.4	Environmental Effects Report	R	60 days after authorization to proceed	As changes occur
3.13.2.5	Configuration Analysis	R	10 days prior to design reviews of paragraph 3.2.1.2.2	As changes occur
3.13.2.6	Circuit Analysis	R	20 days prior to start of breadboard model (par. 3.2.1.2.1)	As changes occur
3.13.2.7	Applied Loads and Stress	R	10 days prior to design reviews of paragraph 3.2.1.2.2	Monthly, as changes occur
3.13.2.7.1	Summary of Structural Test Plans	R	30 days prior to start of first test	As changes occur
3.13.2.8	Failure-Effect Analysis Report	R	30 days after final design freeze	As changes occur
3.13.2.8.1	Failure Simulation Test Plan Summary	N	As required by paragraph 3.13.2.8 and 3.13.2.8.1	
3.13.2.9	Maintainability Analysis Report	R	Same as par. 3.13.2.15 (see schedule below)	Monthly**

*R = Recuring, N = Non-recurring

** All MEAR forms of paragraph 3.13.2.15(a) shall be submitted no later than 30 days subsequent to final design freeze.

			Submittal Date	
Paragraph No.	Title	Type*	Initial submittal	Updating
3.13.2.10	Approved Parts List	R	With Preliminary Reliability Report	Monthly, as changes occur
3.13.2.10.1	Test Plan Summary	R	30 days prior to start of first test.	
3.13.2.11	Preproduction Test Plans	N	60 days prior to start of test	
3.13.2.11.1	Preproduction Test Reports	N	Summary report by telephone or TWX within 48 hours of completion of test; final report no later than 30 days after completion of test	
3.13.2.12	Acceptance Test Plans	N	60 days prior to start of first test	
3.13.12.1	Acceptance Test Reports	N	Summary report by telephone or TWX within 48 hours of test completion; final written report no later than 30 days after test completion.	
3.13.2.13	Failure Reporting and Analysis System Report	R	60 days prior to start of first development tests	As changes occur
3.13.2.13.1	Failure Reports	N	One (1) week after first test in which failure occurs, and weekly thereafter	
3.13.2.13.2	Analysis of Failures	N	As appendices to monthly reports of 3.13.2.14	
3.13.2.14	Reliability Status Report	R	30 days after authorization to proceed	Monthly
3.13.2.15	Maintainability Status Report	R	60 days after authorization to proceed	Monthly
4.2.1	Development Test Plans (other than listed above)	N	30 days prior to start of test	
4.2.1	Development Test Reports (other than listed above)	N	No later than 30 days after completion of test	

* R = Recurring, N = Non-recurring

usually established before any systems tests are run, and only by chance would the test conditions coincide with the maximum operating environments. Since the test specimen survived without failure, the true strength of the item is not established, particularly not in the actual operating environment.

If say three competitive items all pass the same qualification test, there is no way of knowing which one is the best one or the strongest one. Further complications arise when the qualification test specimen is carefully manufactured for test purposes, and the test is conducted by technicians who are expert in the control of test environments and in nursing their product through qualification tests. Normally there is no way of determining the significant differences between production units and the specimen which passed the qualification test.

Qualification testing, therefore, adds little or nothing to reliability demonstration. Under these circumstances I believe that qualification testing should not be started until all reliability and maintainability requirements have been satisfactorily demonstrated. In fact, maintainability should be demonstrated first, on a mock-up, before start of reliability testing. I emphasize this order of testing because it is in direct conflict with the AGREE report which serves as the Bible for so many reliability engineers, and which specifies that reliability tests be run after qualification. This order of testing is acceptable for the after-the-fact reliability measurement of developed and accepted products. But in the case of new developments, the order of testing must be reversed, and the inherent reliability must be demonstrated before the start of the formal acceptance procedures.

In designing a reliability test program, it is necessary to understand under what circumstances statistical techniques must be resorted to. If a device is designed in accordance with known laws of nature, only one test is necessary in order to confirm the analysis. For example, if a design is based solely on the law of gravity, one test is sufficient to determine that no mistakes have been made in the application of the law, and the reliability of the device is 1.0. The principles of structural design, for example, are sufficiently well established that only one destruction test is made of an airframe. If the test confirms the analysis, the analytical results are accepted as the strength of the airframe; not the test results. Many engineering devices fall in this category, and their reliability can be determined analytically, and the analysis confirmed by one test (usually a test-to-failure).

In developing a new steel alloy, however, the relationship between the tensile strength and the new alloying elements may not be understood, and it is necessary to resort to statistical testing. The statistician assumes that the measured tensile strengths of an infinite population of test samples can be represented by a specific frequency distribution curve. He requires a sufficient number of test points to determine the parameters defining this curve.

Most elements of most devices are designed in accordance with known laws of nature, and their reliability is 1.0. Many devices include a few critical elements about which there is some degree of doubt. Usually the situation is not as clean-cut as in the case of a new steel alloy. In developing a reliability test plan, it is necessary to know what elements of a device are designed in accordance with laws of nature, and what aspects of a design contain unknowns. Some parameters of a device can be determined in one test, others may require tests on two units, and others may require 6, 30, or more tests. The design of a good reliability test plan requires an intimate knowledge of the product and of the state of the art.

The difference between an expensive reliability program and an economical one is normally the amount of testing. The approach proposed here is to obtain through careful planning the maximum amount of data from all testing, and to minimize those tests performed for reliability demonstration only. This should result in an adequate program with a minimum of testing and hence least cost.

By redefining reliability as an extension of the margin-of-safety concept, and by basing reliability testing on tests-to-failure, it is possible to establish a reliability monitoring program analogous to the one described above for structural design, and incorporating similar safeguards.

An essential element in a reliability program is for the customer to know what he wants. We at Grumman have been working for several years developing subcontractors specifications incorporating the elements discussed above. At the time of this writing I am attempting to make these specifications effective for the first time on the Lunar Excursion Module of Apollo, and on the Grumman portion of the Interservice Tactical Fighter TFX.

The next task is to assure that our suppliers read and understand the provisions of our reliability specification, and have the capability to satisfy its requirements, before we sign the contracts. We have developed a vendor questionnaire which we hope will help us to detect vendor weaknesses before it is too late.

The headings of a preliminary copy of this specification appears in Table I. The specification requires a heavy analytical effort; no test can be run tunil the device is thoroughly analysed and those aspects which are not amenable to analysis are well defined.

The supplier is required to submit a reliability program plan tailoring our general requirements to his product, defining his organization and his level of effort, his coordinated design freeze points and scheduled design reviews. A schedule for report submittal is included and is shown here in Table II.

Documentation such as listed in this Table is an important part of a reliability control program and should be subject to careful pre-contract agreement. Documentation serves two purposes; to keep the customer informed of the status of the program; and to record the basis for design decisions in those areas where failures may occur. Care must be taken that within these bounds, documentation is kept to an absolute minimum, since the documentation associated even with the best reliability control program tends to become overwhelming.

To give the supplier an idea of an acceptable product, a list of do's and don't's is included in our specification for guidance. This list is kept up to date, and includes much information normally not included in a specification.

It is a critical matter to determine, before the contract is signed, that the supplier's reliability group enjoys the confidence and support of its management. It is important that management insist that reliability engineers stick to tasks that improve the reliability of the product. Reliability analysts must review and sign off on every manufacturing drawing, material specification and purchase order. If a reliability estimate is low, the reliability analyst must have enough organizational stature to get the device redesigned, the schedule readjusted, or the design objective lowered.

DISCUSSION

STEFANCIC (Allis-Chalmers): Having just prepared a technical study for a program of Grummans, one of the things I was concerned with is just how the study is evaluated. You've mentioned a lot of points, but at this time Grumman only asked Allis-Chalmers to make a prediction for a system, and I understood that you make your initial evaluation of the sub-contractor based upon this technical study of the system. You haven't mentioned this technical study in your paper and I was just wondering if you would elaborate on that point.

COUTINHO: If you have been invited to bid by Grumman, you have received a copy of this specification, and we have asked for your program plan in your proposal.

STEFANCIC: It was not a formal proposal request, it was just a request for a technical study in which the first thing wanted was an examination of the technical system Grumman was attempting to describe. There was no effort made to ask about the management portion at this time nor any of the other normal aspects that would go with a full proposal request.

COUTINHO: We have a package of documents that we send to companies being invited to bid, and this package includes a reliability specification that covers all pertinent points. We ask for their reliability program plan, in a certain format, and the proposal will be evaluated on that basis.

STEFANCIC: I'm asking, what the aspect of the technical study is, which occurs prior to the proposal. You've told me what happens during the proposal part, but what I'm still trying to understand is what the effect of doing just the technical study is prior to the request for a proposal.

COUTINHO: I presume that you are asking about the preliminary technical review of vendors. Many companies come in to acquaint us with their technical capabilities. This is primarily a review by technical people to select those who will be invited to bid formally, and in such cases the reliability people are not always in on it. It is purely a technical review for our information.

HUGHES (Allis-Chalmers): I have a question along the same line, but I can make it general. In this specific instance we were asked for specific failure rate data and also our source of data. Why one way? Can't this be in general made a two way street. When a sub-contractor applies failure rate data to the best of his ability the prime contractor should supply in turn any data that he has: Data that the sub-contractor admits he does not have for his reliability analysis.

COUTINHO: We do just that.

HUGHES: How?

COUTINHO: Grumman has participated in the FUR (Failure, Unsatisfactory, Removal) program of Bu Weps since its beginning in 1954. The Navy FUR program requires that whenever a part fails or an unsatisfactory condition exists on any aeronautical material in the Naval Air Service, that a FUR form be filled out, describing the deficiency.

There has to be a monitoring procedure on such a requirement and the policing is done at the stock room window. You cannot withdraw a new part from a navy stock room without making out this form. The system is not foolproof. A mechanic may have a supply of O-rings in his tool box, and when he repairs an actuator and puts it back on the airplane he does not have to go to the stockroom. It becomes difficult to assure that he makes out a report in this case. When a man withdraws a new part from the stockroom, it is difficult to evade the paper work.

These FUR reports are submitted daily to the Naval Air Technical Services Facility (NATSF) in Philadelphia and are punched on IBM cards. I get a copy of each IBM card. In fact Grumman has so much equipment in the field that I get 10,000 IBM cards a month. I have to mention how much equipment we have in the field because I don't want to leave the impression that our equipment is bad just because we're getting 10,000 failure reports a month. I have been accumulating this data since 1955 like a little hamster. I have therefore good failure rate data on all kinds of aeronautical equipment. We do make this material available to our vendors whenever either they ask us for it, or when we see a need to make it available to them for some specific purpose. So failure rate data is available, and all six prime Navy airframe contractors participate in this FUR program and do the same thing. This program is Navy-wide. NATSF at Philadelphia has even better data than I have and is always glad to make it available to anyone supplying products to the government.

The FUR program has deficiencies and there are many people who will gladly list the many things that are wrong with the FUR program. I believe this is mainly because it doesn't happen to solve their problems for them. However, I would like to assure you that there is a lot more good with the Navy FUR program than there are deficiencies; but of course we have to admit that any program can be improved. The Navy FUR program can also be improved, but it's already doing a very remarkable job. In particular it affords us our first opportunity to monitor every piece of equipment that we have in service, and let us know as soon as anything goes wrong.

The timeliness of these reports is quite good. 90% of all reports that I get are less than 6 weeks old, and 80% of the reports are less than 1 month old. So I'm pretty much on top of what is going on. I can coordinate failures in the Mediterranean with those in the Pacific. I watch this and if I want more information all I have to do is pick up a phone. We have service representatives all over the world at all Navy installations. If they can't cope with the situation, they are in daily communication with the plant, we send out technical personnel of as high a technical level as is required to conduct the investigation.

The complaint that we're not getting enough data simply isn't justified. We can get as much as we want and as much as we need to do our specific jobs.

ARNOLD (United Aircraft): I would like to ask you a question in reference to this stress to failure concept that you were talking about a moment ago. In some of the newer weapon systems that you mentioned rigid reliability for long-time periods are specified. The requirements may say for instance that the system shall operate for 10,000 hours or something like that. Simultaneously, the delivery data may be less than a year from the time that the first system can be assembled for operational test.

There doesn't appear to be any possibility of running a test for 10,000 hours — this is the point I'm getting at. Do you have some remarks about the way to cope with this difficult situation?

COUTINHO: Yes sir, I do. I'm not sure how friendly this particular audience is to the AGREE (Advisory Group on the Reliability of Electronic Equipment of DOD) Report, but the type of testing that you are referring to is based on the philosophy established by the AGREE Report. As far as I am concerned this philosophy simply does not work on new design because of the scheduling problem that you mentioned. We don't have the time within the schedules established for development; it is simply impossible to demonstrate reliability in accordance with the sequential tests required by the AGREE Report.

This is one of the reasons why I was able to sell my concept of test to failure. We propose to test to failure under combined environments. This is a difficult thing to do, but I do believe that engineers do get paid for the use of judgment, and we have to combine our environments in such a manner that it makes sense and brings out the weaknesses in the item. A device should be aged for one life-time under maximum expected operating environments, and then the environments should be increased until the device fails. This test should be repeated on several specimens, and they should fail in the same failure mode under the same environmental levels. The environmental spectrum should be dynamic and include peak environments.

If one life-time is too long for practical purposes, a shorter time can be used if it is shown that the degradation with time is not enough to cause failures later in the life of the part. The major point is that there is some basic strength still left after we have gone through the operational cycle. This is our margin of safety.

The reliability number is computed in a very similar manner to the way Prof. Lipson described it this morning. We have to establish the point from which we measure the margin of safety. We have to determine what will be the maximum expected operating loads, and then add a factor just to be sure. From this point on, we measure the margin of safety above this "limit" load.

This procedure should give us good reliability assurance. This program can help us to test a component early in its development stage. If we can identify that there is some particular unknowns associated with the design of this device, you may not even have to complete the entire device in order to test. We test wing panels, sheet string combinations, and many fittings and other items on an aircraft long before we even attempt to assemble the airplane.

So we're trying to test these elements about which there is some doubt in a statistical manner early in the program, and this is all tied in with the schedule. Now I think this is a practical approach. The problem here is that we are not using time as the variable. The normal concept of reliability testing is that you take a certain environment and then you operate the device within that environment until it fails.

This is not our problem in the aerospace industry. We do not have much of a problem with wearout. A much more important problem is our inability to predict the operational environments on new systems. When we find that our operational environment exceeds the predicted design environment, we then have to know if the part is still strong enough to take the more severe environment. Conventional AGREE type testing does not give us a feel for this type of difficulty— whereas testing to failure will do so.

Another basic problem here is that testing time is costly. If you are required to run test for hundreds of hours or 10,000 hours, you simply can't stand the cost of test time, much less fit such tests into your development schedule. So we're using the other approach of increasing the environment, testing to failure, and we feel we get very much better data out of this. Whenever you test to failure and you determine a cause of failure, you've got some engineering information. There is something you can do to the product. You know what to fix, and you have this information early in the program while you can still make changes. This is why I said before qualification, we have to run these tests. This is overstress testing.

KECECIOGLU (U. of Arizona): John, why do you like to call it reliability testing rather than testing for integrity or shake down test so to speak because you don't truly get a reliability value as such, but obviously the modes of failure imply what reliability improvements can be gotten by announced failure and so on, right?

COUTINHO: Doctor, I think you have answered the question when you said this was semantics. My definition of reliability refers to a product that works to the customer's satisfaction when it is in his hands, and this is what I am after. All these other things are tools.

BAZOVSKY (Raytheon): As an aircraft manufacturer you have been exposed recently to the Air Force's new policy on awarding so called reliability incentive contracts where the prime contractor is being rewarded in terms of additional profit if he meets reliability requirements or exceeds them, and he is being penalized if he does not meet them.

What are your comments and how would you go about handling such programs where you really have to prove reliability in operation before you get any profit or before they tell you you have lost your profits?

COUTINHO: I think that eventually we're going to solve this problem of incentive contracting for reliability. I don't think that anybody today has the answer to that question. We have an experimental incentive contract on the A2F with Bu Weps. I guess that this contract was awarded to us because the Department of Defense is anxious to find a way of rewarding a reliable contractor.

I spoke of how to motivate a single source supplier. This is part of the problem, and Grumman was the first company to sign an incentive contract. Both the Navy and Grumman were very much aware that this was highly experimental when they signed it. We still have not run the demonstration test. Part of the problem is that reliability demonstration, being statistically, is going to cost a great deal of money. We are committed to running these tests which are scheduled for approximately 18 months from now. By that time the airplane will have passed its Board of Inspection and Survey (BIS) Trials. There probably will be some units already in the fleet. This is the classical philosophy of testing reliability after qualification.

The Navy is going to have these airplanes by the time we get around to these reliability tests. They will know what the reliability of the airplane is by that time. With money for hardware as short as it is, I wonder if they are going to finance Grumman to go through with a formal reliability test, after they know whether the aircraft is satisfactory or not, purely for the purpose of administering a contractual incentive-penalty clause.

I can only say that we're going to have to find a solution to the problem of incentive contracts, but at this moment, we don't have the answer and I don't know of anybody else who has an answer at this time; and I know that other companies are struggling with this problem.

RELIABILITY—A KEY TO PROGRESS

REAR ADMIRAL R. B. FULTON, USN

Ladies and gentlemen, it is a pleasure to have this opportunity to discuss with you one of the Navy's most critical recent problems: unreliability that undermines our technological progress.

For about four years now, as Chief of the Bureau of Ships, I have felt as though I were standing neck-deep in a flood of reliability problems, at times barely able to keep from drowning in them. Now that we are getting this flood under control, I want to tell you something about it— its significance to the Navy, its causes, what we are doing about it, and future prospects.

Consider the Bureau's job. We are responsible for designing, building, maintaining, repairing and modernizing the Navy's ships and most of the equipment in them. If our radars and sonars don't work, hostile craft can slip past. If our propulsion turbines or aircraft catapults fail, we may never get close enough to to deal with an enemy even if we detect him. We must have ships and gear we can depend on.

This is nothing new. Reliability in Navy equipment has been an obvious essential for a long time. And this means gear that we can depend on in the tropics or polar seas, equipment that will function on a ship tossed in stormy seas, that will withstand the shock of explosions. It means machinery and equipment that will operate reliably with little or no maintenance for extended cruises far from shore bases.

The Navy has striven conscientiously to achieve all this and until recently our efforts were largely rewarded with success.

Naval propulsion turbines and gears are prime examples of this success. For years our turbine and gear specifications have begun with a statement that reliability is a paramount consideration. The resulting hardware reflects this concern: it gives us trouble-free service year after year. Recently we upped our service requirements from about 20 years to 30 years, yet little or no redesign was needed since existing designs had an ample margin. This means, of course, that for a long time we have been carrying around 10 years of reliability we didn't know we had.

Submarine electric propulsion has been legend in the Navy for its reliability— so much so that submarine type commanders occasionally succumb to temptation and skip a major overhaul so they can use the allotment for something they consider more pressing. With few exceptions, the propulsion motors in our current conventionally-powered submarines have provided 20 years of reliable service in spite of minimum maintenance.

Another star performer has been the turbo-electric propulsion systems installed in over 200 World War II escort vessels. Although these units have received very little maintenance and usually have been operated by inexperienced personnel, they have given us virtually no troubles at all.

Note: These items that are reliable for 20 years and more are deep in the bowels of our ships. The price of unreliability here could be disastrous, but reliability has enabled us to avoid that cost.

Familiarity with reliability such as is afforded by Naval turbines and submarine and escort vessel propulsion systems ill prepared me for my experience during these past four years, when hardly a day passed without a report from some quarter of the world that a vital Naval equipment had failed.

What had happened to the Navy's reliability tradition? In part, it had succumbed to the headlong pace of technological developments in the late 1940's and the 1950's. Gone were the pre-World War II days of small scale research, relatively few technical developments in a given decade, with time for consecutive development, testing, debugging, production and assimilation.

Scientists cracked the lid on a Pandora's box of new knowledge during World War II. After the war the new knowledge erupted out of the box to engulf us all. In a few short years we made tremendous strides, obtaining nuclear power, guided missiles and unprecedented progress in electronics. These weren't the only areas, although they are the best known. There were significant developments in almost all of our technical concepts, ranging from the whale shaped submarine hull form for high speed underwater operations all the way down to the use of synthetic fibers for Navy flags.

The new developments brought with them serious reliability problems. Nuclear reactor cores obviously must be built to operate without shipboard maintenance and, of course, have been. Since nuclear powered submarines can remain submerged for weeks on end, the equipment which is inaccessible when they are submerged must be able to function without maintenance for at least that long.

Guided missiles and their associated gear has brought extraordinary complexity and delicacy. We have had to devise means for handling these weapons systems safely and without disrupting their sensitive innards. We have had to devote whole new areas on many of our ships to missile checkout. Unfortunately, we have also had to spend untold hours finding out why systems aren't working right, or at all.

Rapid developments in electronics have also brought extreme complexity, along with many problems that no one could have foreseen back before World War II. A 1942 radar, for instance, was a relatively simple instrument that scanned with a rotating bedspring antenna and provided a rough approximation of range and bearing. By contrast, one of our more recent radars scans electronically and provides accurate information not only on range and bearing but also on height. The new radar corrects antenna errors electronically rather than by stabilizing the antenna base. The new radar furnishes information to missile systems and tracking radars, using a computer to do the job. The standard World War II radar had about 150 tubes; the new radar I have been speaking of has 617 tubes and 1,458 semi-conductors.

Electronic developments brought problems other than complexity. We now have so many different powerful systems aboard warships — radars, communications gear, sonar, electronics countermeasures and others — that it is extremely difficult to build them so that they will function within the confines of the ship without interfering with one another. As another example, some of our systems are so powerful that the radio frequency energy creates possible personnel hazards.

The problems accompanying new technological developments have been accentuated by the speed with which innovations have been introduced to the fleet. Under the pressure of the Cold War, we telescoped research, development, testing and production to bring the progress to the fleet as fast as possible. Nautilus (SSN 571), the world's first nuclear powered ship, went to sea in 1955. Now, only about eight years later, 31 nuclear powered submarines and surface ships have been completed and 48 more have been authorized. Our first guided missile surface ship, Boston (CAG 1), joined the fleet in 1955 too. Now 37 guided missile surface ships have been completed and 35 more are authorized.

To get a notion of the rate of change in electronics, compare the installations on a destroyer of the 1930's, on a World War II destroyer, and on one of our guided missile destroyers now entering the fleet. About all the pre-war ships had was a direction finder, some hydrophones, and radio equipment that could be comfortably fitted in an 80 square foot compartment. The total cost for the gear was less than

$50,000. Maintenance was a minor matter consisting of little more than battery charging, tube changing and an occasional receiver repair job.

By the end of World War II the destroyer installation was more sophisticated and cost roughly $150,000, not including fire control gear. Then we really started piling it on. Now, in 1963, our guided missile destroyers have surface search and air search radars, remote indicators, guidance and height finding radars, active as well as passive sonar, electronic countermeasures, the latest communications and other electronic gear. The total installation, less fire control electronics, costs about $3,300,000. In 15 years the electronics costs rose about 2200%.

When you consider the extent and speed of technological change since 1945, the fact that we encountered severe reliability problems is not surprising. The wonder is that we had as many successes as were achieved. The production of the world's first nuclear power plants in a matter of a few years is an example of one outstanding success. These plants have been operating with superb reliability for some time now.

Inevitably, the rapid pace of change extracted its toll. We overlooked some of the problems and implications of these changes and this led to failures. We neglected conventional equipment that was associated with our exotic new systems, only to have the prosaic items fail. There was a strong tendency for us to take our best people and put them on the new area, which sometimes reduced the effectiveness of work on conventional areas so that troubles cropped up there, too.

We could handle the problems adequately during the early 1950's, when only a few ships in the fleet had the new developments aboard, but by the end of the 1950's the flood was upon us. Maintenance costs soared. One radar costing about $400,000 initially required some $240,000 for maintenance each year. For a while we had to put over $70,000 annually into maintenance of some $6,000 communications gear.

We encountered problems in guided missile systems, not only during installation and checkout but also in the areas of operability, maintenance and repair of the in-service systems. We encountered these problems on such a broad scale that the Secretary of the Navy established a special Navy task force to expedite their solution.

Occasionally failure rates were astronomical: during the first year of operations of one new sonar, for example, it had a 20% failure rate. You can imagine how the fighting forces felt, having submarine detection equipment that had a one in five chance of being inoperable when needed.

Sometimes our troubles are caused by requirements catching up with ignorance. Thus, for years pipefitters had assumed they were making good sil-braze joints. The assumption seemed fair — there were virtually no failures. Then we built deeper diving submarines — and got some serious failures. We had no alternative but to go through the ships and check out every single vital sil-brazed joint and re-do many of them. The basic trouble was the pipefitters had inadequate means for judging the quality of their work. We have since developed non-destructive tests which permit inspectors to evaluate each pipefitter's work. Incidentally, we now have each sil-brazer leave a unique mark on each of his joints — we find that the quality goes way up when a man identifies with his work.

Faced with such serious failures, and on a broad scale, we developed an overall attack on every single facet of the problem that we could discover. Captain Kauffman has already told you about our use of work study techniques at the design stage and how this will improve reliability. We have also come up with reliability criteria which we are including in our specifications for electronics. We expect to extend this technique to non-electronics in the near future.

We are specifying maintainability features, so as to reduce the time spent in locating troubles and in curing them. We are improving our training of personnel who install these new equipments so we don't multiply our own troubles at this stage of the game. We are standardizing as much as we can so as to reduce the variety of problems we have to deal with and of training needed for installation and maintenance.

We started a program for improving calibration of test equipment and measuring devices. Investigation revealed that test devices used in check-out were often erroneous and therefore made a positive contribution to unreliability. By taking steps to assure that the calibration equipment is accurate, we can avoid a passel of problems.

We have a major quality-reliability assurance program aimed at making sure we know, by evaluation, review, measurement and test, the actual reliability and quality of ships and equipments. This will permit us to pinpoint our efforts to improve.

Commander Heenan has discussed our participation in a program for the improvement of shipboard maintenance. As he indicated, the program not only will improve the reliability of in-service equipment but also, by affording better feedback information, will lead to better design for reliability.

These are by no means all of the elements of our overall reliability program. We are convinced that the only way to get reliability in these days, while at the same time achieving technological progress, is *to emphasize reliability at every single stage, from initial concept through design, procurement, production, testing, installation and maintenance.*

At the same time that we are hitting in each of these areas, we are trying to be smarter about the implications of our new developments. What we must do in the future is to foresee in as much detail as possible the conditions our new developments will encounter.

There are other things we are doing to improve reliability. One of the things we are emphasizing is simplicity of design. There has been a strong tendency for an engineer who is faced with a new requirement to reach for the handiest components available and join them with whatever new elements are needed to do the job. All too often the result is unnecessary complexity. We are trying to get our engineers and designers to slow down long enough to see if there isn't a simpler, more direct approach to the problem. We want another lever or wheel or transistor, not more Rube Goldbergisms.

Let me bring this down to practical reality. Here is an example of what this kind of thinking can do. Recently, we were developing a massive new radar. The normal procedure would be to enclose each component in a bulky cabinet, air condition it, and worry later about how to get at the innards. A new approach was needed The thought of trying to maintain all these big black boxes was a nightmare.

Here is the answer that we came up with. We stripped the components of cabinets, blowers, heat exchangers and similar items and placed them in one air conditioned compartment. We set up 1 foot thick deck-to-overhead racks, like book shelves in library stacks, and put all the component elements out on them. This approach greatly simplifies maintenance and at the same time saves a lot of money, weight and space.

All too often such beautifully simple solutions appear only after we have tried half a dozen so-called solutions which create more problems than they solve. The simple solution seems obvious only in retrospect. So when looking for solutions, we must not be content with the first one that comes to mind. We need to view all complex solutions with a jaundiced eye and to keep looking for the new approach that will give us the simple answer.

This emphasis on simplicity, our work study efforts, our value assurance program and all the other programs I have mentioned, plus some not mentioned, comprise our reliability program. It is still in its infancy. However, we already have a return on some of the initial elements begun some years ago.

One of our best examples of this is the reliability of the Naval Tactical Data System — NTDS in alphabet language. This is one of the most complex systems ever installed on a Navy ship. A guided missile frigate's NTDS has about 44 distinct, major elements, including two computers. Just one of them has 3810 printed-circuit cards of 49 types containing 11,000 transistors, 34,000 diodes and about 1,000,000 magnetic cores.

NTDS needs all this for its formidable task. It takes information from sensors aboard its ships or other ships in a task force; keeps track of attacking forces — ships, aircraft, missiles, and submarines; analyzes information in relation to available defense weapons; grinds out recommendations and orders for appropriate defense measures and directs missiles and aircraft to targets. It can do all this virtually instantaneously.

NTDS was designed with extreme care, however. It is no more complex than it has to be to do its job. It uses selected elements of redundancy — such as multiple-computers that can be operated together or singly and a paper tape system to supplement the magnetic tape system. Each component was designed and built with reliability as a primary factor. Relationships with related systems such as radar, along with other implications, were carefully explored.

As a result of this reliability care, the maintenance cost for NTDS has been less than 1/2 or 1% of the equipment cost. During an extensive test period the mean time between failure for the computers — the key elements — was no less than 1500 hours. This mean time is increasing as we obtain additional operating time on the system. So we know we can make even our most complicated systems reliable if we approach the task correctly.

We have good reason, therefore, to believe that our program will greatly reduce our problems in the near future, permitting us to obtain maximum value from our technological progress.

However, it is clear that the reliability program will not work itself out of a job. The political-military pressures continue and we must continue to push back the boundaries of technological knowledge. We are doing so in numerous fields, including such esoteric areas as satellite communications, microelectronics, lasers, thermionic convertors, hydrofoil ships and hydroskimmers. Our work in these fields is bound to create a host of new reliability problems.

Take our work in developing hydrofoil ships and hydroskimmers. As you probably know, hydrofoils are underwater wings on struts that extend down from a craft. When a craft gets going fast enough, the flow of the water past the foils creates lift that raises the hull out of the water, almost free of its speed-killing drag.

We have already launched the 110 ton anti-submarine hydrofoil craft High Point (PCH 1) and are well underway in designing a 300 ton experimental hydrofoil ship. We are pushing the limits of our technological capabilities with these ships. You can imagine the tremendous stresses placed on the struts and wings when a 300 ton ship goes into a tight turn at 60 knots. Developing materials, machinery and equipment that is light enough for hydrofoils is a major problem too. The same problem plagues us in developing hydroskimmers — craft that ride on an air cushion. We now have a contractor working on a 20 ton hydroskimmer that is expected to operate completely clear of the water at speeds of over 70 knots. Four fans will hold the craft aloft. Of course, they must be reliable or they will let us down, literally.

I can assure you that we are going to keep on pushing such technological developments as fast as is feasible. Concern for reliability isn't going to slow us down at all. On the contrary, we consider our reliability program to be a tool that will permit us to get new developments into practical operation more rapidly than is now possible. It will do this by largely eliminating horrendous periods of adjustment such as we have been dealing with the past four years.

Our reliability program may well provide another major benefit: sufficiently trouble free operations so we can reduce the manning of our ships. This might avoid many a war casualty and would permit us to devote spaces now needed for crew members to defense hardware.

We cannot predict with any assurance that our reliability program will permit us to reduce the size of our crews. But we do know we have a program that is helping us get control of the flood of failures that's been plaguing me. We know we have a program that will help us to continue to introduce progress to the fleet rapidly—but with assurance that the fleet can depend on the innovations.

We haven't licked this problem yet. Much remains to be done. Industry, academic institutions and the Navy must work together to find better solutions. That's where this seminar is making a real contribution to enhancing the nation's defense. Speaking both as the Chief of the Bureau of Ships and as a man who has been neck-deep in problems and would like to get his feet dry, I wish you great success.

RELIABILITY CONSIDERATIONS IN THE DESIGN OF MAIN PROPELLING MACHINERY

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Reliability is composed of many ingredients, all of which are independently important. These ingredients may be classified into the following major categories:

Experience
Machinery specifications
Design
Manufacture
Quality Control
Assembly and Test
Preservation, shipment and storage
Installation and alignment
Operation and maintenance

All machinery designers are generally aware of the major and more important factors affecting reliability. Some of the less important factors can greatly influence the reliability of the machinery or at least become annoying nuisances to the operating personnel. I would like to select items of this latter variety for discussion from the listed major categories.

EXPERIENCE

Experience is an extremely important consideration in the makeup of reliability. Field difficulties encountered, properly reported and recorded, result in the acquisition of knowledge and judgment. The machinery manufacturers employ a service department that "feed back" performance information to design, manufacturing and quality control departments permitting the evaluation of the machinery and the elimination of trouble areas on new designs. The Bureau of Ships has its own system for obtaining performance information which is on a much larger scale since it encompasses all the problems of the individual manufacturers.

Other sources of experience come from the research and development programs of the individual manufacturers. The Bureau of Ships awards "study contracts" to the machinery manufacturers and to educational institutions. In addition, the Navy Department has facilities such as the "David Taylor Model Basin" and the "Naval Boiler and Turbine Laboratory" where research and testing are conducted. All of these sources provide the experience and knowledge so necessary for reliability.

MACHINERY SPECIFICATIONS

The specifications describe the requirements of the machinery with respect to size, arrangement, materials, performance, and tests. They provide an excellent opportunity to utilize the experience and knowledge previously gained and thereby become an important vehicle in obtaining reliability.

The specifications should be well organized and explicit. Frequently, requirements for a particular machinery component will appear at several locations and possibly set forth conflicting requirements which can lead to omission or incorrect interpretation of the requirements.

In paragraph S45-1-J of the cleaning and preservation section of the "General Specifications for Ships of the United States Navy," the following appears — "General surfaces that will be in contact with lubricating oil shall be thoroughly cleaned. Scale and corrosion products shall be removed from ferrous surfaces. Except where specified otherwise herein, surface preparation and removal of rust and mill scale shall be accomplished in accordance with the requirements of Section S19-1, except that sand blasting is not permitted. . . . "

The specification clearly states that sand blasting is not permitted. Section S19-1-f permits cleaning by acid pickling, blasting or other mechanical means. We interpret this to permit shot blasting.

Our experience has shown that shot blasting leaves a microscopically "fuzzy" surface that is undesirable. A magnet passed over this surface will pick up tiny metallic slivers. There is also the possibility that some of the shot will become imbedded in the plate or casting surfaces only to later loosen due to the vibration of the operating machinery. This loosened shot circulating in a lube-oil system can raise havoc with controls, journals, bearings and gear teeth. We firmly believe that the cause of reliability is aided by prohibiting shot blasting on surfaces exposed to lube-oil systems and permitting sand blasting.

DESIGN

1. Lube Oil Systems

a. Purifier suction should preferably be located in a well where water may accumulate and easily be drawn off by the purifier. In many cases such a well may not be feasible. An alternate is to provide the sump tank with a sloping bottom. In the case of turbine generator sets where neither of these approaches is practicable, adequate clean out openings should be provided to permit complete access to the sump region.

b. Lube oil headers that are large and long should be vented to prevent the accumulation of air pockets that could conceivably get carried into the bearings during a heavy sea. This consideration is also important in the design of hydraulic controls.

c. A practice that we employ in the design of our reduction gear fabrications is to use straight sections of piping. This avoids blind spots and permits ease of inspection and cleaning.

d. The use of oil filters in the hydraulic control oil. These controls have many close clearance parts. Small amounts of contamination can greatly affect the sensitivity and operation of these controls.

2. Turbine to Gear Flexible Couplings

Traditionally turbine to gear flexible couplings have been double ended. In the last several years the reduction gear specifications permit the use of single-ended flexible couplings for some applications.

We believe that where the conditions permit, the single-ended flexible coupling is a more desirable coupling to use then the double-ended flexible coupling.

Here are some of the reasons that make the single-ended coupling desirable.

a. Stability — Single-ended couplings perform well at light loads. This is important when it is necessary or desirable to check overspeed devices with the turbines coupled to the reduction gear.

   During the overspeed test of one of our large reduction gears which was being conducted at no load, a severe vibration occurred in the turbine coupling. Subsequent examination showed the coupling teeth to be badly cratered even though they had been nitrided and lapped. The vibration had lasted only for the short time necessary to shut the machine down.

b. Balance — The single-ended coupling minimizes the mass that can rotate eccentrically due to pitch circle runout and thereby permits a finer degree of balance to be obtained where precision balance is of great importance.

c. The long length of coupling shaft usually associated with the single-ended coupling to accommodate large misalignments without appreciably reducing the number of teeth in contact or producing large displacements of the hub teeth with respect to the sleeve teeth in the course of one revolution. These are features that permit the application of this coupling to the normally double ended Quill Shaft Coupling.

   Although the long length of the coupling may be considered a disadvantage, it can be offset in many cases by locating the high speed elements of the reduction gear aft of the slow speed elements.

d. Reduction in the number of wearing parts.

e. Excellent service record — We have used the single-ended coupling in a number of commercial applications with outstanding success. One of the vessels on which these couplings were installed has been in continuous Trans-Atlantic service for more than ten years. At a recent inspection, these originally installed couplings were found to be in excellent condition.

3. Positioning of the High Speed Elements

In a reduction gear where the speed is reduced in two stages, some means must be provided to position the high speed or first reduction elements. A very simple and effective technique to locate the high speed elements in their housings is to use quill shaft couplings that have limited end travel. See Fig. 1. In the case of an articulated reduction gear there is only one intermediate gearing assembly for each turbine input, whereas, in the locked train reduction gear there are two. In this latter case where two intermediate gearing assemblies exist, only one of them has the limited end travel couplings. The high speed elements are then positioned as follows:

a. The slow speed gear is positioned by the main thrust bearing.
b. One of the high speed gears is positioned from the slow speed gear by the restricted end travel couplings.
c. This high speed gear positions the high speed pinion, which in turn positions the other high speed gear. This method combines simplicity with the utmost in reliability by eliminating auxiliary devices, such as, high speed gear thrust bearings which have been a source of difficulty in the past.

4. The "Journal Rise" Method of Timing

One of the complexities associated with the "locked train" gear is the need to "time" the gear. Timing is the technique and procedure used to insure that the load divides equally at the high speed pinion between the two engaging high speed gears in order to prevent one branch of the locked train from becoming overloaded when transmitting rated torque.

On highly loaded propulsion reduction gears it is extremely important that the gears be accurately timed to avoid tooth distress or possibly tooth breakage.

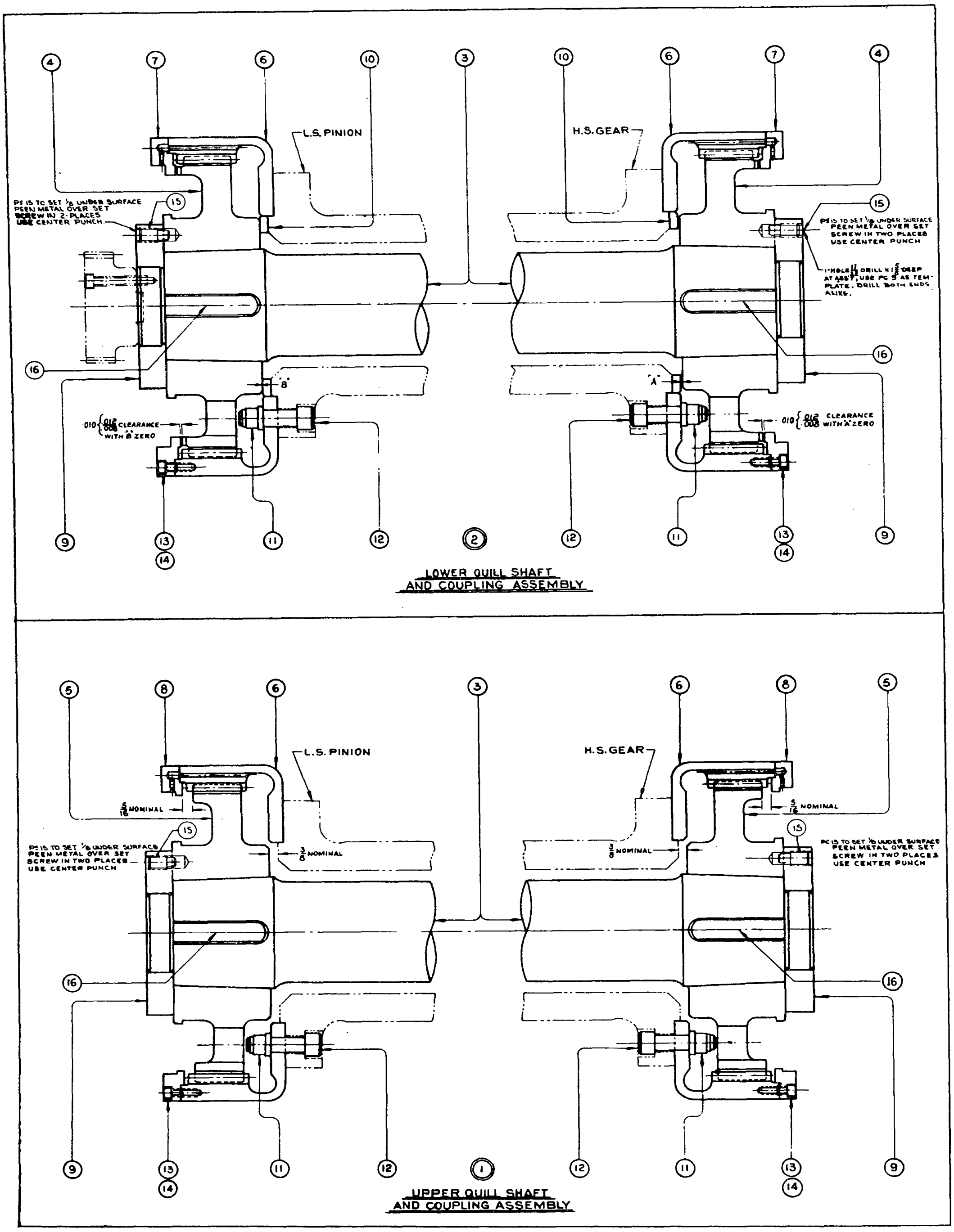


Figure 1. Limited End Travel Couplings

One method of accurately timing gears is the "Journal Rise" method which is quite simple, reliable and repeatable. This method is based on the fact that the high speed pinion journal is unloaded and operates within the bearing clearance when the unit is timed.

The problem, then, is to position the journal of the high speed pinion in the bearing clearance while the pinion is transmitting torque. This can be accomplished by employing the following procedure:

a. Assemble the high speed pinion into the reduction gear omitting the upper halves of the high speed pinion bearings.
b. Install the bearing cap onto which has been machined a reference plane that is parallel to the bearing joint.

c With the high speed pinion resting freely at the bottom of the lower bearing halves, measure the distance between the top of the journal and the reference plane on the bearing cap. See Fig. 2-1.

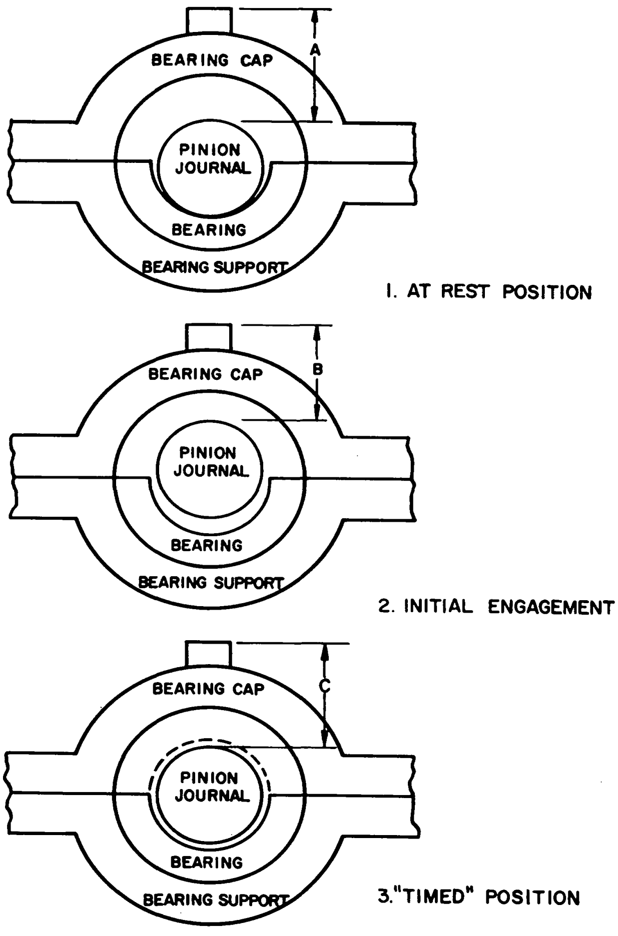
d An engagement of the gear elements is then made so that the pinion rises up off of its bearing when torque is applied. The applied torque should be pure and not contain a bending moment since this could affect the position of the pinion. The torque usually selected ranges between 10 and 20% of full power torque depending upon the weight of the gear elements. This amount of torque will properly position the gear elements and bring the teeth into hard contact.

Another measurement between the top of the pinion journal and the reference plane on the bearing cap is made. See Fig. 2-2.

e By means of a timing table that will predict a journal movement, various meshes are altered, which in effect causes one of the high speed gears to rotate, thereby lowering the pinion journals until they are ultimately within the bearing clearance thus timing the unit. See Fig. 2-3 and Fig. 3.

f The minimum possible pinion journal movement is given by the formula:

$$\text{Minimum journal movement} = \frac{\pi DG}{2L}$$



TIMING

JOURNAL RISE METHOD

Figure 2.

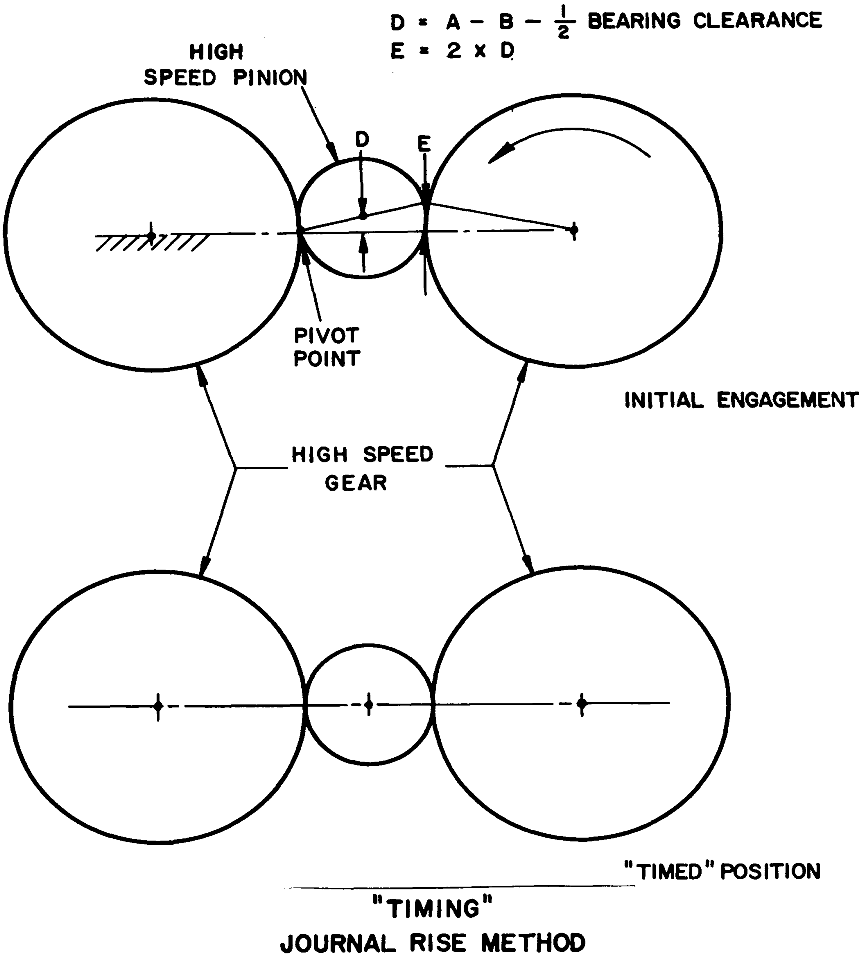


Figure 3.

where DG = pitch diameter of the high speed gear
L = product of the factors of the meshes utilized less certain common factors.

By using the teeth in the high speed gear, slow speed pinion and the quill shaft flexible coupling, minimum journal movements in the order of .001 can usually be obtained.

MANUFACTURE AND QUALITY CONTROL

It is the Manufacturing Department's function to produce equipment that will meet the requirements established by the Engineering Department. Many times these requirements present difficult challenges. One of these challenges has been the production of steam tight joints in the turbine casing, particularly, the horizontal joint. Past practice has been to "hand fit" these joints by scraping. In many cases these joints would develop leaks, and of course, once a leak occurred it became necessary to lift the casing and refit the joint. "Pumping grooves" were placed in the joint so that if a leak developed, a sealant could be pumped into the groove in an effort to stop or at least forestall the leak until such time as the casing could be conveniently lifted.

For sometime now, we have been producing these joints on a micro-mill planer which will machine a joint surface that is accurately flat and obtain a finish of better than 63 micro-inch in a single pass. These accurately machined joints, which are superior to the best handfitting operation, have virtually eliminated the leaking joint problem. Sufficient confidence has been established in the tightness of these joints that pumping grooves are no longer considered necessary and are being eliminated.

Quality Control measures are essential to assure that the design has been properly executed. Deviated parts are carefully reviewed for possible acceptance, correction or rejection. But Quality Control's function extends far beyond individual parts. In the assembly of a complex piece of machinery we have found it absolutely necessary to provide a detailed "manufacturing assembly and quality assurance procedure." Its purpose is to guide and instruct the Manufacturing Department in assembling, erecting and testing, to describe and schedule quality assurance features through inspection functions and to provide field engineers with factory techniques and data that can be duplicated at installation. This procedure lists every major operation, check, and measurement that is necessary in the construction of the particular machine. After each vital operation is completed, a form is signed attesting to this fact by the job foreman and the Quality Control inspector.

These requirements are placed on 8-1/2 x 11 sheets and bound in booklet form. One of these booklets, which contains all of the requirements and instructions for erection, is assigned to an individual machine. Completeness of inspection and performance of all operations are confirmed by signature and date. This technique establishes responsibility and assures that no operation or check is omitted. Communication breakdowns due to a multiple shift operation are avoided. This procedure assures the efficient production of a high quality, reliable product.

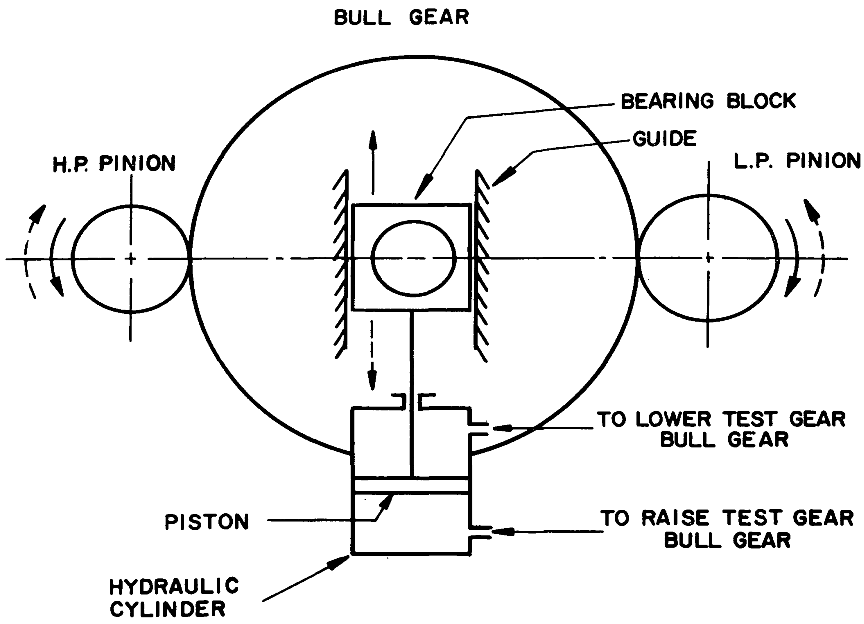
ASSEMBLY AND TEST

One of the best indications of reliability of a new design is the successful completion of shop testing under load. In the case of the reduction gear, the specifications require that "the first propulsion gear furnished under each contract or order shall be assembled in the gear manufacturer's shop and operated continuously for at least 24 hours at full speed and full load torque to check noise and tooth contact." In the case of high horsepower or highly loaded gears, these general specifications are usually modified to include some testing at 115% of full load torque.

Since shop testing is conducted under ideal conditions compared to actual operation in the ship where vibrations, misalignments and heavy sea conditions exist, overload testing appears to be well founded. It is our belief that all large, high horsepower and highly loaded gears should be individually tested at 115% of full load torque for some minimum time interval in addition to or combined with testing at 100% of full power torque.

The conventional method used in testing high horsepower gears is to connect two of them in a "back to back" arrangement, and lock the torque into the reduction gears by torquing one side in the ahead direction and the other side in the astern direction. A drive turbine or motor is used to rotate the machinery. The power required for such testing is simply the frictional and windage losses of the system. No large drive turbine or load absorber is needed. The method is cumbersome however, due to the fact that two reduction gears are required to be assembled and operated simultaneously, due to the difficulty in applying and changing the applied load and in starting the machinery rotating since the load is applied with the unit stopped.

We have developed a very unique device for load testing reduction gears. This device or test gear consists of two pinions and one bull gear. See Fig. 4. The bull gear is supported by bearings that are mounted in movable bearing blocks that can slide vertically in guides. These bearing blocks are connected to hydraulically operated pistons. When the bull gear is moved upward above its central position, the two meshing pinions are rotated in opposite directions and apply torque to the reduction gear under test. One side is torqued in its normal ahead direction while



TEST GEAR SCHEMATIC

Figure 4.

the other side is torqued in the astern direction. Moving the bull gear below the central position causes the direction of applied torques to reverse. Since the bull gear movement is directly proportional to the hydraulic pressure applied to the cylinders, the torque applied to the reduction gear is directly dependent on this same hydraulic pressure.

One end of the test gear is coupled to the reduction gear under test by two identical flexible couplings. The other end of the test gear is flexibly connected to a drive turbine. See Fig. 5.

In order to use the test gear to test a reduction gear, two basic requirements must be met. First, the speed relationship between the two pinions of the test gear must match *exactly* the speed relationship of the two input pinions of the reduction gear. Secondly, the center distance between the two pinions of the test gear must match the center distance of the two input pinions of the reduction gear.

This test gear was originally designed to test the prototype gear for the USS ENTERPRISE. It was also used to test all four of the gears for this aircraft carrier. Fig. 6 shows the test gear coupled to the reduction gear in the foreground. In the background a second gear is being erected. When the testing of the first gear was completed, the test gear, complete with its own foundation, drive turbine and hydraulic controls, was lifted as a unit, rotated 180° , and set down in position to test the second gear and so on until all four units were tested. Since that time, the test gear has been used to test the LPH reduction gear, the DDG reduction gear, and it is presently being used to test the destroyer escort reduction gears.

PRESERVATION, SHIPMENT AND STORAGE

We have long been a proponent of assembled shipment of reduction gears. All of the submarine reduction gears that we have manufactured have been shipped as assembled units. Although the USS NAUTILUS and SKATE class gears were small units, the reduction gears for the current submarines are considerably larger.

With the increased size and weight came an increased desire to continue assembly shipments. This was brought about by the fine teeth employed on these units and the ease with which they could be damaged. Countering this desire for

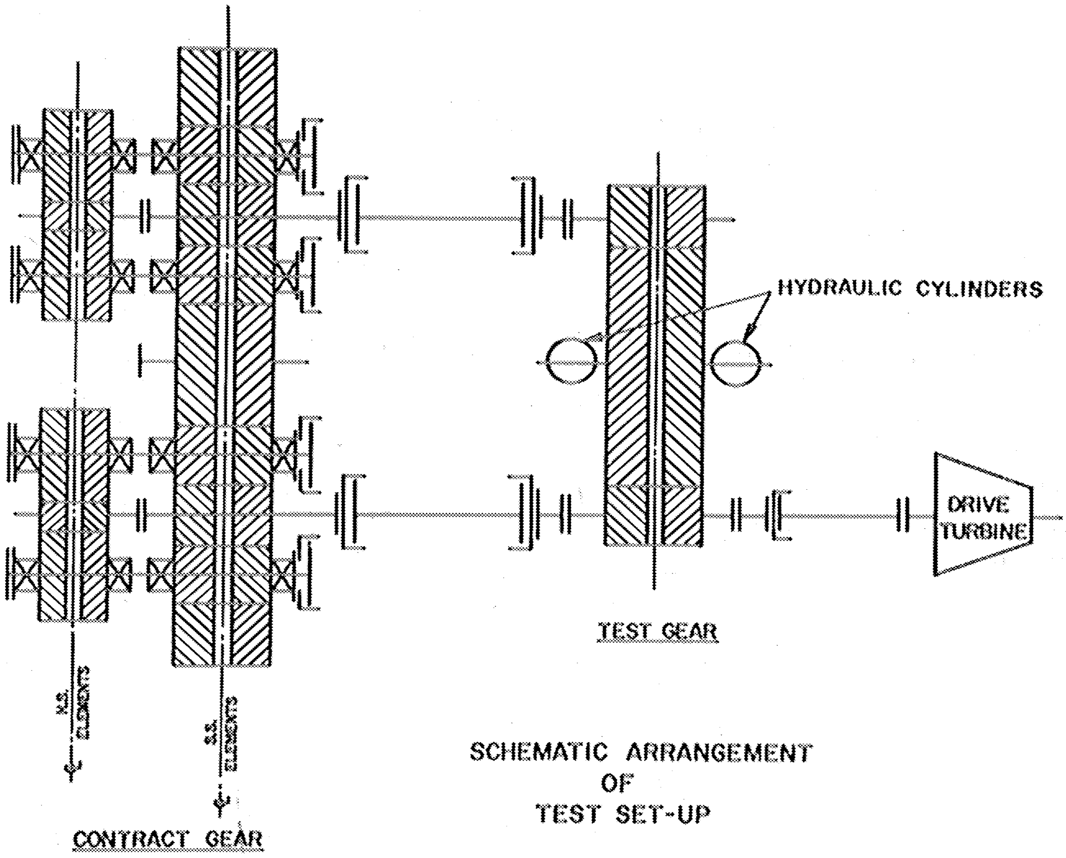


Figure 5.

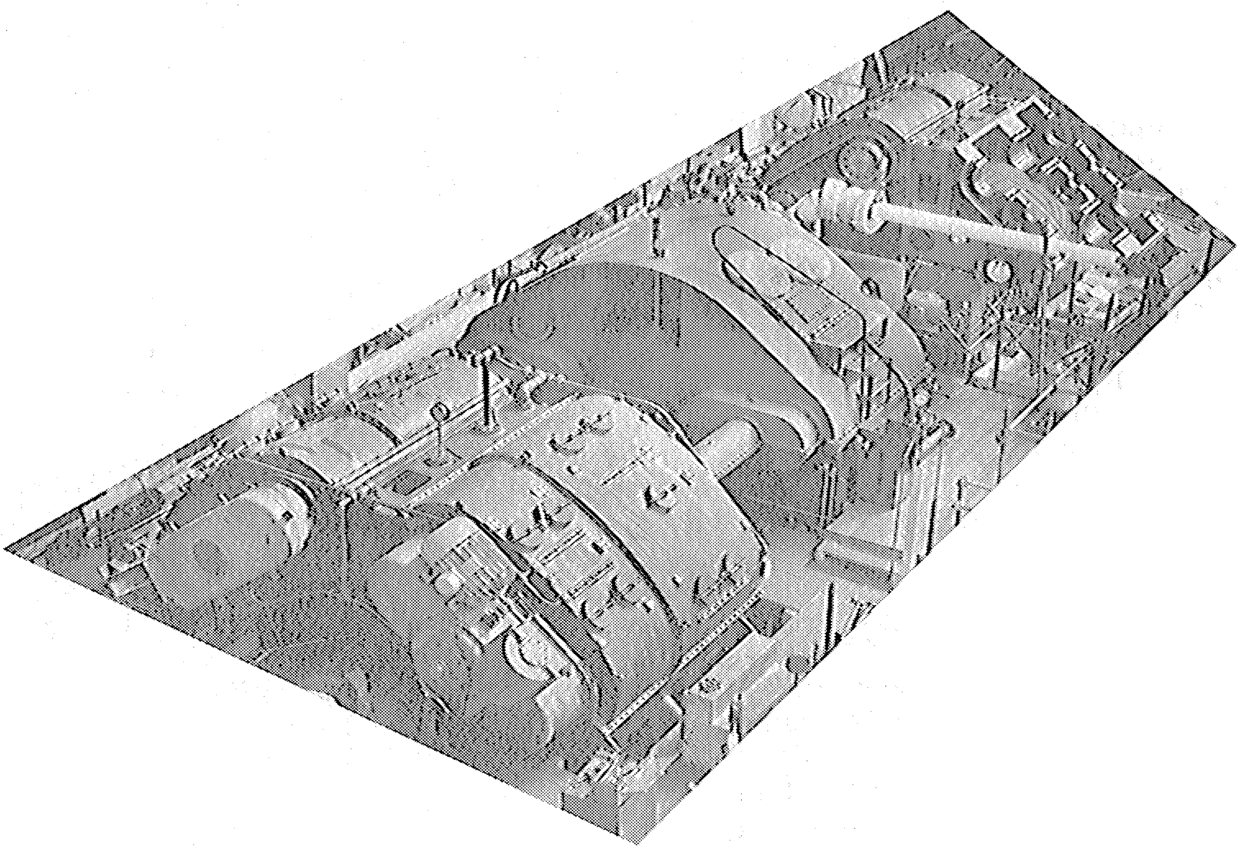


Figure 6. Test Gear - Shop Test.

assembled shipments was the fear that some of the noise reduction features would be lost or impaired on a long transcontinental shipment by rail.

To overcome this fear, a new shipping technique was developed. An investigation was conducted for some substance that could be inserted in between the teeth to prevent "brinelling" or fretting corrosion. Mylar tape was selected. Laboratory tests showed that brinelling or fretting would not occur between metallic materials if the mylar tape separated them. Mylar tapes of .003 and .005 thickness were selected for use in the first and second reduction meshes respectively.

It was found that the tapes would easily tear when attempting to roll them into mesh. It was determined that this could be avoided if the tapes were preformed to approximate the tooth shape. Fig. 7 shows these preformed mylar tapes, which are inserted into each mesh of the gear after the preservation has been applied. Counter torques are then applied to the two input pinions to lock the elements torsionally. A spring force is applied to the slow speed gear to lock all elements axially.

Fig. 8 shows an assembled reduction gear, prepared for shipment as described, loaded onto a deep well box car. This gear was shipped to the West Coast where it was installed in a submarine that is now operational. Vibration analysis conducted aboard ship indicated that no change to the originally established vibration levels had occurred.

Fig. 9 shows a DDG reduction gear being lifted in preparation for shipment. Subsequent units have been designed so that the elaborate lifting beam is not required. Assembled shipment of the gear requires a structure that is inherently rigid. This rigidity contributes considerably to obtaining the correct internal alignment of the gear at installation.

Another technique that we have recently adopted in the assembly shipment of reduction gears is the elimination of the complete "tear down" of the gear after shop testing. The procedure used after test is to:

a. Visually examine all teeth for distress.
b. Remove all bearings to permit inspection and apply preservative.
c. Remove all quill shafts and quill shaft coupling hubs for inspection of the couplings and the application of preservative.
d. Application of preservative to rotating parts and housings.
e. Recheck of timing after all rotating parts have been reassembled.
f. Installation of the mylar tapes and completion of assembly of reduction gear for shipment.

We believe that this technique, which is a refinement to the assembled shipment of the gear, provides the greatest assurance of eliminating damage to the gear teeth since these elements are never disengaged after shop test. This procedure should virtually eliminate the clicking noise of bruised teeth so frequently associated with shipboard startup of the gear.

We have developed an additional technique for the submarine reduction gear which we hope to apply in the near future that will permit the "turbine to gear" alignment and the "make-up" of the coupling flanges without lifting the pinion and gear covers on a unit where the high speed elements are located aft.

Fig. 10 shows a turbine generator set, prepared for assembly shipment complete with mounting structure. Assembly shipment of complex machinery greatly minimizes installation difficulties. This is particularly true of submarine machinery of which we are shipping all major propulsion machinery components as assembled units. These include the:

Propulsion turbine
Propulsion reduction gear
Turbine generator
Propulsion clutch and stub shaft
Main thrust bearing

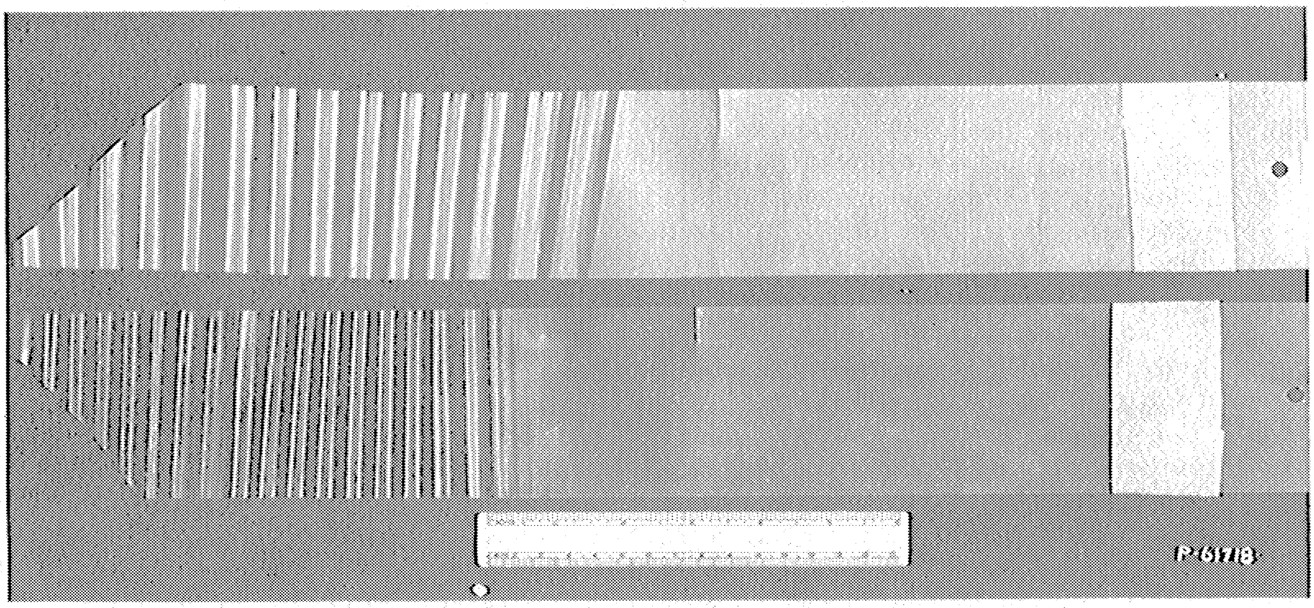


Figure 7. Mylar Tapes

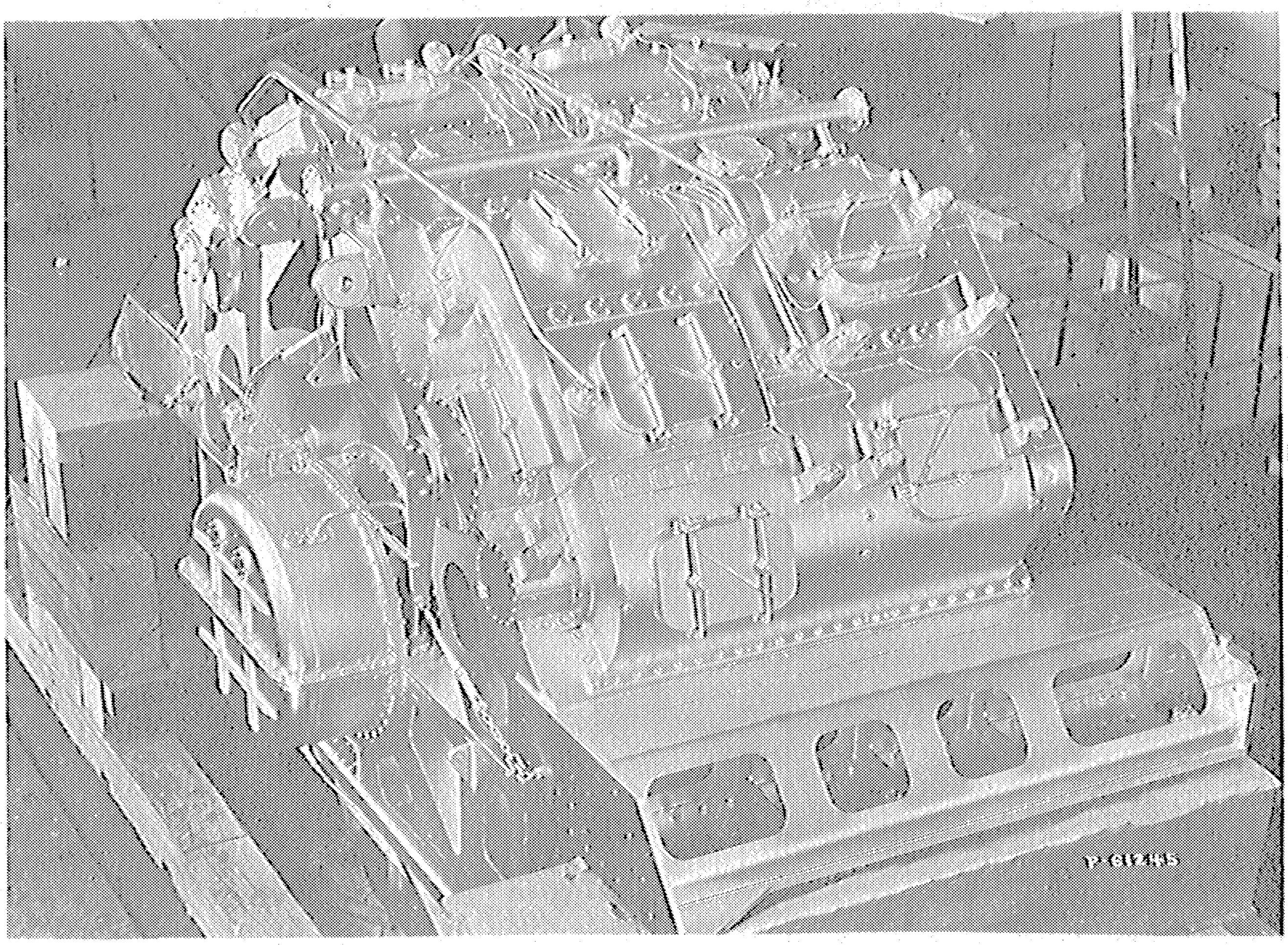


Figure 8. Assembled Shipment - Submarine Reduction Gear.

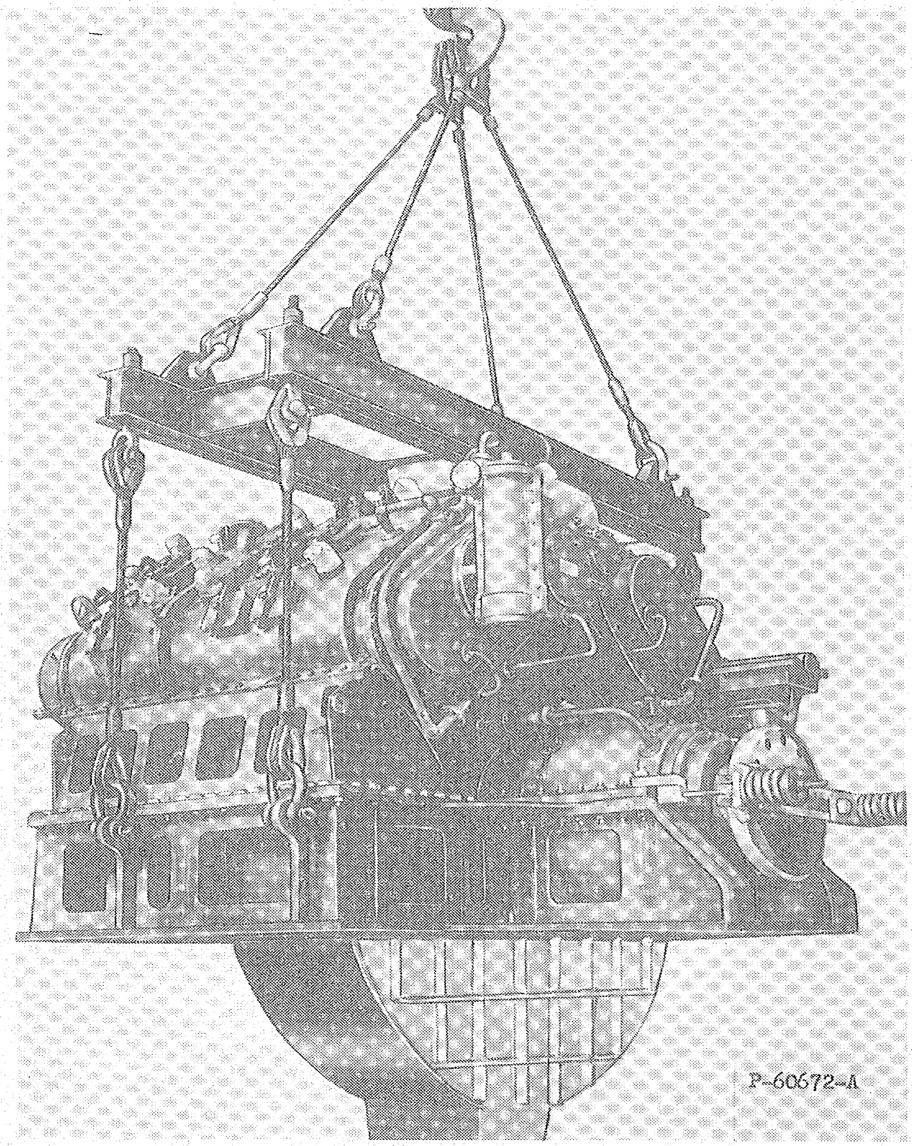


Figure 9. Assembled Lift - DDG Reduction Gear.

INSTALLATION AND ALIGNMENT

The installation of assembled machinery components reduces a difficult task to a simple one and since this is a goal largely accomplished, at least for the submarine machinery, I will not dwell on the subject of installation.

Alignment of the machinery must be given careful attention. Two areas of special concern are the reduction gear to line shaft alignment and the turbine to gear alignment. Carelessness or error in establishing these alignments may seriously affect the reliability of the machinery. If these alignments are established while the ship is on the ways or in a graving dock, it is essential that they be rechecked after becoming water borne. Improper gear alignment, most likely, will result in distressed teeth and in the extreme case tooth failure. Improper turbine alignment will result in a coupling that will wear rapidly or fail with operation.

One of the critical periods in the life of the reduction gear with respect to corrosion is the period after the preservation has been removed but prior to regular sea going operation. Moisture or water can easily be introduced into the system via flushing, displacement or final charge oils. If the sump tank is poorly designed, it may be impossible to remove this water using the purifier. When the lube

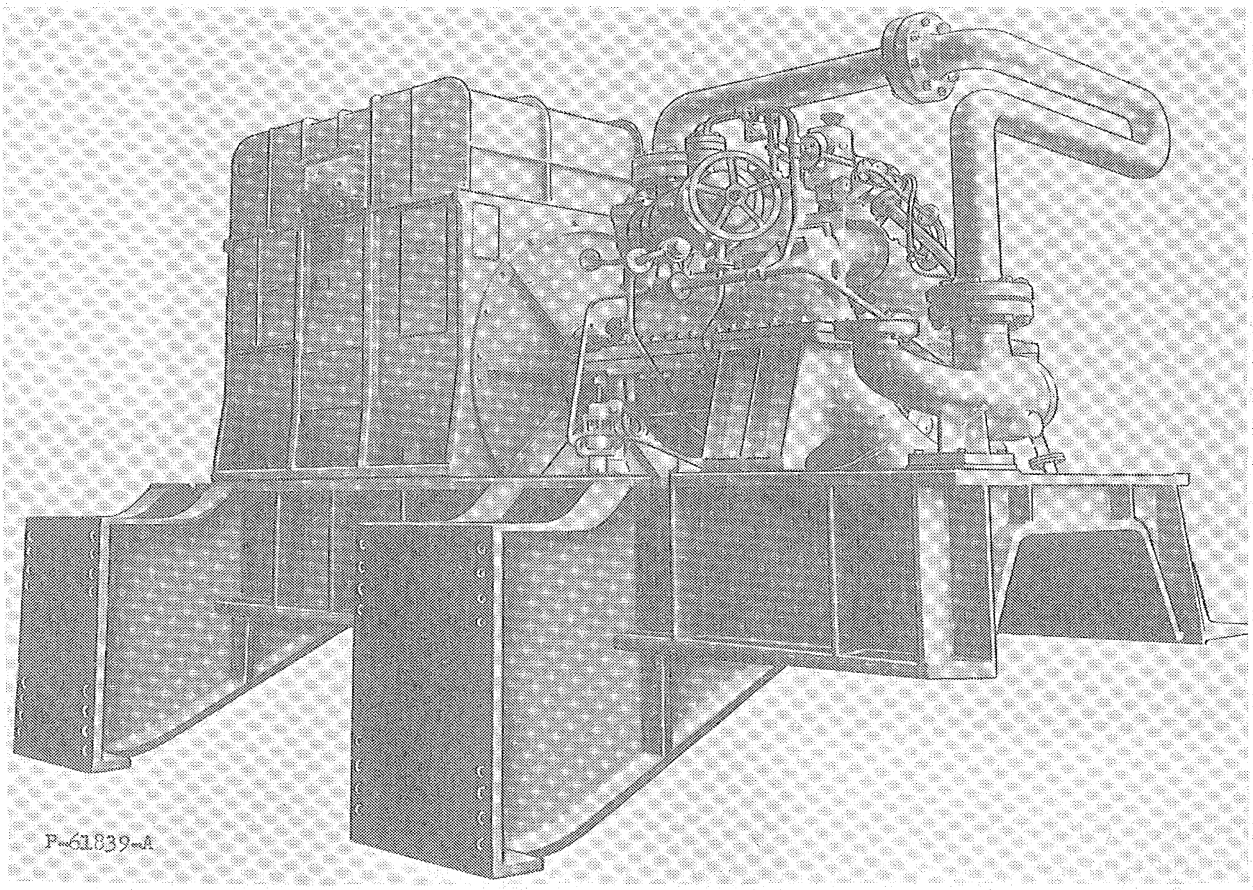


Figure 10. Assembled Shipment - Turbine Generator Set.

system is placed into operation, enough agitation exists to circulate and vaporize some of the water present. Subsequent shut down results in the water vapor condensing in the upper regions of the gear where it begins its corrosion attack.

If the sump tank is equipped with a well for the purifier suction or has a sloping bottom, operation of the purifier will remove the offending water from the system. However, if this is not the case, the probability is good that some corrosion will take place. The gear elements can be protected by periodically turning on the L.O. system and rotating the slow speed gear a minimum of 1-1/4 turns. This procedure does not offer any protection to the upper portions of the gear housings since the oil does not reach these areas.

All too frequently, even this simple expedient of protecting the gear elements is omitted. After some long period of idleness, someone remembers to look into the reduction gear and is appalled by what he sees. Although this corrosion is seldom disastrous, it may require the removal of the pinion and gear covers so that the corrosion products can be removed without contaminating the oil system.

To prevent this corrosion from taking place, "humidials" or similar visual humidity indicators could be installed at the top of the reduction gear. The presence of moisture in the interior of the reduction gear would be detected by these externally mounted visual devices and action could be taken before any serious amount of corrosion took place. In some cases, de-humidifiers are being temporarily installed during this critical period.

OPERATION AND MAINTENANCE

The instruction book plays an important role in the reliability of the machinery. It must adequately describe and illustrate the equipment, carefully explaining any unusual features. Operating instructions, limitations, inspection procedures, maintenance programs and dis-assembly techniques must be clearly set forth in a well organized manner to permit operating personnel to easily obtain this information.

The lube oil system, being the life-blood of the propulsion machinery, requires constant, careful attention in order for the equipment to perform satisfactorily and provide reliable service.

One of the practices set forth in the ASTM "Recommended Practices for the Purification of Marine Propulsion Turbine Lubrication Oil" is to operate the purifier continuously while the propulsion plant lube-oil system is in operation. During the start-up and shut-down of the plant, the turbine and gear is rotated for many hours with the gland seal system in operation. This is a time when moisture can be easily introduced into the system. If introduced at shut down, the moisture will remain in the system for some time and corrosion may take place. Circulation of corrosion products will accelerate wear and tend to sludge couplings. Since continuous operation of the purifier with the lube-oil system requires little additional effort, it would appear that this practice has much merit.

DISCUSSION

FIXMAN (Maritime): I have two questions for Mr. Grey. 1) Do you have any correlation between your ship test and either in-service tests or in-service failures? 2) Do you have any evidence to show that your intermediate quill couplings on your lock train gear actually float under heavy loads?

GREY: I'll answer your second question first. We've used this technique of positioning the high speed elements by means of these limited flexible couplings for some time. I'm sure this period extends back to 6, 7, 8, perhaps more years than that. To my knowledge there has been no difficulty with this type of coupling, with this method of positioning the high speed elements.

With respect to your question about measurements, we don't have any measurements, however, at full power torque, these couplings most likely would lock up, and...

FIXMAN: This is the reason for my question, your main propulsion shaft due to thrust variation, due to main thrust clearance, and due to the flexibility of the main thrust bearing, moves axially.

GREY: But you still have your turbine to gear flexible coupling.

FIXMAN: Yes, but you are also endloading your teeth in your herring bone gears.

GREY: You're endloading your teeth only by the friction that is necessary to slide the high speed flexible coupling and this is usually perhaps half or so of the frictional forces necessary to slide your quill shaft flexible coupling. Even though the quill shaft couplings lock up, you still have the flexibility of the turbine to gear coupling.

FIXMAN: You have never instrumented the coupling to find out whether it is moving or not.

GREY: No, not to my knowledge. Correlation between shop testing and difficulties encountered at sea — I made the recommendation of shop testing in connection with highly loaded reduction gears and large reduction gears. It would seem prudent to us to conduct such power testing because the margins are reduced in units of this type.

As an example the units for the Enterprise by specification were required to be each individually tested and in addition to that two of these units were tested for some time at 115% of full power torque.

KAUFFMAN (Naval Boiler & Turbine Lab): On page 5 you have a term "badly cratered flexible coupling teeth." Could you explain that term a little bit more? It is not familiar to me.

GREY: Well, I think cratering is a common term in describing coupling failures. You almost have to see one to really appreciate it. In some cases they are practically unbelievable, but it is just simply a crater that will develop in the tooth profile as a result of perhaps lack of lubrication or large sliding motion between a hub tooth and a sleeve tooth. I don't know whether you are aware of it or not, but the Bureau of Ships has recognized the coupling problem and has directed NBTL to conduct coupling tests. They have issued a report on at least the first of these couplings that was tested. The failure was of the cratering type.

KAUFFMAN: This work is being done in our division and we have been using a term called worm tracking.

GREY: I see. This is essentially the same.

BAZOVSKY (Raytheon): Do you establish a maintenance policy or maintenance schedule for these propulsion plants, or do you leave it up to the Navy to establish their own maintenance policies?

GREY: The instruction book that we issue contains certain recommendations with respect to the machinery. Off hand I don't recall what the specific time interval may be regarding schedules. However, the Bureau of Ships has instructions as to when certain equipment is to be inspected and they state the time element involved.

BAZOVSKY: Basically that means you don't make your own decisions about establishing a maintenance schedule.

GREY: I think this is right. We may make a recommendation in the instruction book as to when certain equipment is to be inspected, but the fulfillment of this is dependent upon the Navy.

BAZOVSKY: On what basis would these recommendations be made, this time limit?

GREY: Well, one of the inspections that you might want to conduct at a convenient time, after the course of one year of operation would be to lift all of the pinion and gear covers and visually examine all the teeth for signs of distress. Perhaps you would like to examine the mesh sprays with the lube-oil system on. Things of this nature can be checked periodically. I can't recall exactly, but I believe the Bureau of Ships' instructions have a time interval for the inspection of the flexible couplings. In many cases in order to disassemble them, it is necessary to remove the couplings completely and withdraw the sleeves from the hubs, and this is done on a routine basis or a recommended basis, but I think the Navy Department controls the inspection and maintenance programs more than the manufacturers. I'm speaking from the standpoint of reduction gears. This may not apply to some of the other equipment.

If a gear is well designed, manufactured, in other words if you fulfill the requirements that pertain to reliability, this gear should operate practically indefinitely with little or no maintenance.

BAZOVSKY: You mean indefinitely say for 20 years or . . .

GREY: In some cases this is so.

JACKSON (Bureau of Ships): May I comment on what he has said. The Bureau of Ships requires the ships to carry out the instructions as given in the manufacturer's instruction booklets, unless we obtain some modification to those instructions. This is part of the program that Captain Kauffman was telling you about yesterday in the Work Study. We do carry out the inspections and the requirements in your instruction manuals.

LAWRENCE (U.S. Army): In your article and also in your discussion you mentioned that in a new design, the first unit is set up and shop tested for approximately 24 hours under load. From this information am I to understand that you would ship units out without being load tested, other than just the first one of that particular run?

GREY: Yes, these reduction gears are built to the specifications given to us by the Navy, the Bureau of Ships, and this is a standard requirement in their specifications. For instance, we are supplying several destroyer escort units at the present time. The first one is on shop test at this very moment. By specification requirement we must test this first unit through 24 hours at full power torque and full speed and in addition give it an overspeed test.

With the subsequent reduction gears, we would simply have to overspeed them, and conduct marking tests that would indicate satisfactory gear alignment. This can be done without operating the unit at full power torque and full power speed. You can apply full power torque at low speed and mark the unit, and it is for this reason that we're suggesting that reduction gears of high horsepower or very large reduction gears be individually tested to an overload condition, because you have variations in one gear to another gear, manufacturing variations, thermal effects and so forth, that may show themselves once you get the unit actually operating.

LAWRENCE: How does this correspond then with units that you furnish commercially? Does anyone buy a number of units and specify that only one be load tested?

GREY: On commercial applications, the governing agency is ABS or Lloyds in addition to the specifications of the design agent for the company purchasing the machinery, and in this case we simply mark at low speed and conduct an overspeed test. I can't recall a commercial unit that has been full power tested, but there is a vast difference between a commercial, the average commercial reduction gear and the average navy reduction gear.

FRANKEL (MIT): After your marking process, say, for instance, a marking process that doesn't come up to your usual standards, what are the subsequent procedures? Another question — I don't know if you can answer — is what are the approximate factors you are using in the latest reduction gear trains?

GREY: The reduction gears have sleeve bearings, and there is no provision for adjustment. So when we test a gear unit and conduct a marking test, if we find that the marking is not satisfactory, we have several courses of action. If the marking is unsatisfactory to only a minor degree perhaps the housing can be scrapped, if the bearing reaction say in a bearing cap why some stock can be taken off the joint, some material scrapped out of the lower bore, and the bearing actually shifted without introducting any clearances in the system.

In some cases this cannot be accomplished. If the bearing diametral clearances are not critical, a bearing may be scraped a thousandth or two perhaps three to obtain the desired alignment. In some cases the misalignment is a function perhaps of a mismatch of the helical angles of the rotating elements, in which case the offending element can be removed and reworked.

FRANKEL: So you occasionally find that you have to work a kind of iteration process of manually refinishing the gears afterwards and by refinishing after your marking process and working on your bearings, finding the spots on your tooth profiles and then have to manually rework as well occasionally before remarking?

GREY: When you say rework are you referring to the profiles of the teeth?

FRANKEL: Yes, and also the hard spots.

GREY: The navy specifications prohibit any extensive modification to the tooth profile. In some cases, I believe it is permissable to end relieve teeth where loading is high, but outside of this the element would have to be perhaps re-hobbed or re-shaved to obtain proper marking if the element was at fault.

With respect to your question on K-factors, the Navy specifies the K-factors for the reduction gear and it's usually a function of the service that the vessel is going to be in. For instance, a destroyer where weight and size is a consideration, the K-factc may be quite high. The destroyer doesn't operate at maximum power but only a small percentage of its life time and so in this particular case we would have high K-factors

The volume of the reduction gear, the size of the reduction gear is inversely proportional to the K-factors, so with a large K-factor we have a smaller gear. With units that have to be operated at or near their maximum power rating for a good percentage of their life time, lower K-factors are selected.

We have gear materials that we use for navy applications. Here the stresses and K-factors are higher than the commercial reduction gears. Commercial reduction gears generally have K-factors in the order of 70-80-90, with the possibility of going somewhat higher.

DUNN (Electric Boat): I noticed in some specifications that you have the requirement to design for a life expectancy of perhaps 20,000 hours at full power as a design target. What are you going to do when you have to prove this?

GREY: I'm afraid I can't answer that one fully. I think we'll just have to design for it and observe the unit after it has operated at the elevated powers for certain time intervals, and see if the unit is holding up satisfactorily.

DUNN: Of course that is a trick question, but the gentlemen from Arizona this morning pointed out to us and prophesied for us that the time would come when we would have to design to failure rates, and we could forget all about factors of safety. We aren't even at the point yet where we can design and prove that our factors of safety are satisfactory for a predetermined requirement on life expectancy. In other words we do not know what is a satisfactory safety margin to account for wear, drift, aging and usage degradations.

GREY: I think that some of the information was based on a large sample of failures, and in propulsion equipment individual manufacturers don't put out enough units to get sufficient data. Designing for long life, 20,000 hours, or whatever the figure is, at elevated K-factors and stresses. The only way I can answer that is to operate the unit for some period of time at an overload condition of 115% of power torque for some minimum time interval. This would give a clue to the ability of the reduction gear to operate satisfactorily for the extended life.

DUNN: I was making the observation that for the kinds of machines that you are discussing which include such things as submarine hulls, bridges, buildings, etc. You said the failure rate analysis is almost an impossibility to achieve. Therefore we ought not to look so carefully and so precisely at a single method, especially one based on failure predictions for electronic piece parts. Mr. Lauser has evolved a method that should be considered in the cases where you are required to exhibit or demonstrate that your unit is satisfactory with 115% target. This method provides assurance and confidence that you can demonstrate that your factor of safety is such that you do have an adequate (for required age) separation between the significant maximum stress and the significant minimum strength.

KECECIOGLU (U. of Arizona): When we have a case like the one you mentioned where you can never put in enough test time, and you know that it is over designed, your only salvation, because you can't test to establish failure rates, is to be able to use the scientific method of estimating or predicting, at the beginning, at the drawing level, what the product failure rate might be, and then see if this is satisfactory.

The only salvation is to at least establish a range in which the actual product failure rate might be.

In one shot equipment nobody can tell you what the reliability is going to be. The only salvation is to go through a technique of reliability prediction and try to optimize the reliability in advance because you don't have the opportunity to go through tests and development.

DUNN: I agree with you 100%. My main point is simply that there is no one single method, scientific method, or statistical procedure that solves all the problems found in ships. The failure rate analysis and statistical predictions on failure rate are not at all satisfactory for all components and structures found in ships. We as ship builders and ship designers are concerned with methods which will be used in the future. The safety factor or the safety margin concept is obviously a better one to start with for those components which cannot afford to be multiple life tested statistically. Test to failure is generally all we can afford and that only in real problem areas.

Sub-test of parameter detail to failure forms extremely useful design data and would complement the safety margin type concept to control the endurance of such things as reduction gears, bridges and ship hull structures.

HALEY (New York Shipbuilding): I think that one of the things that affects the reliability of a reduction gear is the loading on the forward and aft low speed bearings, this of course is affected by the shafting alignment designed from the reduction gear aft. I wonder if you would care to incorporate this in your discussion?

GREY: Yes, this is an important consideration, there is no question about it. It is one of the more obvious ones. In my paper under the heading of "Installation and Alignment," I mentioned two areas of special concern, the reduction gear to line shaft alignment, and the turbine to gear alignment.

The gear manufacturers supply gap and sag measurements for the shipyards in order to accomplish this alignment. If it is not done properly or correctly, it can certainly cause internal misalignment of the reduction gear. The slow speed gear will cook and internal forces will be imposed on it. This can result in, and usually does in many cases, tooth distress or in extreme cases, tooth failure.

In the assembled shipment of the reduction gear, a certain amount of rigidity must be built into the reduction gear and we think that this helps a great deal in obtaining the correct internal alignment at the time of installation. However, it is still necessary to correctly align to the line shafting.

HALEY: I was particularly impressed because if your bearings are so designed that you can't take excessive loading due to the thermo growth of your reduction gear which raises your bearing up, and of course the old concept of aligning shafting was to have everything in a cold straight line with zero, zero gap all the way around. Of course this concept is completely changed over a period of years and we find that the bearing reaction or the reaction of the shaft toward the reduction gear bull gear will throw it out of alignment, and cause excessive wear. Most gear manufacturers review the ship builders calculations and make certain that not only the gap and sag figures are correct, but also in some cases have to allow for athwartship alignment due to roll up of the hournals in the bearings. These will affect reliability of the gears.

GREY: Yes, I know the practice is to request the shafting arrangement, the shafting drawings, so that the torsional critical calculations can be made of the system. In addition this information is used to establish the gap and sag measurements, the alignment figures that we usually place on our outline drawings which are supplied, of course, to the shipyard.

FRANCIS (Boston Naval Shipyard): I would like to comment on that point you made. Shafting calculations can point out two or three things. If the deflection sensitivity of the shafting, the longitudinal positioning of the bearings is such that relatively small vertical movements of .005 or .004 can significantly influence the forward or aft bull gear bearing, then gaps and sags may not be accurate enough in order to arrive at a reasonable evaluation of the manufacturer's specifications for the minimum or maximum loading on the bearing.

The other thing is that if you use these calculations, you may find that the radial clearances that you allow in your forward and aft bull gear bearings may be significantly influenced by the longitudinal sensitivity of the shaft, and you may specify .010, .015, .020 or .030 bearing wear down. If you look at this from the shafting alignment point of view, you will find that this will have a big influence on the change in your forward and aftward bull gear bearings.

GREY: My comment is that I agree.

WELLING (Bureau of Ships): In that regard our Bureau, of course, works out every shafting alignment and shafting connected gear alignment in the computor, and does compute the load on each bull gear bearing to keep it within limits. If it is not right we have the ship builder shift the line shaft bearing to correct this.

There is one other point I wanted to bring out. The Navy doesn't quite accept follow reduction gears without checking. You know there is a year guarantee period and during that year the gears get pretty well examined. They get examined by the Board of Inspection Survey, after the preliminary acceptance trial and after the final acceptance trial.

GREY: This is true, but the gear is in the ship now.

WELLING: Yes, but they can and do examine them after these trials which include a fc hour low power run. There have been complaints of course, and gear elements have be replaced by vendors during the guarantee period. We don't accept the gears blindly.

GREY: Yes, I didn't mean to imply that. I actually did state that in the case of the highly loaded reduction gears or large gears that the specifications are usually modified to include some overload testing.

RELIABILITY CONSIDERATIONS IN AN INTEGRATED MARINE STEAM TURBINE POWER PLANT

H. J. KAISER, Project Manager, Allis-Chalmers Manufacturing Company; JENS T. HOLM, Professor of Marine Engineering, Webb Institute of Naval Architecture; DAVE TAWSE, Combustion Engineering Inc.

INTRODUCTION

In the design of an automated power plant, a certain minimum level of reliability is a prerequisite. To achieve a satisfactory design, this paper proposes an inherently self-regulating plant and simple thermal arrangement. Further reliability of the final product is achieved by using shop assembled and tested installation units combined from individual components in a subassembly.

The American Maritime industry is at present engaged in a broad effort to improve its competitive position. Since machinery represents 25 to 35% of the first cost and up to 50% of the total operating cost, a substantial reduction in these costs would be a significant gain in competitive ability.

In the report of the Maritime Research Advisory Committee (ref. (a)) the following recommendation was made: "We believe that there is much that could be done to improve the steam turbine power plant by a serious attempt to create a truly integrated machinery arrangement. The present steam plant is now an assembly of more or less compatible elements, manufactured under varying circumstances by many different firms. Unless such integration is accomplished, the steam turbine may lose out in competition to the gas turbine, particularly since automation promises to become a factor of primary importance."

To implement these findings, the MARAD issued research contracts for two steam and two gas turbine studies. One of these is the Allis-Chalmers—J. J. Henry Co.—Combustion Engineering Co. study discussed in this paper. The study encompasses a self-regulatory low cost power plant capable of operation with an engine room watch of two men.

A. GENERAL CONSIDERATIONS

Before any power plant can be automated, it must achieve a certain minimum level of reliability, commensurate with the maintenance ability of the remaining operating crew, if any. In the steam power plant, such a level of reliability has out of necessity been achieved by the main units; the propulsion turbines and gears, and the boilers.

In addition, automation presupposes a certain simplification of the control problem by reducing the amount of control required. Assuming that reliability has been achieved, any power plant can be automated, but such automation would be a dubious economy. Additionally, this simplification can aid in achieving the required reliability of the basic plant and the automating equipment.

To achieve reliable performance in an automated plant, it is desirable that regulation be inherent and not achieved by additional controls, essentially resulting in a single lever control system. Many of the complications and maintenance problems arising in marine propulsion plants are not inherent in the cycle. Some arise from building up of safety factors and others arise from auxiliary fluid systems, principally fuel oil transfer, ballasting, bilge pumping and hotel services.

Mere duplication of equipment does not assure reliability. Essentially, reliability reduces to a function of the number of moving parts and the type of moving part. For example, a turbine rotor, without contact surfaces except for well lubricated bearings, has higher reliability than a reciprocating piston, an orifice is more reliable than a trap, etc. To this, must be overlaid the human factor, his carelessness and fallibilities.

Reliability is also a function of the quality of workmanship, both in the shop and during erection aboard ship. Working conditions aboard ship during construction are at best very bad. Indeed, it is remarkable that so few casualties occur traceable to faulty installation. The greatest enemy is dirt and foreign matter, with misalignment as a secondary factor. Without question, work done in the cleaner and more comfortable working conditions of a shop will be of higher quality than field erection work. It follows, therefore, that machinery should be more extensively subassembled in the shops. Only final unit connections and aligning of the main propulsion unit to the propeller should remain to be done in the ship in the ideal case.

To achieve our ends then, it was necessary to design a steam plant capable of single lever control by developing a power plant that regulates itself in response to load and in which the various components maintain a balanced relationship to each other. Such a plant must by nature be less costly to build since there are few redundancies.

A three pronged attack on the problem was made, as follows:

1. A cycle that permits the steam generator to inherently respond to demand without external controls was developed.

2. The number of fluid systems was reduced.

3. A minimum number of shop assembled installation units, with a minimum of interconnections were made by combining components.

A boiler can be designed so that as steam demand is met, an equivalent quantity of heat is added in the furnace by the action of the steam flow, resulting in a steam plant that is inherently self-regulatory. To accomplish this, we have employed a so-called series turbine as illustrated in Fig. 1. This turbine, a single stage wheel in a high pressure casing, acts as a power generating meter driving a fuel metering pump and combustion air blower at the correct speed to respond to steam flow.

Fluid systems were reduced by either eliminating the need for a system, or by combining functions, as follows.

Salt water is employed only in the main condenser (with the distiller condenser in parallel) and the flash evaporator salt water heater. All other cooling functions employ condensate from the main condenser. Lubricating oil is condensate cooled, together with air ejector and gland condensers.

One lubricating system is employed, common to the series unit, the main unit, and the turbo-generator. Only cold fresh water (for potable and flushing service) and electric power are supplied the deck. Heating is entirely electric, including hot water. Most deck house functions can be handled by domestic type equipment, including air conditioning.

The feed system involves only air ejector intercondenser, oil cooler, drain cooler, air ejector after condenser, gland steam condenser, deaerator and two duplicate motor driven feed pumps. Turbines are designed so that no drains are necessary; the air ejectors (using superheated steam) function as nozzle chest drains during warm up. The evaporator plant parallels the main condenser and is mounted on the main unit.

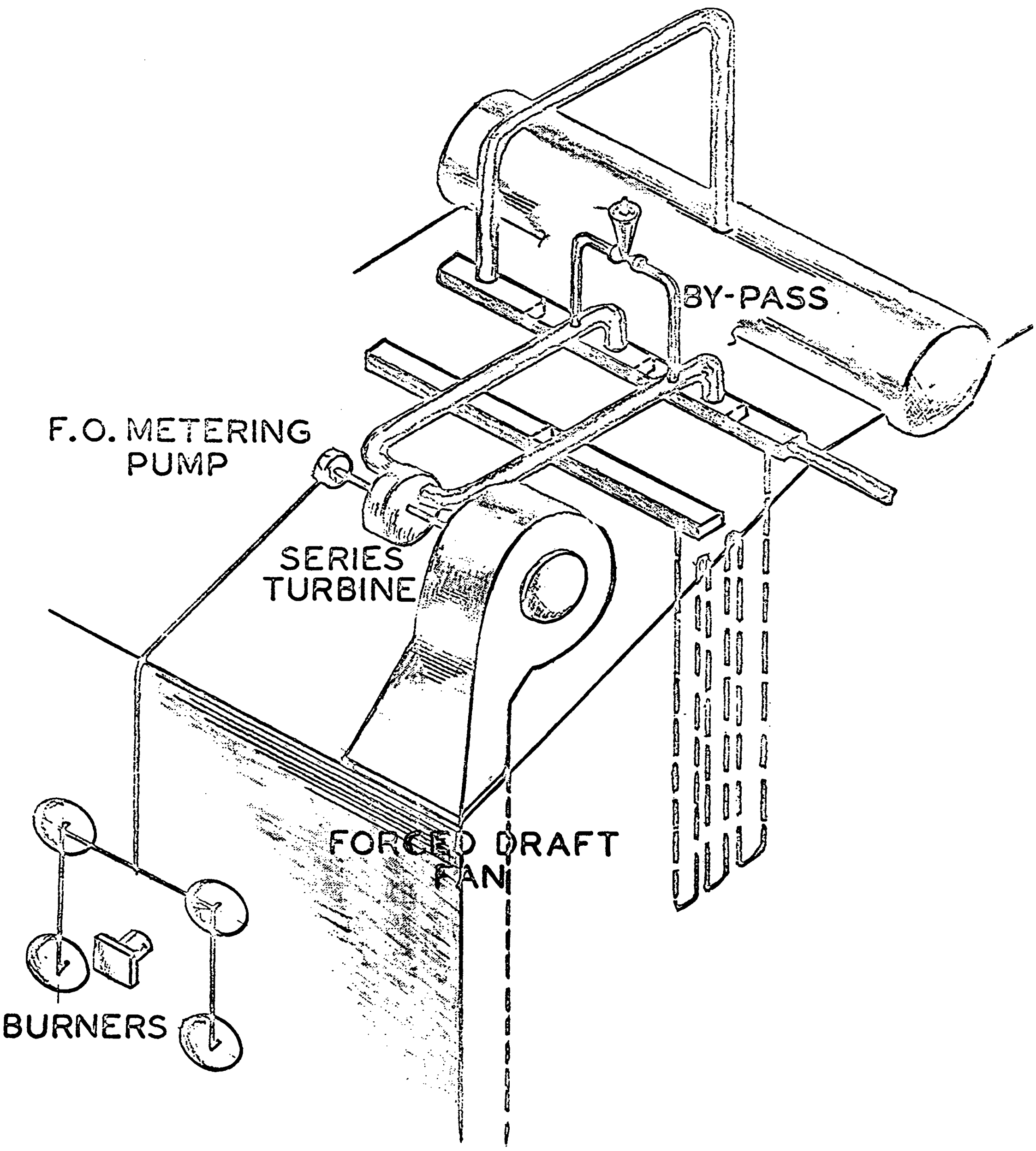


Figure 1. Series Turbine-Forced Draft Fuel Oil Metering Pump Unit.

Electric power at sea will be drawn from a single turbo-generator exhausting to a 35 PSIG auxiliary exhaust system which supplies heating steam for the deaerator, augmented by a bleed from the main turbines at 55 PSIG, through a reducing valve.

The bilge and ballast system consists of a single main led forward and aft of the machinery, with air operated remote controlled valves in each compartment. Three identical self-priming rotary pumps are used, acting as bilge pumps when running in one direction and as ballast pumps when reversed. Check valves will prevent accidental flooding of nonballast spaces.

A single main steam line leads from the boiler to the high pressure turbine nozzle valve chest. The nozzle valve chest contains the ahead nozzle valves and astern valve and acts as a distribution point for steam to the air ejectors and turbo-generator. No other high pressure piping is installed except the soot blower piping on the boiler. There is no main steam stop valve and no desuperheated steam system.

Combination of components is accomplished by providing two basic units; a boiler unit (one boiler) complete with series turbine, fan, fuel pump, feed pumps, steam atomizing burners, starting pilot burner, soot blowers, etc. all integrally mounted, and a turbine-gear-condenser unit with lube oil pumps, coolers, ejectors, and salt water flash evaporator, condensate, brine and distillate pumps, all mounted on an integral foundation structure.

The resulting plant operates on residual fuel oil at sea. In port, the plant may run on steam, using black oil, but is more advantageously run on the stand-by generators, using diesel oil. Diesel oil is also used for starting.

No system aboard ship is as difficult to automate as the fuel oil system, especially the fuel oil transfer. The most reliable system would undoubtedly use individual suction/filling lines to each tank, with valve manifolds in the engine space, in the usual manner. Such an arrangement involves a large amount of piping in the tanks and is generally expensive. A simpler, single main system, running fore and aft was finally adopted, with remote operated valves in each tank.

In designing as radically different a plant as this, a hull design concession was assumed, namely that fuel tanks would not be ballasted. This greatly simplifies fuel handling, eliminating the use of settling tanks. To eliminate the need for tank heating coils and the consequent risk of fuel oil in the condensate, tank heating is accomplished by recirculating heated oil fuel. In this system, two duplicate self-priming rotary fuel service transfer pumps take suction on the fuel tanks, passing the oil to a heater in the engine space. This heater is a refinement of the steam heated steam generator now commonly used, using steam to heat boiling water which heats the oil, all within a single shell. About half of the fuel passes to the boiler fuel metering pump, the remainder is recirculated to the fuel tank through a return line. A fuel main, with air operated remote controlled valves in each tank, with the return main, either inside, or as a chaser, underneath the main, returns the hot oil, with the same type of valve in each space. This fuel handling system is not only simple, but appreciably less costly.

This design was worked out in detail for a 20,000 SHP Mariner plant with the machinery located in the three quarters aft position. The arrangement of the plant is illustrated by Figs. 2 and 3.

B. THE SERIES TURBINE CYCLE

In the marine steam power plant, three separate powering needs must be met, namely, the main propulsion unit providing power at the revolutions demanded by the propeller, constant revolutions for driving ship's service generators (and indirectly other auxiliaries), and variable speed drive for those auxiliaries that must vary their output to suit load.

In most steam plants, the variable output auxiliaries are driven by constant speed motors, control being achieved by throttling the discharges or other artificial means. A few vessels use turbine driven forced draft fans with variable speed drives but in no case has a separate, load responsive turbine been used for these auxiliaries.

Fig. 4 illustrates the cycle used in this plant. The load response turbine, driving the forced draft fan and fuel metering pump, is placed between the second and third pass of the superheater, in the same manner as a reheat installation. The cycle is simple in the extreme, beginning with the main condenser, condensate pump (constant speed, motor driven) and first stage air ejector, followed by the lubricating oil cooler, drain cooler, gland condensers and deaerator. The feed pump is constant speed motor driven. The constant speed turbo-generator uses main steam and exhausts to the deaerator. Additional feed heating steam is drawn from the 80 PSIA bleed point of the main turbine unit through a constant pressure regulator. Low pressure steam is bled for the flash evaporating plant.

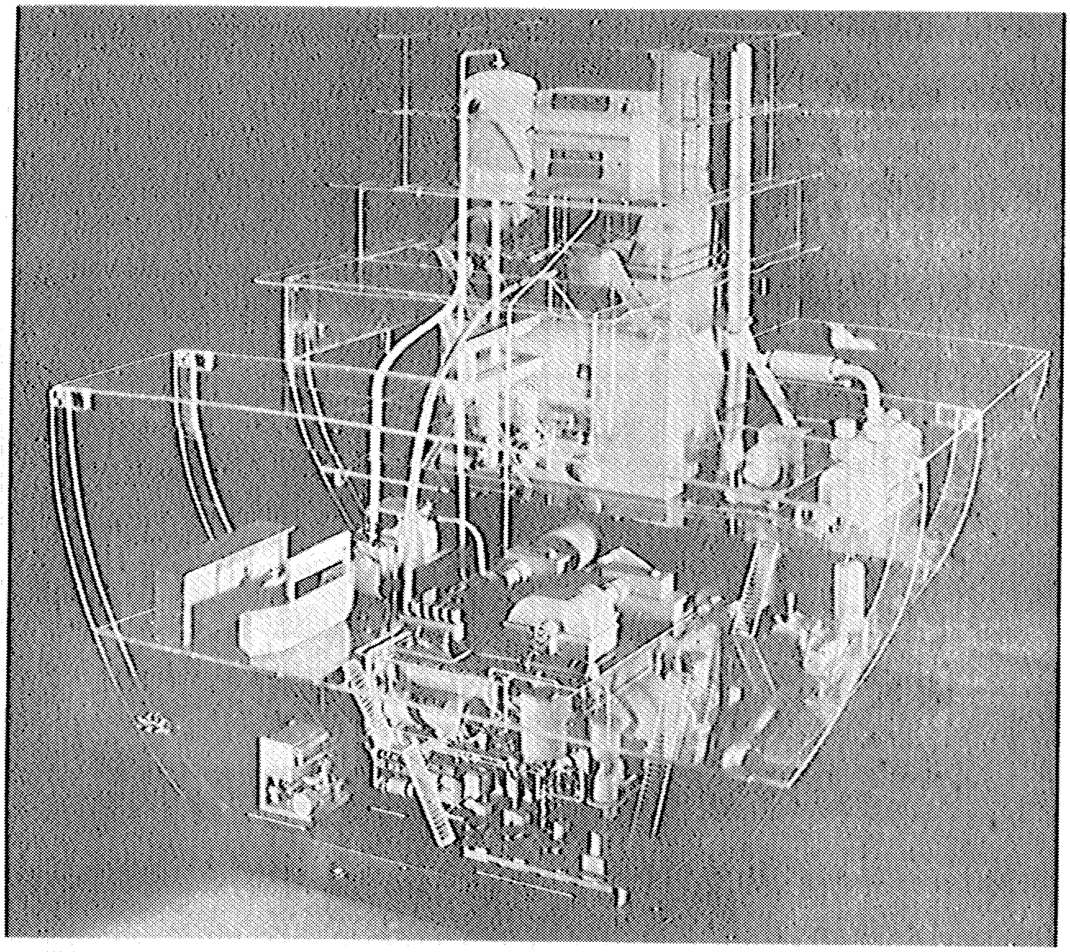


Figure 2. Integrated Marine Steam Turbine Power Plant Model.

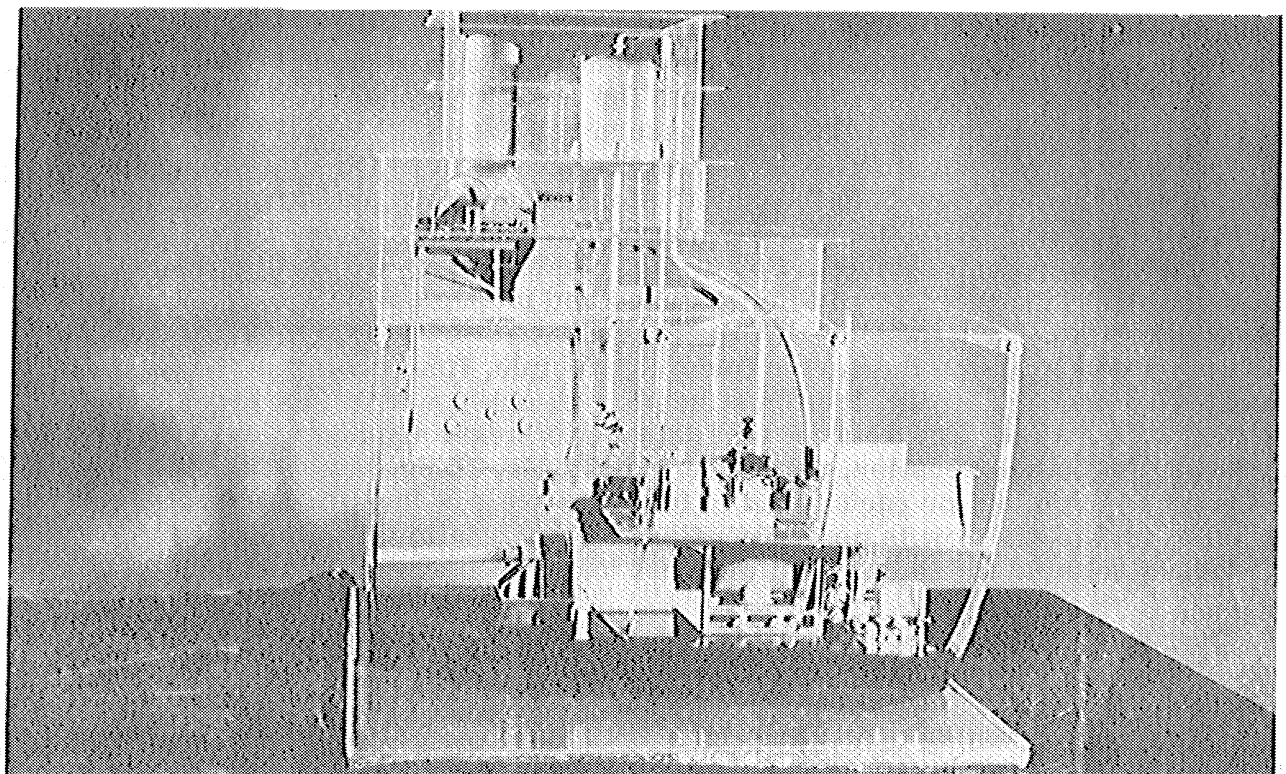


Figure 3. Integrated Marine Steam Turbine Power Plant Model.

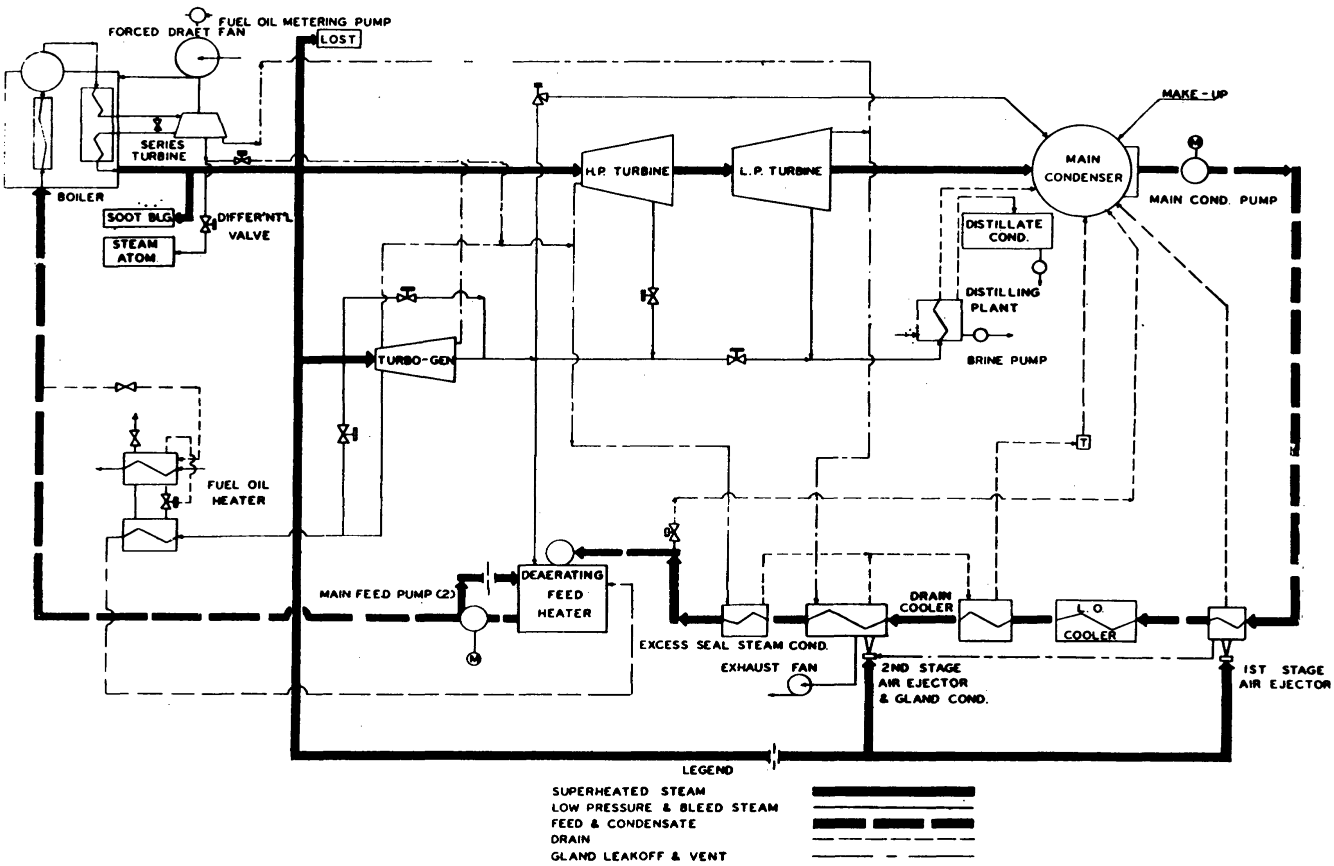


Figure 4. Flow Diagram of Self-Regulated Steam Turbine Power Plant.

To assure adequate oil cooling under all conditions, a thermostatic recirculating line is provided to recirculate condensate to the condenser whenever the condensate leaving the seal condenser rises above 145°F. A bypass regulator on the turbo-generator will provide additional feed heating steam for the deaerator during maneuvering and a cross connection from the auxiliary exhaust system to the low pressure distiller system through a pressure regulator.

This cycle has been simplified so that there is no need for extensive control devices, except for a few "off-the-shelf" steam regulators and a level controller for the deaerator. The cycle is not a maximum efficiency cycle, but will achieve excellent service performance. For a pilot plant design such as this, the added complications of higher steam temperatures and multiple feed heaters were deliberately avoided.

Let us now examine the series turbine principle, first reviewing some basic theory. To produce steam, fuel must be burned, one pound of fuel producing about 14 pounds of steam. The ratio of fuel demand to steam demand is essentially constant. With good combustion, using 15% excess air, 16 pounds or 233 cubic feet of air must be supplied for each pound of fuel oil. We see then, that the ratio of steam to fuel to air flow is nearly constant. Unfortunately, at very low rates this ratio does not hold for the air and we must make special provisions. More of this later.

The series turbine receives all the steam from the second pass of the superheater (at 675 PSIG and 715°F.), exhausting to the third pass. The pressure drop is small, only 22 psi at full power, so that the behavior of the steam may be approximated by assuming that the specific volume of the steam is constant through the machine. This means that the steam velocity leaving the turbine inlet nozzles is linearly proportional to flow, and if turbine blade speed-steam speed ratio is preserved, the series turbine RPM is linearly proportional to steam flow. Note also that the pressure drop and enthalpy drop though the turbine nozzles varies as the square of the flow and the power developed varies as the cube of the steam flow.

A centrifugal forced draft fan will deliver air in linear proportion to its revolutions, developing a static pressure in proportion to the square of the revolutions, absorbing power in proportion to the cube of the turns. The boiler draft loss (the air-gas flow resistance) varies as the square of the flow as well. Thus, if the fan is coupled to the turbine, a convenient proportionality exists between steam flow and air flow.

A rotary positive displacement fuel pump will display the same ratios of flow and power. We thus have our power developing meter in the steam line, responding to load.

Our studies indicate that this device is quite feasible except for the very low firing rates. In this range, it is customary to use more than the usual 15% excess air. We are in an area of unknowns here, since we are not sure that this high amount of excess air is really necessary or if we cannot throttle down the air flow properly. To handle this problem, a motor driven booster blower is provided, designed to start automatically at about 20% boiler load. Such a boost may not be necessary, however, since natural draft may produce the necessary boost effect automatically.

Since the forced draft fan pressure must be adjusted as the boiler fouls, a discharge gate is provided. Also, to exactly adjust the turbine to the power demand, a bypass valve is provided. With these two manual controls, the boiler unit may be "tuned" for proper excess air.

We should also note that any change in pressure at the series turbine inlet must result in a change in specific volume and, therefore, a change in steam velocity. A reduction in pressure results in a rise in turbine RPM giving the unit a stable operation sensitive to pressure changes.

One result of this characteristic is that the steam temperature entering the series turbine may vary slightly with steam flow rate. A loss of steam temperature will result in a reduction in steam pressure and, therefore, there will be a small adjustment in steam pressure with load.

In the course of investigations of the series turbine, numerous examples of this scheme were found including an application in a power plant in Manheim, Germany. This system, however, seems to be the only one where the fuel metering is also done by the series turbine.

Operation of the series turbine boiler unit has been simulated on the Analogue Computer at the Naval Boiler and Turbine Laboratory. First trials indicated satisfactory response except for excessive fluctuation in drum water level. By making modifications in the boiler convection bank and in the circulating characteristics, this fluctuation was reduced to a satisfactory degree. As now arranged, the unit has excellent response and satisfactory stability without the use of modulating controls of any sort. For the pilot design, a simple air operated pressure regulating valve will be incorporated in the steam bypass line to provide control if needed. The analogue study indicates that such a valve will have the effect of speeding up response.

C. SINGLE BOILER CONCEPT

Investigation into the design of the boiler was begun by first compiling a list of features considered to be essential for any boiler to achieve maximum reliability.

The design requirements can be broadly categorized into the following principal groupings:

a) Provide the ability to burn over a wide range, inexpensive, low quality, untreated fuel oil for long, continuous periods without extensive fouling of heating surfaces or burners.

b) Build in maximum component reliability aiming at extra long life and the lowest maintenance factors.

c) Employ construction features to lower costs and in-service repairs.

The first place to look when studying reliability is to scan back over the years and determine just where the major items for maintenance have occurred.

A study of these items has revealed that the major items are:

a) Refractory maintenance
b) Soot accumulation
c) Accessory failures

Starting at the furnace of the boiler we will consider refractory maintenance first. The most positive guide to furnace temperature is the release rate per square foot of effective projected radiant heating surface; this release rate and the furnace temperature are a good indication of how hard the furnace is working and, therefore, of wear and tear on refractory. If we can reduce the furnace release rate and furnace temperature, the maintenance on the refractory will lessen (assuming good practice is used in bringing the boiler up to pressure from cold condition).

There are two ways to reduce the furnace temperature, one being to increase the furnace size until we can install the radiant surface necessary to produce these cooler temperatures and the other is to line the furnace walls with as much radiant surface as possible, including floors and burner walls.

The trend in recent years is to allot less space to the machinery and demand that more steam be generated in the smaller space allotted. This has led us to the release rates of today. The boiler manufacturer has designed furnaces so that they will be completely water cooled including water cooled floors and burner walls, so there is maximum radiant surface for a given furnace volume, the refractory being completely covered by tangent tubes.

Another method to lessen or eliminate refractory problems is to use a welded wall construction which is becoming increasingly popular in other fields of boiler operation. This is done by welding tubes together, usually with a bar space between tubes. A gas tight furnace envelope is the result. The need for refractory is eliminated and all that is necessary is an insulating blanket and a light skin casing. This method is widely used on shore side boilers but has not yet been applied in the marine field.

If the welded wall marine boiler became a reality we would not have to size the furnace for release rates in that there is no refractory to consider; the furnace volume would then be based upon the ability of the furnace to complete combustion within a given space.

Fig. 5 shows that the burners are located so as to fire at right angles to the steam and water drum. This was done to direct the gases uniformly across the superheater and generating banks. From tests on highly rated Naval boilers where the burners fire parallel to the steam and water drum we have found that the gases tend to build up at the rear of the furnace before turning to the screen bank and superheater. This leads to high mass flows and transfer rates in localized areas with high tube metal temperatures in the superheater in that area. The generating bank directly opposite the burners has a nose shaped cone from just below the pendant superheater. From two dimensional smoke tests we have determined that this furnace configuration would tend to give us a better mixing and burning of the fuel in the lower half of the furnace, and also contributes towards a more even gas

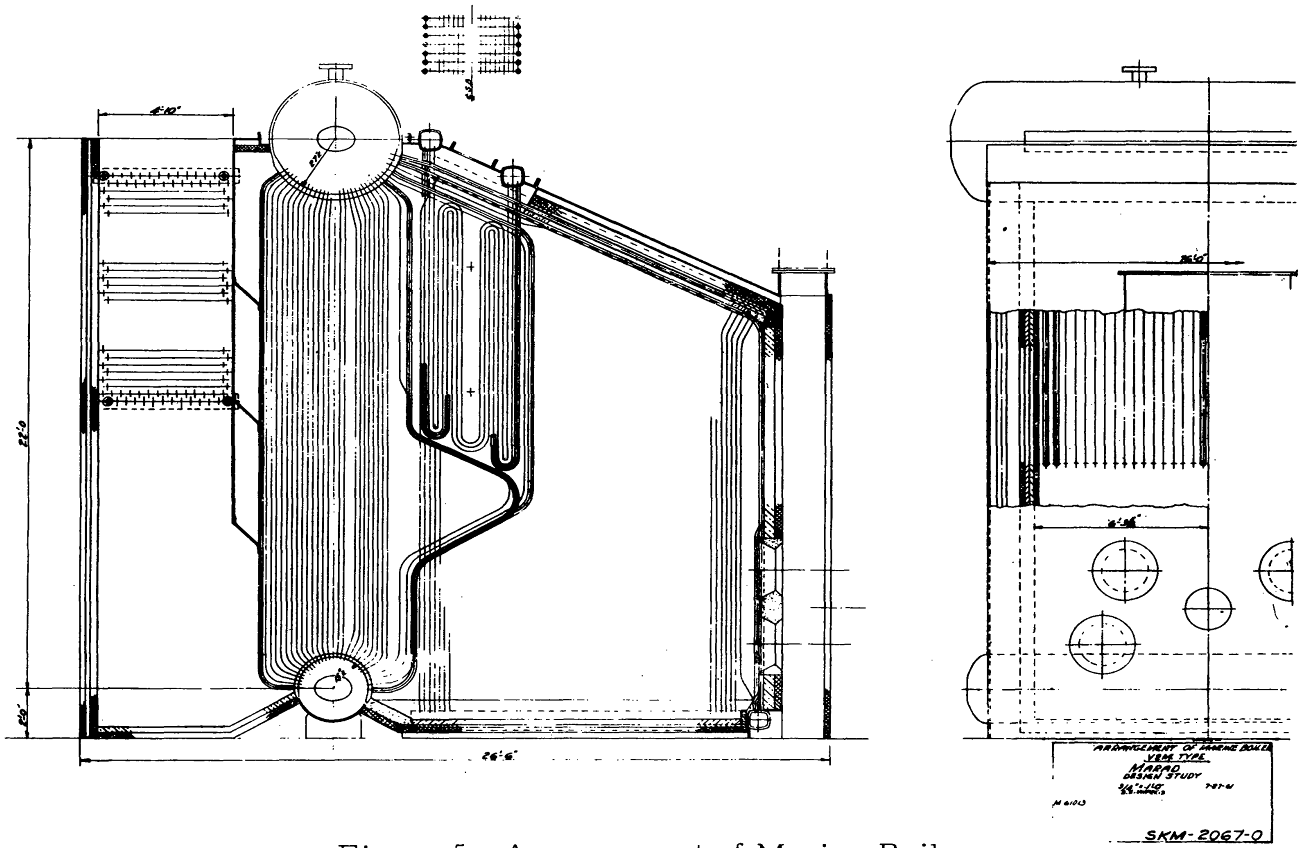


Figure 5. Arrangement of Marine Boiler.

flow pattern across the superheater. The gas path is improved by making the gases double back before entering the superheater, giving better chances of fall out of noncombustibles in the furnace instead of on the superheater surfaces.

A water cooled floor is not included in this design. This could be easily accomplished, however, the cost versus the lowering of the furnace temperature indicates that its inclusion in the design does not become significantly important until we consider the possibility of modular furnace welded wall construction.

The second item to be considered is the superheater area which is vulnerable to slag attack and corrosion from slag attack. The trend towards lower grade fuels and higher steam temperatures has led to the slag accumulation problem which exists today in marine boilers. Excessive slag in the superheater contributes towards less efficiency, high fuel consumption, corrosion and larger overhaul periods.

Slag will form on the superheater tubes if the gas and tube metal temperatures are above the ash fusion temperatures of the gases. Some of the various ways to minimize this condition are:

a) Water wash the fuel before it is introduced into the furnace.

b) Remove the superheater to an area in the boiler where the high gas and tube metal temperature no longer exists. (After the generating bank.)

c) Reduce the absorption rate of the superheater.

Water Washing of the Fuel – Brine is always present in crude petroleum; depending on the facilities at the refinery, small quantities of salts may be left in the crude oil during processing. Since these salts are nonvolatile, they tend to be concentrated into the bottom products and go into the heavy fuel. Water washing of the fuel has proved successful in the past. Where boilers have burnt low sodium fuels, cleaning of the boilers has been extended to periods of over twelve months. The scale that results when using this low sodium fuel is soft and can be easily removed.

The boiler design, however, should incorporate features which would minimize the slag problem without having to go to the expense of installing a purifying station to clean the fuel.

Superheater in a Different Area – Placing the superheater after the generating bank means that a greater amount of surface is necessary for the superheater due to the log mean temperature difference being smaller in the lower gas temperature zones. In order to keep the superheater surface within reasonable limits the gas temperature leaving the superheater must be greater than what is normally employed today leaving the generating bank, hence a large economizer is necessary to reduce the gas temperatures to a level required for boiler efficiency. The alternative to these measures would be to increase the transfer rate which would increase the duty on the forced draft fan. Even when this is done part of the superheater still has to be of alloy material. Generating bank surface is relatively inexpensive to install compared to superheater and economizer surface.

Reduction of Absorption Rate – We can reduce the absorption rate of the superheater and, therefore, reduce the tube metal temperatures by opening the spacing of the superheater tubes. This reduces the mass flow and transfer rate of the superheater, lowers the gas film temperature drop and reduces the tube metal temperatures. This will mean slightly more surface than is normally employed and if we keep the tube metal temperature close to the steam temperature the metals employed would be about the same as if we located the superheater behind the tube bank. If the tube metal temperature is limited to less than 1050° F., the condition for slag build-up would not exist.

Referring to the sketch, the superheater is composed of 2 inch tubes on 4-1/4 inch centers which is 2-1/4 inches clear between tubes. This is about three times the normal clearance which exists today and the chances for slag build-up are greatly lessened with this wide spacing. The superheater is of the pendant type which means it is nondrainable, however, in this reliability study one boiler per ship is considered and the boiler is to be on the line or under steam for the entire period between overhaul (twelve months assumed between overhauls).

The generating bank has a long history of successful operation and can be considered reliable for the requirements of present day operation. The design shown on Fig. 5 is about the same construction as normally employed in present day marine boilers.

For heat recovery equipment the economizer has proven to be the most rugged, reliable, easily operated piece of equipment known today, therefore, a cast iron extended fin surface is employed.

In the design of the boiler accessories, the following items should be considered:

Oil Burners – There should be steam assist to enable ship to run for long periods without changing burner tips. Tests have been conducted with these burners without changing tips and the end of the test was terminated by the end of the voyage (about ten days). A high turn-down ratio (16 to 1) can be achieved with steam assist burners, so for changes in evaporation it is only necessary to adjust the oil pressure to the burners.

Boiler Mountings – We have reached the conclusion that for some of these items (gage glasses, blow-down valves, safety valves, drain valves, etc.) it would be justifiable to use equipment which is designed for operating pressures and temperatures much higher than those required by the plant under consideration. The additional margins are calculated to result in a greater dependability and in turn lower maintenance. The Navy has been forced to adopt this design philosophy in connection with boiler gage glasses for 1200 psi service. After experiencing excessive maintenance with the 1500 psi standard gages, the Navy resorted to 2400 psi standard gages to achieve an acceptable dependability.

The boiler as shown in Fig. 5 is based on using one unit per ship. A study into the reliability of the unit revealed that the risk in loosing a propeller at sea measured in ship years would be in the order of 800 years. The same risk in dry firing a boiler causing partial damage at sea is in the order of 6,000 ship years.

D. PROPULSION PACKAGE

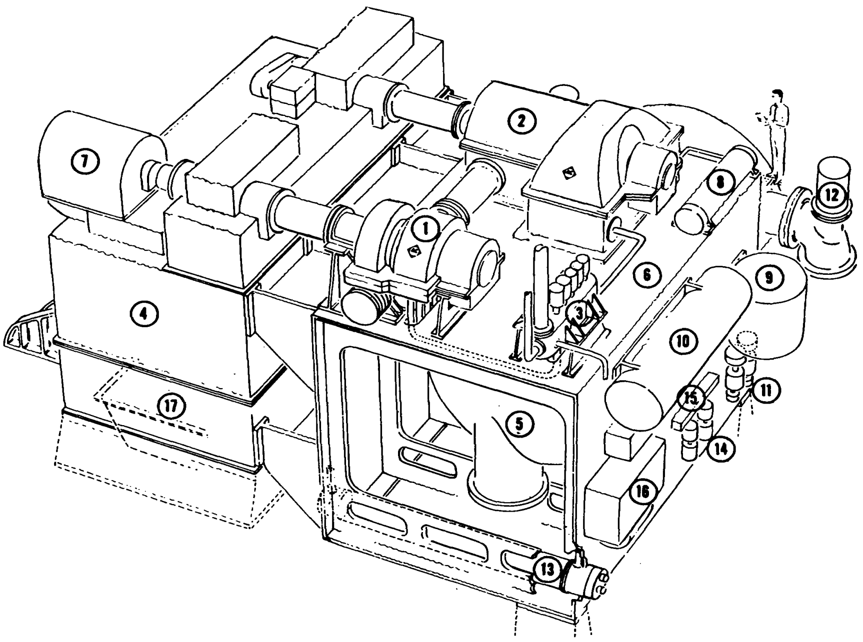
In our study of an integrated propulsion plant our concern for reliability was focussed on the evaluation of three main areas: first, the elimination of redundant components, second, the self-regulation of components and third, the simplification of design in known regions of casualties.

In the following discussion of the packaged turbine-gear-condenser unit, it will be shown how the study of the above three areas resulted in a reliable, self-regulated, integrated propulsion package.

The main propulsion package shown on Fig. 6 consists basically of five major items: the high and low pressure turbines, the gear, the condenser, the support structure, and the system components.

The high and low pressure turbines are cross-compounded, single flow connected with a low level cross over. This low level cross over allows maximum ease of lifting turbine casings. The low pressure unit has the astern turbine located in the forward end of the casing resulting in a short piping connection to the valve chest.

A separate valve chest, main steam strainer, and distribution manifold are housed in one component. This is one example of elimination of individual components. It became apparent that it would be desirable to isolate the steam chest from the high pressure casing. Doing this accomplished three basic objectives:



1. HP turbine
2. LP turbine
3. Separate valve chest
4. Main reduction gear
5. Main condenser
6. Support structure
7. Emergency take home motor
8. Salt water heater
9. Flash chamber
10. Vapor condenser
11. Brine, distillate pumps
12. Main circulating pump
13. Lube oil cooler
14. Main condensate pumps
15. Air ejectors
16. Gland seal condenser
17. Integral lube oil sump

Figure 6.

1. It resulted in a short, simple, steam line from the boiler to a multi-purpose steam chamber for the distribution of high pressure steam to the turbines, turbo-generator, astern turbine, and air ejectors.
2. It simplified the high pressure turbine casing by removing the thermal stress resulting in a more reliable and safe unit.
3. It lends itself to the single lever control which will operate the turbines in the ahead and astern direction.

The remote valve chest proposed is a functional multi-purpose component with minimum valves which will allow warming the boiler and turbines simultaneously, and automatically bring the main propulsion turbine to a ready-to-operate condition.

The gear is a double reduction articulated type with slightly higher than conventional K factors. The factors chosen for this study were 95 and 75 for the first and second reductions, respectively.

An attached lube oil pump serves the main units down to approximately one-half speed. A separate stand-by motor driven lube oil pump is mounted between the turbine and gear. In addition, a gravity lube system of six minutes' supply is incorporated to increase reliability and safety. The sump of the reduction gear is integral with the main gear casing eliminating the sealing and cleaning problem during installation. The gear would be prealigned with the turbines at the manufacturer's plant.

The condenser is basically a single-pass athwart ships hung type which lends itself to minimum maintenance and allows free expansion during ahead and astern maneuvering. Two circulating pumps are mounted on the port side for ease of maintenance and operating flexibility. These can be operated separately or in parallel for optimum performance. Sufficient space is available for tube removal in the event this should become necessary.

The condenser is encapsulated within the support structure which gives sufficient rigidity, yet allows free expansion. Restraining keys are utilized to prevent sway during periods of heavy rolling and pitching.

The support structure is a rigid frame type box which will support the high and low pressure turbines, the condenser, and the system components. It provides a rigid foundation to facilitate alignment of the turbines to the reduction gear and provides a means of packaging of the entire propulsion unit. The structure is fitted to the gear casing by machined, flanged pads which have fitted bolts and doweling. The structure would be fabricated and assembled together with the turbine, the gear, and the systems components at the manufacturer's site.

Foundation checking has been reduced by the use of four support pads, two for the support structure and two for the gear with a supporting foundation for the thrust bearing. The unit would be aligned at the factory after manufacture and shipped as an assembled unit or in three packages depending upon the final destination and shipping limitations.

The system components shown on Fig. 7 are mounted on the forward wall of the support structure. These components consist basically of the distilling plant, air ejectors and gland exhauster system, condensate system and lube oil cooler. Such an arrangement allows for a unitized system, eliminating separate foundations, piping, valves and control equipment. The distilling plant consists of a salt water heater, flash chamber, vapor condenser, and brine and distillate pumps. The flash type of distiller was chosen because of its simplicity and low maintenance characteristics. The low pressure extraction line supplies steam to the salt water heater. The hot salt water passes into the flash chamber, vaporizes and passes into the vapor condenser. The distillate is pumped into fresh water tanks from where make-up, potable and flushing water will be furnished for needs of the ship.

The distilling plant is self-regulating in that it operates without separate

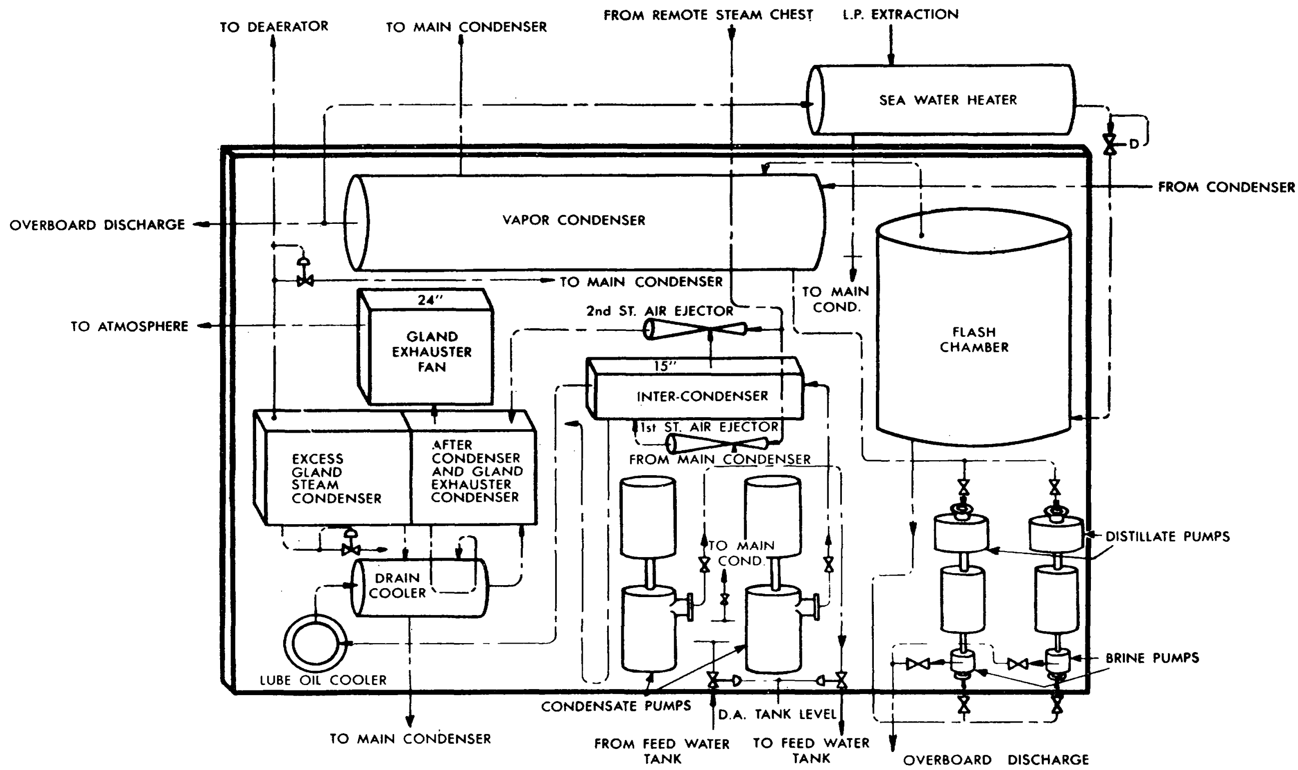


Figure 7. Flow Diagram of Auxiliary System Components.

control equipment. For normal operation, a single temperature controlled valve on the inlet of the flash chamber is required.

The condensate pumps, mounted on the support structure, take a suction from the main condenser and discharge through the intercondenser, the lube oil cooler, the drain cooler, the after condenser, the excess gland condenser and to the deaerator. You will note that condensate is used rather than salt water for cooling purposes. We have eliminated all salt water cooling with the exception of the main condenser and the vapor condenser. This will reduce corrosion and maintenance problems.

In line with the concept of completely integrating the components of this plant, we are featuring a single gland seal steam manifold and a single gland steam vent manifold. This system will be common to the series turbine, to the turbogenerator and to the main turbines. The system proposed is unique in that it is designed to automatically perform its functions. It is also different from a conventional propulsion plant in that the heat of the excess gland steam is recovered by employing a separate gland steam condenser.

The gland steam manifold will automatically pressurize when steam is generated in the boiler. It does this automatically because there are no stop valves in the main steam line to the remote valve chest. As steam pressure is established, excess steam from the steam atomizing line from the series turbine will furnish an adequate amount of gland sealing for the main turbines.

The series turbine, as previously mentioned, is located between the second and third pass of the boiler superheater. It is there primarily for reliability and cost reduction. This location limits the maximum operating temperature of this turbine to approximately 710° F. resulting in a simple, single-row impulse turbine which can be manufactured with low-alloy materials. Since this turbine is in the main steam circuit it is subjected to a pressure of about 670 pounds at the inlet and about 635 pounds at the exhaust end. This very low pressure drop across the turbine results in heavier than normal exhaust flanges, and slightly higher leakages

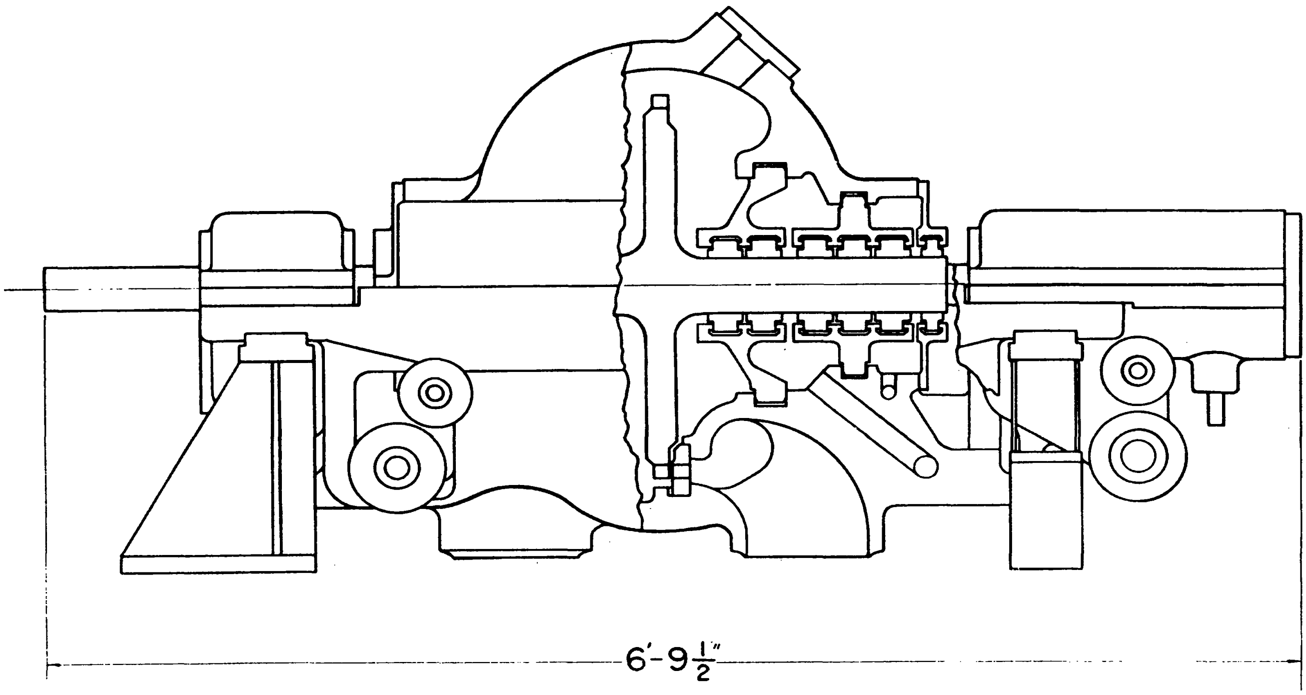


Figure 8. Proposed Series Turbine (Conventional Bearing and Sealing).

which we will utilize for steam atomization. The unit we have proposed will employ conventional bearings, lubrication and gland sealing. Bearing centers are roughly four feet with a single row wheel of approximately 26 inches in diameter. The unit has a 150 pound leakoff to the steam atomizing burners and a second leakoff to the gland seal system.

Two alternate designs were investigated for use of an integrated series turbine-blower-pump unit. One was the use of steam lubricated bearings which has several advantages such as eliminating the need of lube oil, less space with only one foot between bearing centers, and less leakage. In this arrangement the turbine and blower are mounted on circular flanges connected together by ribs and supported on a hinged plate. The center bearing housing contains a thrust bearing, a set of spiral gears to drive the fuel pump and an overspeed impeller. The reliability factor on this unit was not considered sufficient in comparison to the unit we have proposed.

A second alternate of the series turbine was considered. To combat the problem of having a long machine due to the large amount of shaft sealing needed, it was decided to study the feasibility of designing an overhung shaft machine. This would reduce the sealing length by one-half but would demand a more sturdy shaft. It was considered as being impractical unless it could be integrated completely with the fuel oil pump and forced draft blower. Such integration would result in a special fuel metering pump and a special blower which would increase costs and again reduce the reliability factor.

We feel the features of the series turbine we have chosen will satisfy the requirements we are after, especially reliability.

We have successfully eliminated some components. For example, we have eliminated the gland seal steam regulators and supply line. Some components have been eliminated through unitizing with already existing equipment. The distilling system typifies this arrangement. By using the main condenser vacuum and the circulating system, we have not only eliminated a complete set of air ejectors, but also an air ejector condenser, a source of cooling water, circulating pumps and heater drain pumps. Self-regulation is achieved in the distilling system since no control valves, except for a temperature regulated valve, are used. Integration of the valve chest and distribution manifold is another example as to the simplicity and reliability. In this instance, we have eliminated all valves between the boiler and the distribution manifold. Supporting the auxiliary equipment on the forward wall of the support structure eliminates separate foundations, reduces piping costs, and allows for compactness. The equipment can be arranged, predowelled, piping prefabricated at the manufacturer's site, ready for installation.

Self-regulation or what might be called simple automation shows up in the flash chamber of the distilling plant in that this is kept constant by the effect of the brine pump seeking its proper suction level. The self-draining valve chest through the air ejector controlled by a simple orifice is another example of self-regulation.

Elimination of redundant components and valves reduces maintenance. Restricting salt water to only the main condenser, and vapor condenser, adds to reliability.

The present concept has been evolved and has been realistically evaluated. Still more improvements can be made during the final design of such a unit.

E. OPERATION

Because of the inherent plant characteristics and reduction of fluid systems, this plant requires very little in the way of controls. In a plant console no bigger than that now used in a nonautomated conventional plant, complete operating control can be centralized, with a single lever doing the actual control. Even bilge and ballast and fuel tank selection is readily centralized.

This plant was designed to be operated with two trained specialists in the engine space and a total vessel complement of eighteen men. With bridge control of the turbine throttle lever, both engine men spend their time in servicing and maintenance duties. The savings with such a reduced crew, aside from yearly salary costs, also include the savings in the cost of crew accommodations.

The fuel rate for this plant, .514 with an eighteen man complement is 2% higher than that normally achieved at 20,000 SHP with these steam conditions. This 2% high fuel rate means an added cost of $10,000 per year for fuel, a small price to pay for the advantages accrued.

F. ECONOMICS

The economies to be effected by reduced manning scales are sufficiently well understood that they need not be repeated. Much less obvious is the possible savings in installed first cost. By eliminating redundant equipment and using remaining equipment to the utmost, combined with inherently automated operation and large shop assemblies, savings in first cost are large. A careful estimate of costs for the 20,000 SHP plant indicates a total saving of better than $700,000 for the complete installation, excluding cost saving from reduced accommodations. This more than balances the 2% increase in fuel cost.

A high level of engineering design quality is necessary in a plant such as this. It was repeatedly necessary to investigate obscure design details because of their effect on other components.

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(a) Proposed Program for Maritime Administration Research – Report by the Maritime Research Advisory Committee National Academy of Sciences.

(b) Reliability Analysis of Present and Proposed Steam Turbine Inlet and Stop Valve Systems by Dr. Dimitri Kececioglu.

DISCUSSION

THAYER (Lykes Bros.): I would like to ask if your single main bilge system with the air operated valves has been approved by the Coast Guard?

HOLM: It has not been specifically approved by the Coast Guard but it has been designed in such a way that it meets the written requirement of the Coast Guard.

ARNOLD (United Aircraft): Is your .514 plant rate an overall rate including distilling water and so forth, or is this just the propulsion plant?

KAISER: That fuel rate is for the entire plant, hotel services and everything.

KIN (Esso International): I've a serious statement to ask you. What is the lowest excess air that you can get with this arrangement?

HOLM: The lowest excess air rate that you can get with this will not be a function of the series turbine at all, but a function of what we can get out of the burner in the furnace.

KIN: Another question: when you speak of reliability in this type of an installation, and I don't think there are any around here, how do you account for the reliability?

KAISER: We did not make a statistical analysis of this plant on reliability. The design philosophy that we used was to down grade or simplify the existing types of cycles that are in existence today on merchant ships. To simplify this then, or to down grade it, to make it less complex, inherently we're going to get more reliability. With a less complex system, with a simple unit and combined system components, reliability is very definitely going to increase, but as I say we did not go into an analytical, statistical analysis of the reliability on the plant. This I think of course is a definite area where a study could be made.

MCMULLEN (McMullen Assoc.): I have two specific questions: On your estimate of the $700,000 saving in capital investment including cost of material and installation, I would like to know if you have received firm quotations for the manufacture of some of this equipment which has yet to be designed. It seems to me that until a number of units have been built, your cost will probably be greater than the conventional type of plant which is available today. After all, General Electric offers a 20,000 shaft horsepower packaged power plant at the present time in the range of $1,500,000 and if you can produce one at $800,000, I think you will get quite a few orders.

My next question has to do with the reliability that you put on the dry firing of the boiler versus that of the propeller. I notice that you gave the propeller 800 ship years of life, which is a very high reliability factor. Then you gave the dry firing possibility approximately 8 times greater reliability. Well, as you may know, the chief engineers have insurance against loosing their licenses, and I know in my own particular case several clients of ours in at least three separate instances in the last year have had boilers dry fired in port.

Now it seems to me that you've got to crank in some sort of a human reliability factor into this thing. What I'm questioning actually is this reliability of this dry firing being 8 times greater than that of the propeller and shafting.

TAWSE (Combustion Engineering): You must have had a bad year last year. This is from tables from ABS, insurance companies and shipping companies. This is all I can say.

KAISER: I would like to make a comment on dry firing. I've seen it happen where the water has gone out of the guage glass and the fires are still on, and when this came to the attention of the oiler or the fireman, immediate panic ensued and all fires were secured, and you could call that dry firing, but yet there was no damage to that boiler, and I'm sure that if you are in port and you momentarily loose your water in your gauge glass, you're not dry firing that boiler to the extent of failure. The boiler will take a certain amount of dry firing if you catch it in time.

FRANCIS (Boston Naval Shipyard): I noticed that you did away with the ballasting of fuel tanks. If this were a requirement, how did you fulfill this requirement since you did away with the ballasting?

HOLM: Actually we investigated this in the case of the Mariner and we found that on that particular design at least it was possible to do without ballasting fuel tanks. This requires a certain amount of modification in the existing designs, but it can be done.

Also while I have the floor, I would like to answer John McMullen's other question about the $700,000 which we have seemed to have glossed over. That was not deliberate John. It is perfectly right that you can get these phenomenonly low quotations from some companies now adays, but remember this is not the installed cost of the package. We're talking about the total cost of the installed plant in the ship, and that is nearer $4,000,000 than it is $1,500,000. So on the basis of the fact that we expect to reduce a great deal of the cost of installation, we can save money there. In addition to that, the use of a single boiler instead of two definitely cuts down on your costs — not only the direct cost of the boiler itself, but in the piping, and the additional apparatus that is necessary with two boilers over one. It is surprising how much you can eliminate by going to one boiler.

MCMULLEN: I accept this, but what about the quotation on this material? How much was the material cost reduced?

HOLM: I can't tell you offhand John. We would have to go back over our records.

KAISER: We did get quotations on all the equipment. These were not hard and fast specifications that we sent out to individual suppliers. They were more liberal specifications since the actual design of equipment had not been finalized. The quotations were estimates from the standpoint that these were not hard and fast specifications.

HIRSCHKOWITZ (U.S. Merchant Marine Academy): With respect to reliability and getting back to our fuel tanks, if I recall correctly you suggested eliminating the settling tank. Are you considering bottom tanks? When they run over a sand bar what would we do if they got leaks?

Also I notice you are using condensate for lube oil cooling. Was this a reliability choice or perhaps another type of choice?

Thirdly, I think it is very encouraging to use fresh water for our sanitary systems, but will you be running these evaporators in port, and if not, where do you get fresh water?

HOLM: Actually we now have a certain number of ships operating coast wise which do not use their settlers. I believe some of the oil companies are running coastal tankers where they are not settling their oil. They take it directly from the tank. Now admittedly on a tanker this is usually in the form of a cross bunker and the problem is less severe, but with a modern welded tank we should not have much trouble with water entrainment into the tanks. Certainly the probability of this is appreciably less than that of getting water into the tank due to ballasting, and we feel we can lick this problem. Between the combination of not ballasting your tanks and using steam atomizing burners and so forth, we think this can be settled without any great trouble.

KAISER: The condensate being used as a coolant for the lube oil system was based on the premise of reducing maintenance. We felt that eliminating all salt water systems, except to the main condenser, that the maintenance would be greatly reduced, so you could in effect say that this is concerned with reliability.

The other question you had was on the evaporator, in port, the evaporator would not be on the line. When coming into port, or steaming up a river, it could be on the line if you are low on fresh water. If you are steaming up the channel or coming into port, we could run the evaporator, as I understand it, correct me Captain if this is in error from the standpoint of the Coast Guard, you can come into polluted waters and still run your evaporator since there is a specification change where you are required to have a thermostatic temperature control.

We have a 10 pound bleed from the low pressure turbine to the evaporator and if this gets down below 75% of full power, we can tap off the high pressure bleed for steam.

HIRSCHKOWITZ: I'm concerned with the health problem here, and you might be forced to a potable in a wash water system.

KAISER: No, the evaporators we have are so designed that they will handle all the needs of the ship, and again as I understand it you can run your evaporator in polluted waters. This is a very recent change.

MCINTOSH (U.S. Coast Guard): This is not a Coast Guard regulation. It is a Public Health Service regulation. Operating temperature of the evaporator would be the determining factor.

KAISER: There is a spec now that requires a temperature control in the system, but either way, we can secure the evaporator or have it running continually.

BOATWRIGHT (Bureau of Ships): First, I'm very pleased to see that you have eliminated a great number of traps. I don't know if you looked into the reliability of those; but although they leak, they don't stop operations. They are disconcerting with respect to fuel rate.

I did have a question about your simultaneous warm up of the boiler and the turbine. Now I was under the impression that usually the boiler took a longer warm up period than did the turbine. Then the second question: what gland seal pressure are you using on the turbine?

KAISER: On the first question, the simultaneous warm up, and this is true that the boiler does require considerably more warm up. We would have the separate starting burner lit off in the boiler to pre-warm the boiler, and since there are no valves in the system, a certain amount of steam will carry over into the turbine. The steam will come over to the valve distribution manifold first which actually acts as a drain, so you won't get the water carry over or any moisture in the line into the turbine directly. It will come into the distribution chamber first where it will then leak off through the air ejector, and incidentally the air ejector does use the superheated steam, but we don't use the high pressure, we do use an orifice between the valve chest and the air ejector. We reduce the pressure with an orifice down to 150 pounds at the air ejector. So consequently the simultaneous warm up is the boiler first, and eventually steam will carry over into the turbine.

The second question on the gland sealing;. the series turbine has 150 pound leak off and this is used for steam atomization on the boiler. The main unit is five to ten pounds.

FRANKEL (M.I.T.): I wonder if you could enlighten me on this series fuel metering and forced draft fan combination. There is a certain discrepancy in the dynamic response of the air flow and the fuel flow during rapid load changes. How do you expect to solve this?

HOLM: There is not a tremendous lag as you might think. Surprisingly enough when we put this on the electronic simulation down at NBTL, we discovered that the lag between the series turbine and the demand is very, very short. It is less than 2 seconds. You must remember that there is very little inertia in this system, less than there is actually in a normal system because you have a motor and a fan and you get a sudden demand for response in a normal system, you have a blast gate which has to open or shut, and all you have to do is attend one trial trip and you see how slow that blast gate moves. The response here is appreciably faster.

FRANKEL: I agree with you with regard to the turbine and the response of the flow, and that this is a low inertia device, but the actual mass rate of flow of the air versus the fuel...

HOLM: You mean you're afraid of the inertia of forcing the air through the boiler?

FRANKEL: No, that right amount of air arrives at the same time as the fuel. . The rate of change of air flow will not be consistent with the rate of fuel oil.

HOLM: This is probably true, but again this is equally true in the modern combustion control system which we are using today. You'll get a puff of smoke, that's all.

FRANKEL: But then you need feed back loops.

HOLM: Well, it's probably not necessary because, as I mentioned, the response is quick enough that we can put up with what little puff of smoke we get.

With the simulation that we made, which of course is not a power plant, but a reasonable facsimile thereof in behavior, we found that you could take the lever — it was very interesting as we had a throttle arrangement on the simulation — and throw it from one to the other or move it at any given speed and the response was quite remarkable. We were very pleased with it. Actually, the only area where we got into any trouble was unanticipated. We had a boiler water level fluctuation, and this was not something that was inherent in the series turbine part of the cycle.

HIRSCHKOWITZ: What was the incentive to go to automatic bilge pumping? We talked about maintenance and perhaps this just could be put under the catalog of maintenance: the man goes around and pumps the bilges. In terms of reliability you might have an added savings here by just using a simple hand valve.

HOLM: Actually this is not an automatic bilge system as such. The point was this: with the present day bilge system, you have a separate line from the engine room to every compartment. It costs a fortune. A much simpler system involves a common main with a valve in each compartment. Now it would at least be theoretically possible to do this with existing equipment by just using reach rods up to the deck. In fact this idea has been proposed. Usually the objection raised is that you will have to wind up paying some member of the engine room force overtime to go out on deck. That is, he would have to go forward on the deck and open valves in compartment after compartment as similar operations are now carried on in tankers. They think nothing of going out on deck to open a valve to pump a compartment. Normally you do it along side of the dock. They don't use this kind of system, but in a merchant cargo ship we feel that this enormous run of pipe that has to be made, one pipe for each compartment is a prohibitively expensive thing, and we can reduce the cost of the ship by doing this.

The fact that by using air operated valves is that you can centralize the control of them with an incidental advantage, it was not the engine itself. We do not feel in the operation of this plant that the crews are supposed to stay in a cubical. You'll notice that on the arrangements that we have there is no cubical. We would like to have the crew listen to what is going on. The ears are a very good censor.

HIRSCHKOWITZ: Would this have a hand valve anyhow in the event of the failure of the air?

HOLM: Well, it is kind of hard to put a hand valve in the forepeak that you can reach from the engine room or wherever you are going to put the hand valve. We are a little leary of this particular item ourselves. In other words what is involved here is that we've got to find a valve which can be buried in the innards of the ship and be reliable. I don't think that is impossible. I think it can be done. There are any number of remotely operated valves available on the market and surely we can find one that will do the job.

AN INTEGRATED MARINE GAS TURBINE POWER PLANT

R. P. TILLSON, General Electric Co.
E. H. MCCALLIG, George G. Sharp, Inc.

I. INTRODUCTION

Seven years have elapsed since tests began on the first wholly gas turbine powered merchant ship. This highly successful test on the GTS "John Sergeant" has been followed by a period of only casual interest in the use of gas turbines for marine propulsion. In the meantime, gas turbine applications have made tremendous strides in the industrial and power generation field. Operating experience of units in these applications has demonstrated a high reliability, an economic advantage, as well as, a versatility unmatched by any other prime mover.

The Maritime Administration in recognition of the gas turbine's potential, has within the past three years, undertaken a broad and well-organized program to evaluate various propulsion plant concepts. This program, involving both steam and gas turbines, is to evaluate the power plant that offers the greatest potential of improving the operating economy of U.S. Merchant ships. Improvement in operating economy is expected to be achieved through low first cost, low maintenance costs, fuel economy and automation.

This paper is a summation of a study conducted by the General Electric Company with George G. Sharp, Inc. as subcontractor, on a gas turbine power plant for a merchant ship. The propulsion gas turbine is based on industrial design technology and the utilization of bunker C fuel.

II. BACKGROUND

The study on which this power plant is based, began in April 1962, and is planned to be completed within the next few weeks. The starting point or base for beginning the study was the design, manufacture and operating experience on 263 industrial type gas turbines which have been installed or shipped for various applications. This progress of applications, Fig. 1, begins in 1949 and shows the application increase through 1962. More than 6.7 million hours of operation have been accumulated including over 1 million hours of operation on residual fuel.

Twenty-nine of these gas turbines have operated over 60,000 hours each and nearly 100 units have operated over 30,000 hours each.

III. APPROACH

The power plant concept described in this paper is the result of an evaluation of various gas turbine cycles and machinery arrangements, including auxiliaries. This evaluation was based on the following objectives:

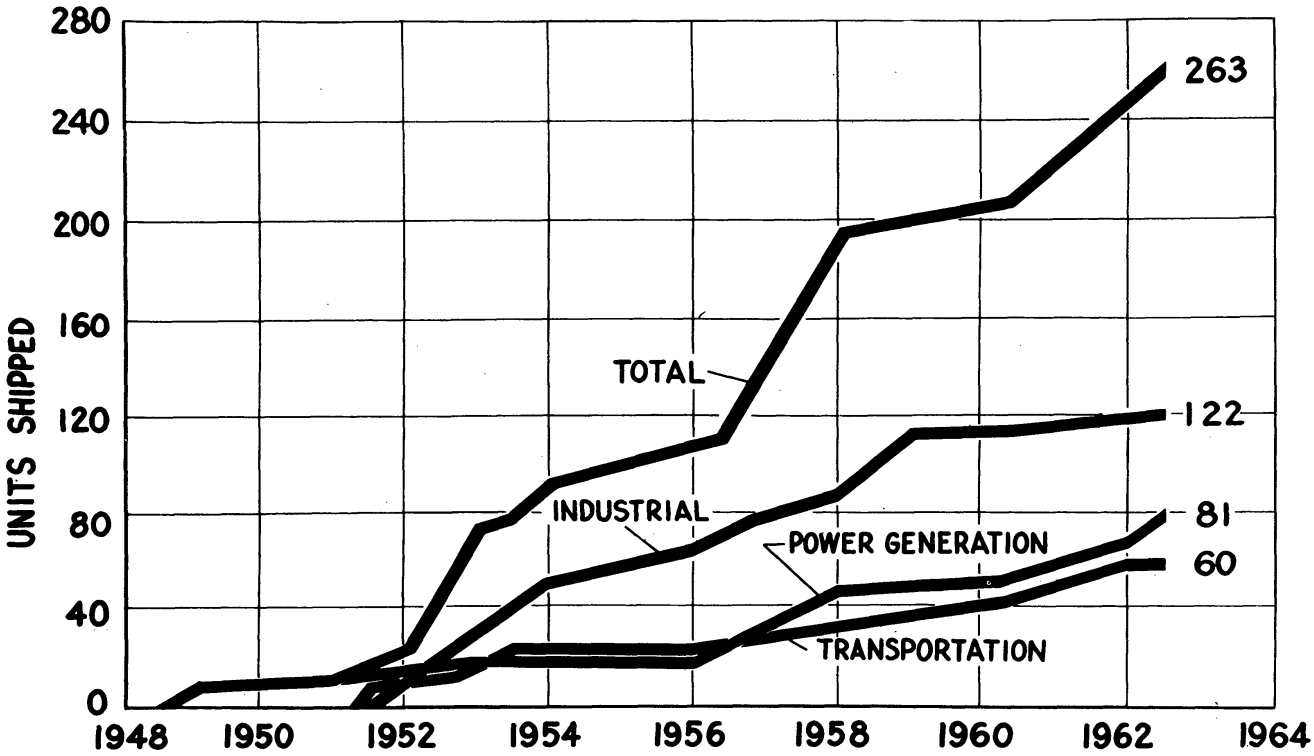


Figure 1. Progress of General Electric Gas Turbines.

1. Develop a gas turbine power plant with minimum yearly operating costs, which includes capital charges, fuel, maintenance, repair and crew costs.
2. Simplify power plant systems and reduce complexity while maintaining high reliability.
3. Select a gas turbine cycle with a growth potential in rating, as well as performance, for future marine applications.
4. Utilize the full range of available bunker C fuels.
5. Develop a design with pre-assembled, packaged systems; accessories and auxiliaries integrated with the main propulsion unit, and having minimum foundations and connections.
6. Provide self-regulating capability sufficient to operate the power plant with two men or less on watch.

Of these objectives, operating costs had the greatest bearing on the selection of the plant to be studied. Capital charges and fuel costs are the major elements of operating costs.

In addition, various types of transmissions were analyzed involving both mechanical and electrical systems. The transmission system selected is considered as offering the best solution to ship maneuvering and reversing although investigations will continue on a scheme which promises to offer a potential for the development of a direct reversing turbine.

Improved fuel economy was achieved by maximum utilization of the heat energy in the exhaust gases. Although this approach will involve some reduction in simplicity, the automatic control features of the plant will be maintained.

IV. DESCRIPTION OF POWER PLANT

A. MAIN PROPULSION UNIT

The main propulsion plant consists of a regenerative-cycle, two-shaft gas turbine driving a controllable and reversible pitch propeller through a double reduction gear. This unit is rated at 20,000 shaft horsepower at a propeller speed of 105 rpm with an inlet air temperature to the compressor of 75°F. An elevation and plan of the main propulsion components is shown in Figs. 2 and 3 respectively.

1. Gas Turbine

The regenerative, two-shaft gas turbine consists of an axial flow compressor, a combustion system, a high pressure turbine and a mechanically independent load turbine and a regenerator.

Air is taken from topside and passes through an inlet silencer located near the compressor inlet hood and is compressed in the 16 stage, axial flow compressor to a pressure ratio of 6.6:1. The air then passes to the regenerator where heat is added from the exhaust gases and then flows to combustion headers located on each side of the unit centerline. Each combustion header supplies heated compressed air to four (4) combustion chambers (8 total) where fuel is added and the mixture burned. The resulting high temperature gases discharge through a 90 degree angle to the first stage nozzle of the turbine at a temperature of 1500°F. The gas passes through the single stage, high pressure turbine which drives the axial flow compressor and then through the variable angle second stage nozzle of the load turbine. This nozzle controls compressor speed and hence air flow. The gases then expand through the single stage load turbine which is connected to the main reduction gear by means of a flexible coupling. After passing through the load turbine the gases pass through the exhaust hood, through the gas side of the regenerator located directly above the exhaust hood, to the heat recovery boiler and then to the stack.

Components, with the exception of the combustion system, are combined into a single assembly and mounted on a short base. These components are split on the horizontal joint for ease of accessibility for maintenance. The base is supported from the ship's structure by three support points.

Compressor:

A partial section of the gas turbine, Fig. 4 shows the 16 stage, axial flow compressor of separate wheels held together by through bolts. Concentricity is assured by the use of rabbet fits near the wheel centers. This is of extreme importance not only for balance reasons but to maintain the proper tip clearances which are essential to the high efficiency of the compressor.

Compressor blades are fixed to the wheels with an axial dovetail arrangement. Stator blades are dovetailed into segmented blade rings which fit into circular grooves in the base of the casings. These ring assemblies are easily rotated out of the casing for maintenance purposes.

The inlet casing, cast steel compressor casing and discharge casing are all horizontally split without rabbets on the vertical joints. This permits any sequence of lifting casings for inspection or work in the compressor area.

Extraction air is taken off the compressor for use in sealing and cooling various parts of the machine. The extraction rings extend entirely around the casing to a flange and are designed to extract the proper amount of air at minimum head loss.

Combustion System:

Air, the working fluid of the gas turbine is heated by the burning of bunker C fuel in multiple combustion chambers. Bunker C fuel is injected into each chamber

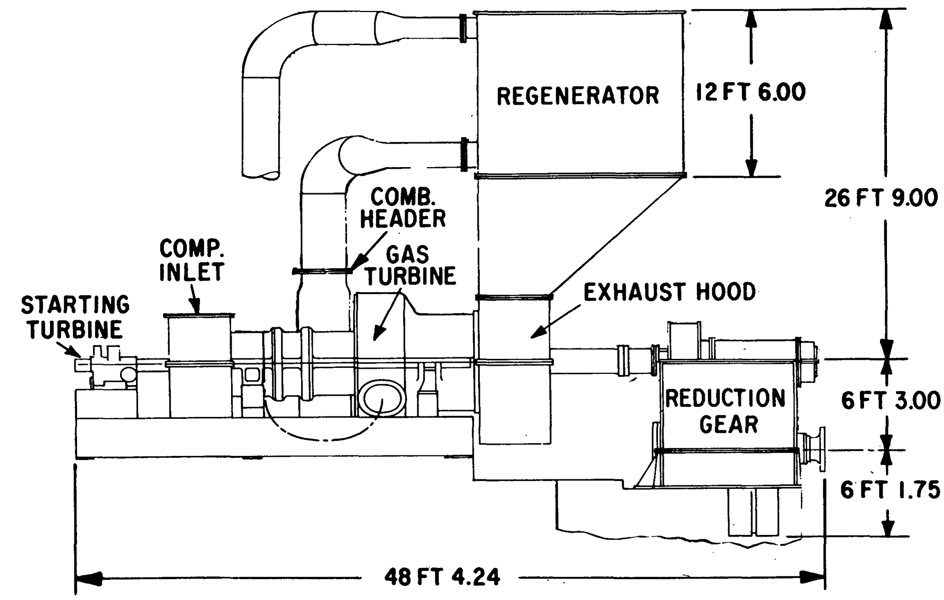


Figure 2. Elevation - Gas Turbine, Regenerator and Reduction Gear.

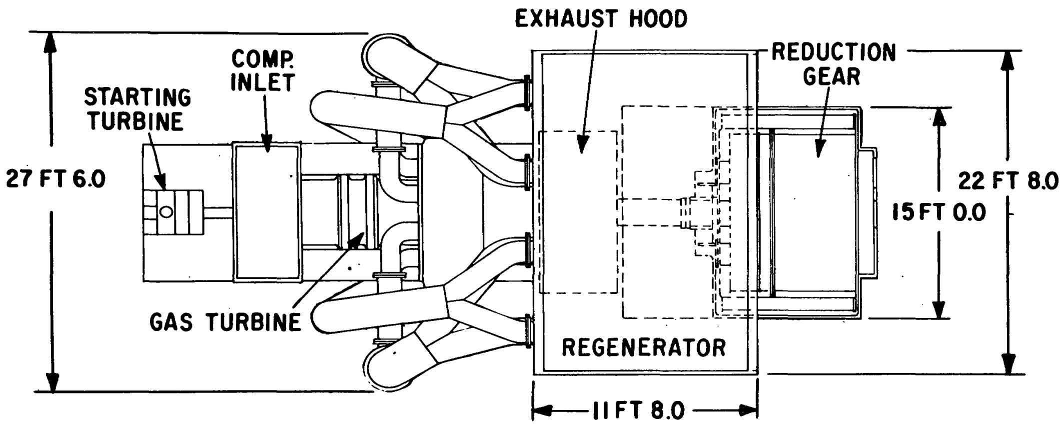


Figure 3. Plan - Gas Turbine, Regenerator and Reduction Gear.

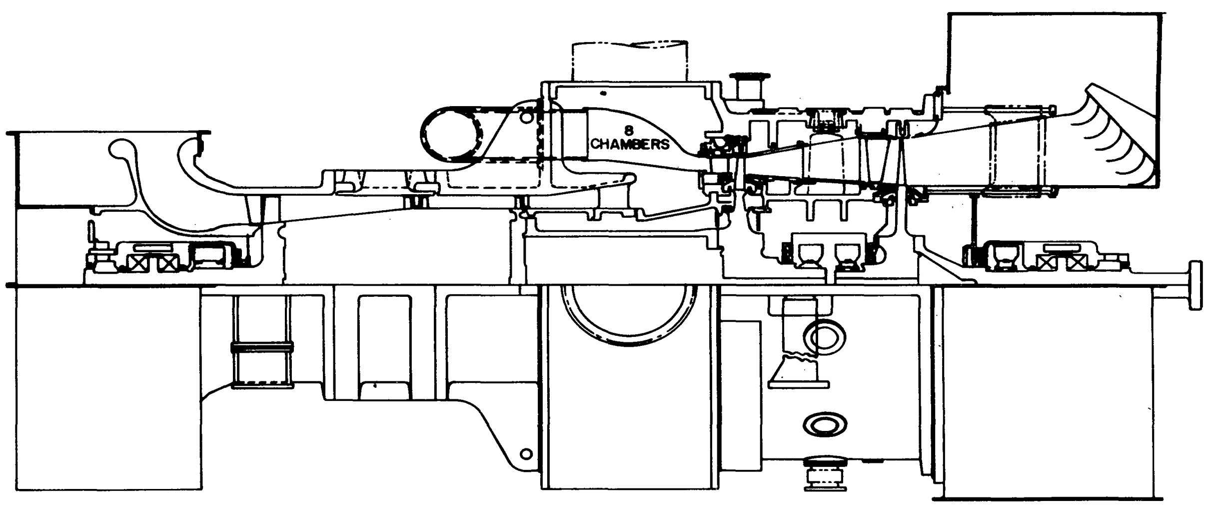


Figure 4. Section - Gas Turbine.

through the fuel nozzle and is mixed with part of the total air flow for combustion, and the remaining air mixes with the resulting gases and also cools the liner.

Initial ignition of fuel is accomplished electrically by spark plugs energized in two or more combustion chambers. Flame is propagated to the other chambers by cross fire tubes. After ignition, the combustion is self-sustaining and the spark plugs are automatically retracted and de-energized.

Flame detectors are used to show that ignition has occurred and is sustained. This signal is indicated by lights at the control panel.

Dual fluid nozzles are used and atomization of the bunker C is accomplished within the nozzles by means of high pressure steam. Steam provides good atomization of residual fuels with efficient combustion and a good temperature pattern over long periods of time. Steam atomization lengthens periods of operation between inspection and cleaning.

Turbines:

The first, or high pressure, stage of the turbine drives the axial flow compressor while the second, or low pressure stage, drives the propeller. These two turbine stages are not connected mechanically and operate at different speeds.

The first-stage turbine rotor consists of a turbine wheel and stub shaft machined from a high strength chrome steel forging and a distance piece which connects the turbine to the compressor. The first-stage buckets incorporate a long shank design between the dovetail and vane section. Pins placed between adjacent buckets at the shank section are used to reduce vibrating stresses and prevent leakage losses between bucket shanks. The long shank design reduces the operating temperatures in the critical dovetail region which increases bucket life and reliability.

The second-stage turbine consists of the turbine wheel and its stub shaft which is connected by a flexible coupling to the main reduction gear. The second-stage wheel is machined from a solid forging of high strength chrome steel alloy.

All turbine buckets are made from forgings. Buckets on both first- and second-stage wheels are tied together with pins or wires at about 2/3 the radial height of the vane section. These pins reduce the vibrating stresses in individual buckets since adjacent buckets will share any loading due to unusual stimuli.

The first-stage nozzle is made up of segments consisting of one or more partitions cast with integral side walls. These segments are held and located by a separate retaining ring which is mounted so as to be centered in the turbine shell but free to expand radially. Joints between segments are sealed mechanically and no welding will be required in the assembly of the nozzle. Nozzle material will be determined from the results of a proposed materials investigation which will be aimed at a turbine inlet temperature of 1500°F on bunker C fuel treated to specification.

The second-stage variable angle nozzles consist of either cast or forged partitions welded to operating stems which extend radially through bushings in the turbine shell. The position of these nozzles is changed by means of levers linked to an operating ring, rotated by means of a hydraulic cylinder. This device positions all the nozzles simultaneously and sets the required nozzle angle to satisfy the operating requirements. These nozzle stems are hollow and water cooled to assure long life and reliable operation of the stem in the shell bushing. A fresh water system is used to provide cooling water to the second-stage nozzle stems and the turbine support legs.

The second stage or load turbine exhausts through an annular diffuser inside the exhaust hood. This hood is arranged to exhaust upward through turning vanes. The structure is fabricated from plate, structural shapes and castings, to make a light structure with good recovery of the last stage leaving energy and minimum back pressure.

In addition, cooling air, extracted from the compressor is used to cool the turbine shell. The air flows between the cast shell and an outer jacket. A portion of this air is used to cool the struts in the exhaust hood that support the aft load turbine bearing and the load turbine thrust bearing. This cooling air is bled from appropriate stages of the compressor.

Bearings:

The two elliptical type and two tilting pad type journal bearings in the unit are designed to improve the stability of the shaft at high speeds. Thrust bearings are located on each shaft and are of the Kingsbury type which equally divide the load on all bearing shoes. Labyrinth packing oil deflectors confine the lubricating oil to the bearing region. The oil deflectors are designed with double seals with an annular space between. A positive pressure of sealing air is maintained in this annular space to contain the lube oil and vapor. This also prevents hot cycle air from the unit from coming into contact with the oil and bearings.

Starting System:

The gas turbine will be started by means of a small single-stage, steam turbine. Steam is supplied by a heat recovery boiler which is heated for the starting cycle by gas from a separate hot-gas generator. A motor operated valve opens so that additional steam is passed to provide sufficient "break-away" torque when the unit is started. After break-away, this valve closes and the unit operates on a single valve throttling control from the governor. After the gas turbine unit reaches self-sustaining speed, steam not required by other services is passed through the starting turbine to develop supplemental power to the compressor. This improves the over-all ship's fuel rate.

The starting turbine is directly connected to the gas turbine compressor shaft by a flexible coupling and runs at compressor speed.

Regenerator:

The regenerator, mounted vertically directly above the exhaust hood of the turbine is a four-section, plate-fin type. Turbine exhaust gases are ducted to the regenerator at the bottom and flow upward through banks of rectangular channels. Compressed air enters at the top, passes into a manifold and then into air passages between gas channels. The air travels downward, counterflow to the direction of the exhaust gas and takes heat from the hot gas. The heated air then enters a manifold at the bottom and is ducted to the combustion headers located on each side of the turbine.

Individual regenerator channel banks are copper brazed and constructed of carbon steel. Reinforced latticework and other components under stress are fabricated of carbon-molybdenum steel.

Vertical installation facilitates occasional cleaning of the regenerator by water washing and reliability is improved by the selection of a design which has accumulated considerable operating experience.

2. Transmission System

The transmission system of the integrated gas turbine power plant consists of a main reduction gear and a controllable and reversible pitch propeller.

Main Reduction Gear:

The main reduction gear will be an articulated, locked train, double reduction, double helical type consisting of rotating elements of one first-reduction pinion, two first-reduction gears and two second-reduction pinions and one main gear. At 20,000 SHP the high speed pinion will have a speed of 3600 rpm, the low speed pinion 575 rpm, and a propeller speed of 105 rpm at design pitch setting. The "K" factors at design condition will be 160 and 105 on the high and low speed elements, respectively.

The reduction gear assembly will be an integrated package consisting also of a pinion for emergency propulsion, a complete lube system for the entire propulsion set, a turning gear, thrust bearing and shaft brake (optional), designed to stop the propeller shaft from turning at idle speed setting of the main turbine.

A provision for emergency propulsion will consist of installing a coupling between the gear of the ship's service steam turbine-generator set, located adjacent to the main gear, and a pinion on a shaft extension of the high speed. In an emergency this arrangement could drive the main gear at reduced speed to obtain a ship's speed of approximately 7 knots.

The lube system consists of a tank built into the base of the gear casing, a shaft driven lube pump, an a-c electric motor driven pump, an emergency d-c motor driven pump, and lube oil coolers.

Propeller:

A four bladed hydraulically operated controllable pitch propeller, 21 feet in diameter and having a full pitch of about 22.2 feet is proposed. This propeller will drive the reference ship selected for the study by MARAD (PD 108), at a speed of just over 20.85 knots at design operating conditions.

Propeller pitch will be adjusted by means of a hydraulic servo-motor system. Control of the pitch will be accomplished by the use of a pneumatic control system that matches pitch to gas turbine load. Normal control of the power plant will be by a single lever which controls both turbine power and propeller pitch. A separate lever will be provided in order to control propeller pitch when desired.

B. SELECTION OF AUXILIARIES

1. Steam Generating System

The gas turbine, in common with other forms of the internal combustion engine, exhausts at a temperature which is relatively high compared to the ambient temperature. In distinction from other internal combustion engines the gas turbine requires a very high air/fuel ratio. The combination of large air flow and high exhaust temperature results in a very large quantity of energy being discharged to the atmosphere. Fuel economy requires that a portion of this otherwise rejected heat be recovered. The regenerator reduces the temperature of the exhaust gas. However, in order to further improve fuel economy, a heat recovery boiler is added to the system. For example, the large amount of heat which can be recovered from the exhaust gas by a boiler can supply all auxiliary power and ship's service requirements. To avoid the use of several components and controls normally associated with the addition of a steam generating system to a gas turbine power plant, several features have been employed. These features render the system completely self-regulating with a minimum of automatic controls thereby reducing cost, maintenance and manning requirements and increasing reliability.

A heat recovery boiler, of the extended-surface type, is located directly above the regenerator. Saturated steam will be generated at 200 psi. The temperature of the turbine exhaust leaving the regenerator, about 600 F, does not justify the additional cost and complication of a superheater. The very large air flow/steam flow ratio makes it impossible for an economizer to lower the stack gas temperature more than a few degrees. This often troublesome component is therefore omitted and the 200 F feed water is introduced directly into the drum. The minimum metal temperature is thereby maintained at steam saturation temperature of 388 F, which is above the minimum temperature required to avoid stack gas corrosion.

The exhaust gas temperature is low by conventional boiler standards and precludes boiler damage resulting from low water level so a single element feed regulator responsive to drum level only is sufficient. To further simplify operation and to make the steam plant self-regulating it is planned to control steam pressure by using all the steam that can be generated. All of the gas exhausting from the main turbine flows across the steam generating surfaces of the boiler. This scheme eliminates the requirement for large bypasses around the heat recovery boiler and the automatic damper control associated with such bypasses.

The steam generated in the heat recovery boiler normally supplies a steam turbine driven generator, steam for fuel atomization and steam for fuel oil heating. As mentioned earlier, steam pressure is controlled by using all the steam that can be generated. This is accomplished without dumping excess steam to a condenser. The fuel atomizing nozzles and turbine generator will receive as much steam as they demand, and the excess, controlled by a back-pressure valve in the supply, will expand in the starting turbine producing useful shaft horsepower. With this arrangement the starting steam turbine remains coupled to the gas turbine compressor shaft at all times eliminating the need for a clutch on the main unit shaft. A bypass around the back-pressure valve provides a minimum flow to the starting turbine at all times to provide cooling. Thus the entire steam generating system is inherently self-regulating and is normally controlled by only two elements, the feed regulator and a back-pressure controller.

The boiler will have sufficient capacity to supply all the steam required at full power for fuel atomizing, the fuel oil heaters, and for generating 750 KW of electrical power. Exhaust steam supplies the distilling plants and all ship's service heating requirements.

Atomization and starting require that steam be available with the main unit secured. Rather than compromise the reliability of the heat recovery boiler by oil-firing or increase the complexity of the plant by adding a separate boiler the heat recovery boiler is provided with a source of hot gas other than the gas turbine exhaust. For this purpose a hot gas generator consisting of a combustion chamber with a vortex burner is used. This unit has had extensive development and more than ten years of successful industrial use. The burner and combustion chamber assembly has a very high heat release rate of approximately one million BTU per hour per cubic foot of combustion chamber volume. The combustion gases are diluted with a sufficient quantity of free air to reduce the mixed gas temperature to the normal gas turbine exhaust temperature so that the heat recovery boiler can remain a relatively low temperature gas-to-boiling-water heat exchanger without furnace or refractory even though it can, in effect, be oil fired. The burner is completely automatic, including ignition, and is equipped with suitable flame failure safeguards. Since instant ignition is possible, even with a cold burner, the burner is arranged to fire automatically on low steam pressure and to shut off on restoration of the set steam pressure thus maintaining a steam supply during periods of maneuvering.

A maximum utilization of self-regulating features provides the effect of automation and increases reliability through freedom from operator errors without introducing complex automatic control devices.

2. Ship's Service T-G Sets

Electrical power is provided by two 750 KW generators. A steam turbine driven generator normally provides electrical power at sea. The turbine operates noncondensing with 200 psi saturated steam at the throttle. With steam furnished by exhaust heat recovery the greater thermal efficiency of condensing operation would not, during the life of the ship, pay off the greater first cost of a multi-stage condensing turbine with its condenser, air ejector, condensate pump and gland sealing provisions.

Electrical power in port is supplied by a 750 KW gas turbine generator set mounted on the main deck so that the engine room may be completely secured. No steam is generated in port, and all engine room components and piping are available for preventive maintenance or repair thus increasing the reliability at sea of all engine room machinery. Also, installing the gas turbine generator on the main deck may enable it to satisfy emergency generator requirements. Duplicate controls and instrumentation allow the unit to be started and controlled from the engine room if desired. The gas turbine generator is started by a 24 volt electric starter and would normally be started upon entering port prior to maneuvering and secured upon clearing port after steady steam conditions had been established by the main turbine exhaust.

3. Distillation Plant

Both the starting turbine and the steam turbo-generator exhaust at a back pressure of 5 psi. This auxiliary exhaust supplies two (2) - 20,000 GPD, two-stage, flash type distilling plants and miscellaneous heating services. The distilling plant capacity, which is large for a marine power plant of this rating, is determined by the need to replace the fresh water losses resulting from steam atomization. The atomizing steam makeup is slightly more than 20,000 GPD; however, the saving in fuel cost resulting from steam atomization as compared with compressed air atomization as generally used with industrial gas turbines is more than $20,000 yearly. This saving will soon pay for the increased distilling plant capacity. The total capacity required is about 23,000 GPD for makeup, potable and wash water. With additional capacity available it will be possible to use a fresh water sanitary system thus simplifying the ship's plumbing systems. The total installed capacity of 40,000 GPD is divided between two units for reliability.

4. Fuel Treating System

In order to utilize a wide range of bunker C fuels, special handling and treating equipment is required in order to make the fuel suitable to burn at a high temperature in a gas turbine. This equipment must not only heat the fuel to a suitable viscosity for pumping and atomizing, but also clean and treat the fuel to inhibit the corrosive attack of metal salts such as sodium and vanadium. Present available technology and maintenance goals limits the allowable firing temperature to 1450°F when using bunker C fuel but a firing temperature of 1500°F is expected to be attained by some development work on fuel treating and in turbine materials to attain maintenance goals. Also, the treating plant proposed which consists of separate systems for washing the fuel, injecting an additive and analyzing the results to determine proper specification will be automated to the maximum practical extent. To accomplish this goal it will be necessary to develop a means of getting the additive from the storage area to the treating plant and to provide in-line sampling and indicating equipment.

Another area which requires investigation is in the type of additive used. Fuel treating systems in stationary power plants use magnesium sulphate as an additive to inhibit the corrosive action of vanadium. The higher the vanadium content of the fuel, the greater the amount of additive needed. The space for additive on ships making long voyages may present a problem when burning high vanadium fuels. Therefore, a development work is proposed to develop new additives which will require less storage area.

5. Miscellaneous

Auxiliary exhaust is used for heating water for fuel oil bunker and cargo oil heating, forced hot water heating of quarters as part of the air conditioning system, and hot potable water. During port operation when the engine room is secured, a heating coil in the exhaust duct of the gas turbine-generator set supplies hot water for quarters heating and hot potable water. Boiler pressure steam for fuel oil heating and butterworth heating, if required, is also available. Excess auxiliary exhaust is automatically unloaded to a small, sea water cooled, atmospheric condenser. A small, motor driven centrifugal feed pump returns the condensate to the boiler.

C. ARRANGEMENT OF MACHINERY

The machinery arrangement has been planned to provide generous accessibility to all components for maintenance and operation without wastage of space. Pre-packaging of components by the manufacturer has been planned to minimize foundation construction, alignment and installation work by the shipyard. It is considered axiomatic that assembly and alignment can be performed more accurately, more reliably, and more cheaply in the shop than aboard the ship.

The main turbine and other major components are shipped mounted on rigid box-shaped sub-bases and are lifted into the ship already assembled and set upon a three-point support type of foundation. The use of a three-point support insures that each support point carries its design load and thus contributes to reliability by helping to maintain alignment between components.

Since the components of the main unit are arranged in tandem it is possible to

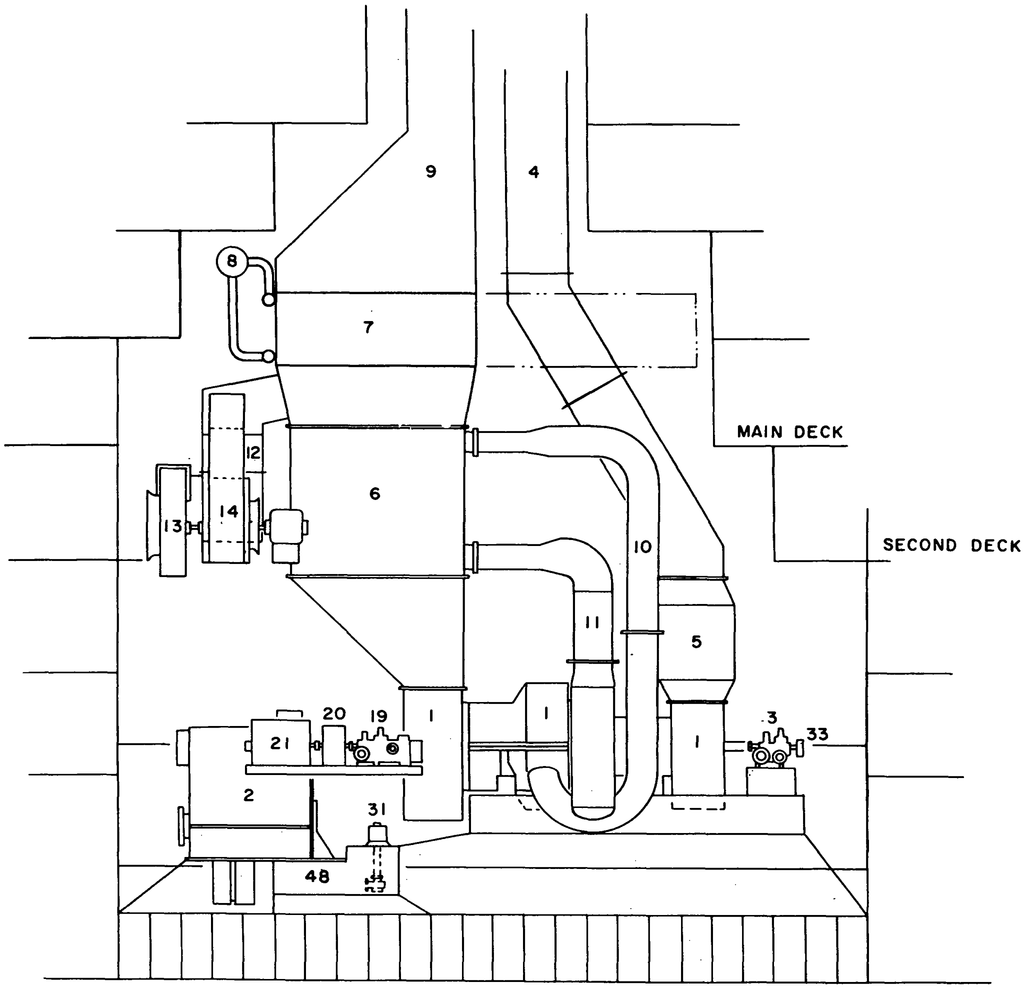


Figure 5. Machinery Arrangement - Elevation.

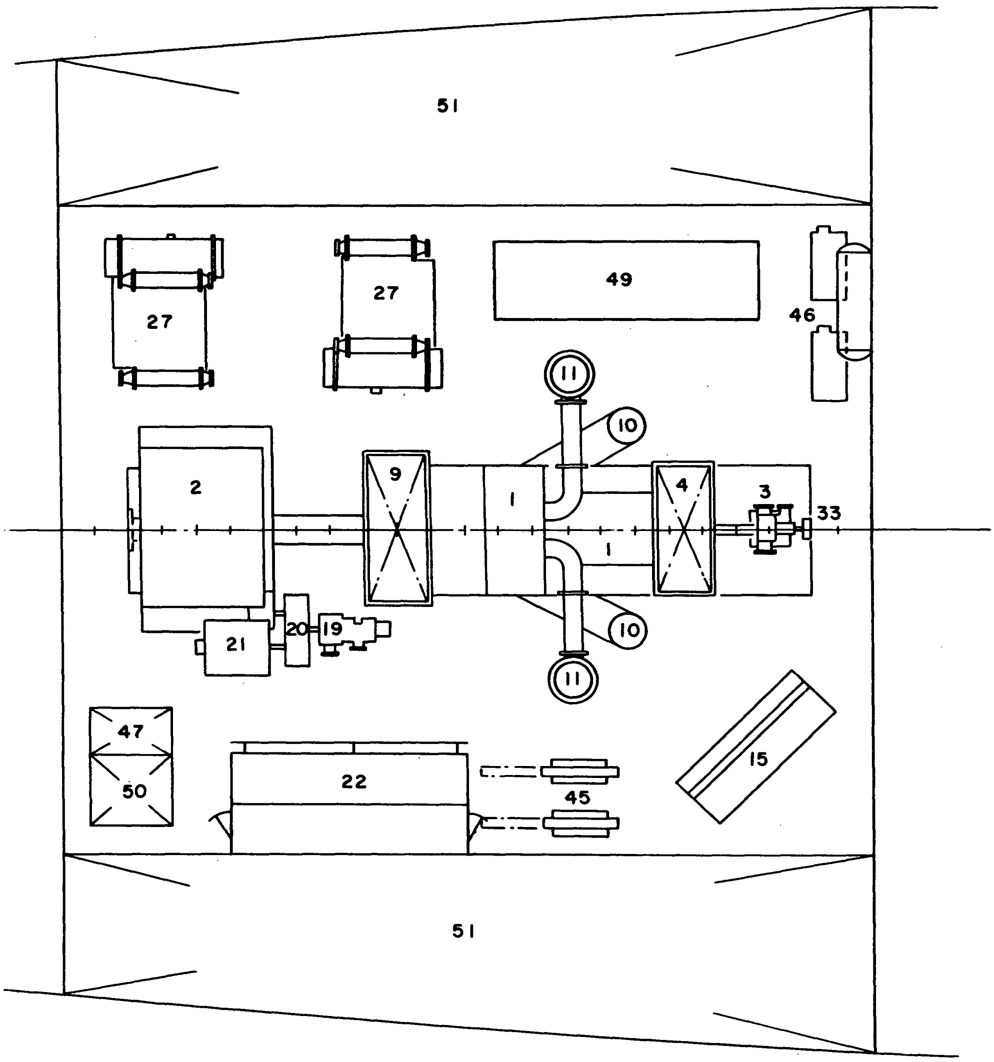


Figure 6. Machinery Arrangement - Operating Level.

arrange the auxiliary machinery without using the full breadth of the ship and the outboard space in way of the engine room may be utilized for fuel oil deep tanks. This arrangement provides substantial flexibility in the location of the machinery space.

The overall length of the main unit including reduction gear is forty-nine feet and easily fits an engine room length of 60 feet. The total volume of the engine room including casing is 140,000 ft.^{3} compared to about 155,000 ft.^{3} for a comparable steam turbine power plant.

An elevation of the machinery arrangement is shown on Fig. 5 and plan views at different levels are shown on Figs. 6 through 8.

MACHINERY LIST

INTEGRATED GAS TURBINE POWER PLANT

(Refer to Figs. 5, 6, 7, and 8)

1. Main Turbine
2. Main Reduction Gear
3. Steam Starting Turbine
4. Inlet Duct

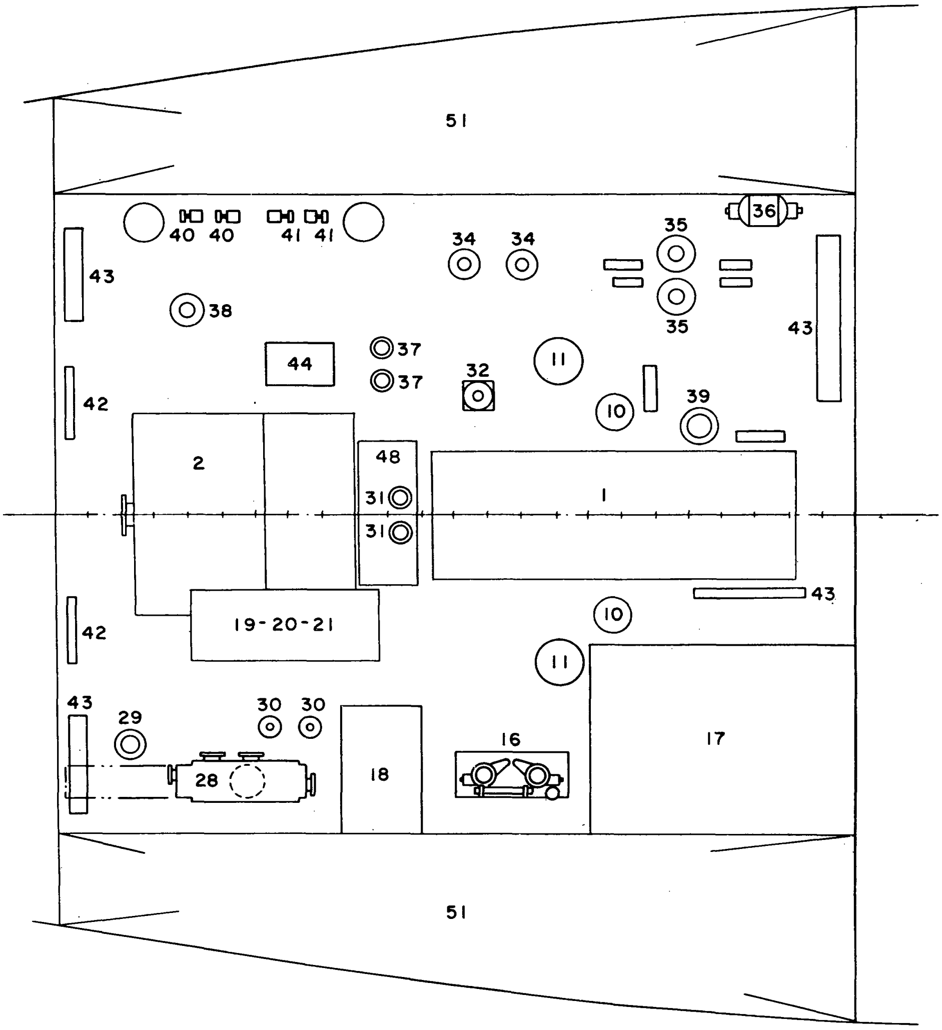


Figure 7. Machinery Arrangement - "B" Floor Level.

5. Inlet Silencer
6. Regenerator
7. Heat Recovery Boiler
8. Steam Drum
9. Exhaust Duct
10. Air Duct-Compressor Discharge to Regenerator Inlet
11. Air Duct-Regenerator Outlet to Combustor
12. Vortex Oil Burner
13. Combustion Air Blower
14. Diluent Air Blower
15. Main Operating Console
16. Fuel Washing Apparatus
17. (Space For) Fuel Additive Apparatus
18. (Space For) Fuel Analysis Equipment
19. Steam Turbine for Ship's Service Generator Drive Or Emergency Propulsion
20. Double Output Reduction Gear
21. Ship's Service Generator
22. Main Switchboard
23. Gas Turbine Generator
24. Starting and Control Console for Gas Turbine Generator
25. Emergency Switchboard and Gas Turbine Generator Distribution Panel

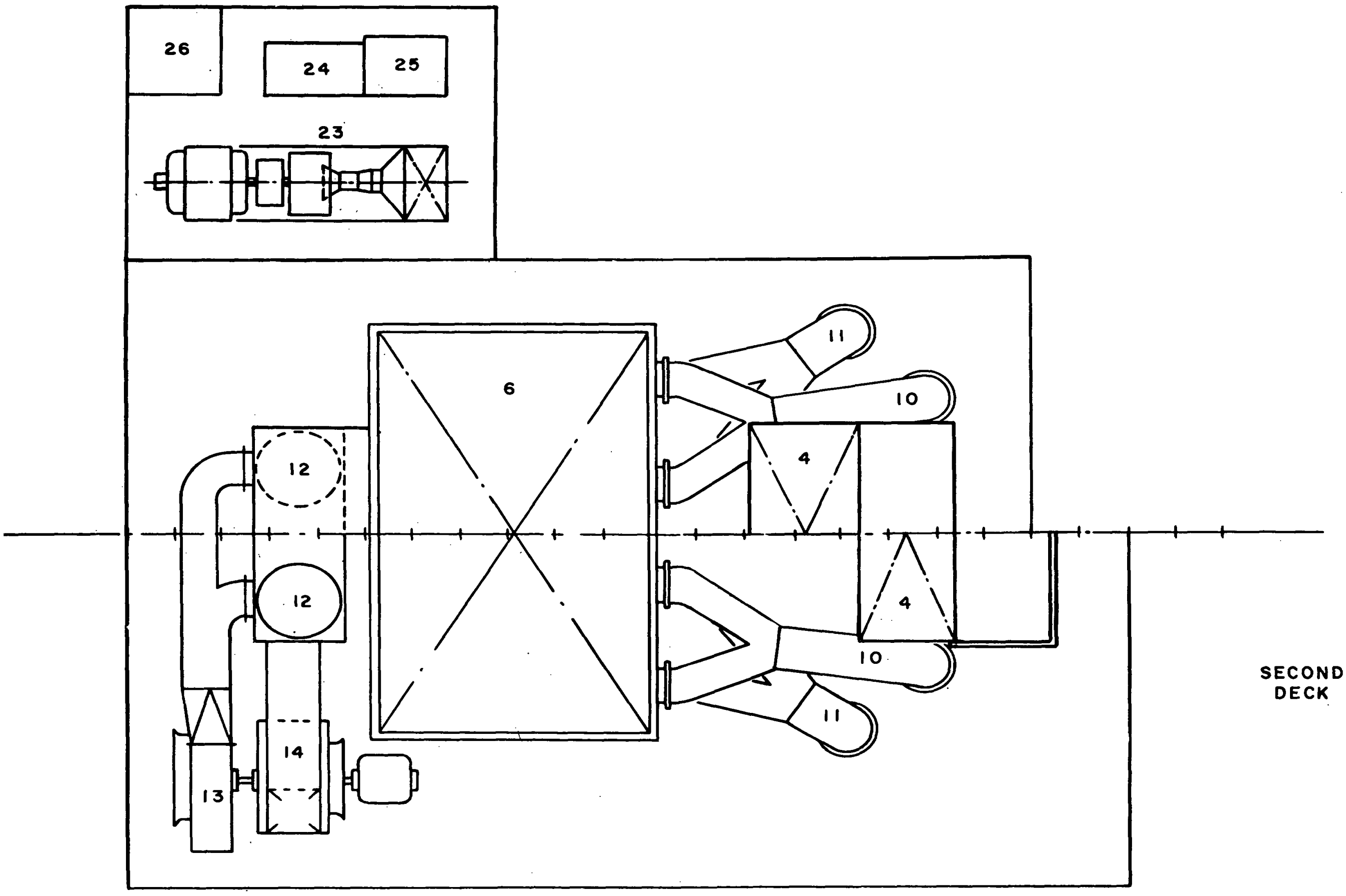


Figure 8. Machinery Arrangement - Second and Main Decks.

26. Battery Locker
27. Distilling Plant
28. Atmospheric Condenser
29. Condenser Circulating Pump
30. Boiler Feed Pump
31. Lubricating Oil Pump
32. Fuel Oil Transfer Pump
33. Fuel Oil Service Pump
34. Salt Water Service Pump
35. Bilge and Ballast Pumps and Manifolds
36. Priming Pumps and Vacuum Tank
37. Lube Oil Purifier
38. Fire Pump
39. General Service Pumps and Manifolds
40. Sanitary Pumps and Compression Tank
41. Fresh Water Pumps and Compression Tank
42. Bilge Manifold
43. Fuel Oil Transfer Manifolds
44. Lubricating Oil Purifier and Workbench
45. Air Conditioning and Refrigeration Machinery
46. Ship's Service and Control Air Compressors and Receiver
47. Lubricating Oil Settling Tank
48. Lubricating Oil Sump Tank
49. Cargo Dehumidification Unit
50. Lubricating Oil Storage Tank
51. Fuel Oil Tanks

D. WEIGHT COMPARISON

The following table compares the weight of the gas turbine plant described in this paper with conventional steam machinery of equivalent rating. Both plants are considered as installed amidships in a dry cargo vessel of about 20,000 tons displacement and 525 feet in length. All weights are in long tons. These weights do not include items which have essentially the same weight in either plant.

Item	Gas Turbine	Conventional Steam Turbine
Main Turbine	65.7	41.9
Regenerator	112.4	----
Reduction Gear	67.0	117.3
Main Cond. AE, Circ. Pump	----	37.8
Shafting, Brgs., Prop. and Brake	233.4	178.5
Steam Turb. and Emerg. Diesel Gens.	15.0	40.5
Gas Turbine Gen.	5.4	----
Piping and Misc. Heat Exch.	25.5	100.0
Boilers incld. FD and FO Sys.	39.7	296.0
Fuel Treatment Sys.	5.5	----
Uptakes, Ducts, Stack	30.0	22.8
Distilling Plants	25.0	16.9
Ladders and Gratings	25.3	44.6
Machy. Space Vent	8.5	11.6
Liquids in Piping and Machy.	60.1	94.2
Total	718.5	1002.1

Difference 283.6 LT

V. CONTROL AND OPERATION

The control concept for the integrated gas turbine power plant design will involve:

a. Remote start-up, operation, control and shutdown from a central control station located close to the power plant.

b. Minimization of manual operation consistent with ship's safety, reliability and maintainability.

c. Consideration for ultimate transfer of the control function to the bridge.

The gas turbine and propeller pitch control will be integrated so that speed of the ship can be controlled by means of a single lever. Two maneuvering or control stations will be provided — one located in the engine room at the central control station, and the other on the bridge. A block schematic functional diagram of the control, Fig. 9 indicates that an operator may select either of the following set-ups.

1. Single-lever operation from the bridge.
2. Single-lever operation from the engine room.
3. Two-lever operation from the engine room — one lever for speed adjustment; the other for propeller pitch adjustments.
4. Direct manual from the engine room — adjustment made directly at the gas turbine and at the propeller pitch control servo system.

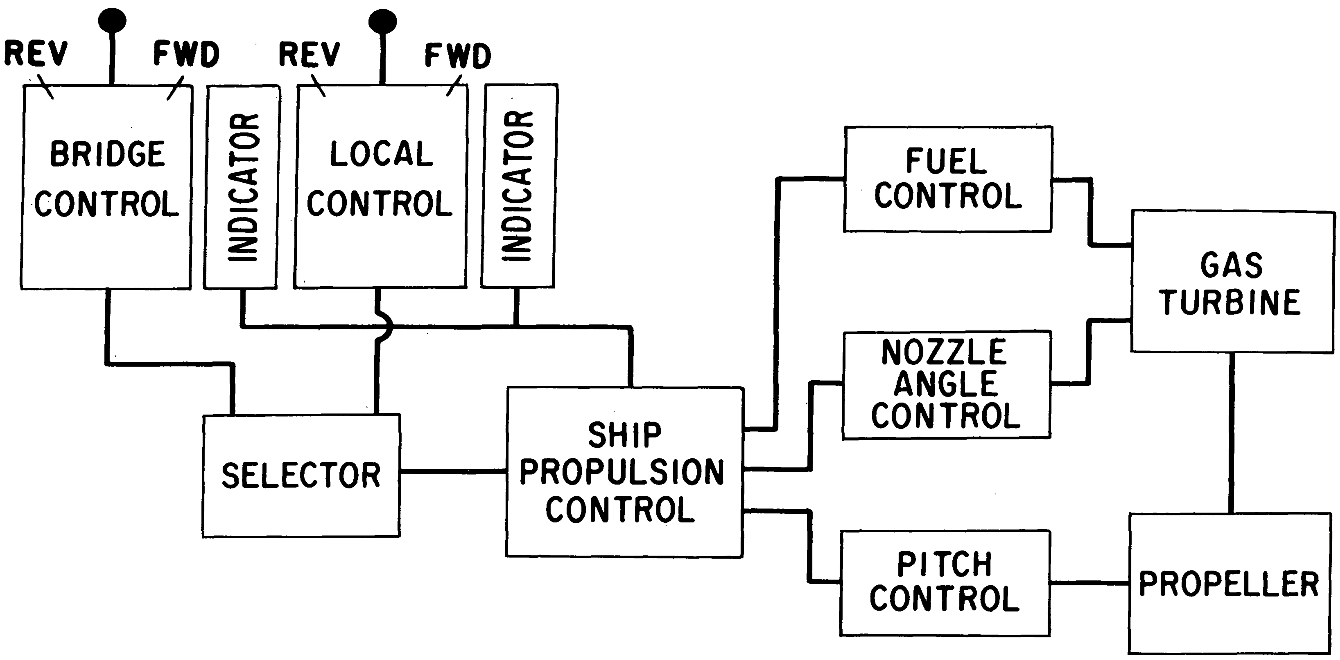


Figure 9. Schematic Block Functional Diagram - Propulsion Control.

Once the starting indication is given to the gas turbine all functions up to idle gas turbine speed will be performed automatically. Normal operation of the power plant will then be accomplished by the single lever control to set the ship's speed and direction. All of the adjustment required to insure optimum operation of the unit and protect against overspeed and over temperature will be automatic.

Complete details of the control are not included in this summary. However, one feature that warrants mentioning is the use of variable area nozzles between the high and low pressure turbines. This adjustable nozzle will provide one of the primary control functions for the system. This variable nozzle will permit the control to compensate for variations in the air conditions at the compressor inlet and thereby optimize the gas turbine components over a wide range of ambient conditions. Also, the nozzle will maintain the maximum allowable firing temperature over a wide range of power and will improve full and part load fuel rates.

The central control console will have instrumentation, read both directly and by demand, as well as monitoring devices and alarms so that an operator can determine the operating condition of all plant components at any time. The eventual location of the main control console on the bridge is not recommended since this would require three more engineering watch standers to monitor the operation on the bridge.

Two manning schedules are proposed; one for operation of the power plant from the engine room only, and a schedule for operation from the propeller control console on the bridge. These schedules result in an engine room crew of 11 and 8 men respectively, as tabulated below:

Position	No. Men Bridge Control	No. Men Engine Room Control
Chief Engineer	1	1
1st Asst. Engineer	1	1
2nd Asst. Engineer	1	1
3rd Asst. Engineer	1	1
Oiler	0	3
Utility Man	1	1
Electrician	1	1
Wiper	2	2
Total	8	11

VI. PERFORMANCE

The over-all ship's fuel rate at the design conditions of 20,000 SHP, at a propeller speed of 105 rpm and with an inlet temperature to the compressor of 75 F will be 0.510LBS/SHP-HR. This fuel rate will be a guaranteed figure; however, an average fuel rate of several power plants would be expected to be equal to or slightly less than 0.500LBS/SHP-HR. These figures are based on a 1500 F turbine inlet temperature.

The initial design of the gas turbine will be based on eventually operating at a turbine inlet temperature of 1550 F. This can be accomplished by modifications in the hot gas path components after further improvements have been made in the fuel treatment for bunker C and improved materials are available.

When operation at 1550 F turbine inlet temperature can be achieved, the performance will improve by approximately 8 percent in output capability and by 4 percent in thermal efficiency.

A heat balance near design conditions and based on initial computer data is shown on Fig. 10. The computer results on part load performance are expected to be available by the time this paper is presented.

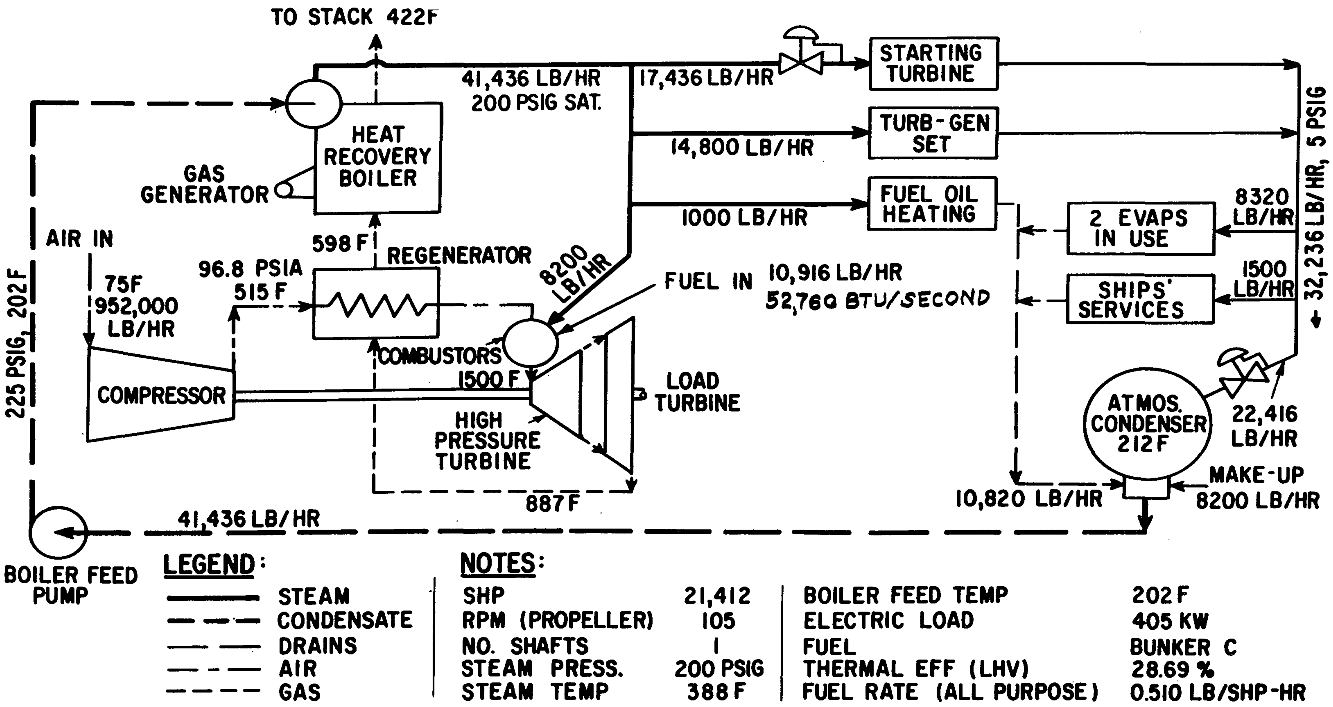


Figure 10. Heat Balance of Integrated Gas Turbine Power Plant.

VII. RELIABILITY CONSIDERATIONS

One of the basic objectives of the design study is to develop an integrated gas turbine power plant for marine propulsion with a high degree of reliability. Several factors contribute to improved reliability of this design concept — namely,

1. Simplicity through a reduced number of components and systems as compared to a steam turbine power plant. In some cases simplicity has been sacrificed to achieve a better plant performance.

2. Self-regulation of various systems eliminates operator error without elaborate automatic controls. Minimum control functions are involved when self-regulation is not possible.

3. Design of the systems involves equipment well proven in actual operating experience. This includes:

   a. a highly developed regenerative, two-shaft gas turbine cycle similar to 86 units in service.

   b. a regenerator of a plate-fin design with over 20 units of this type in service.

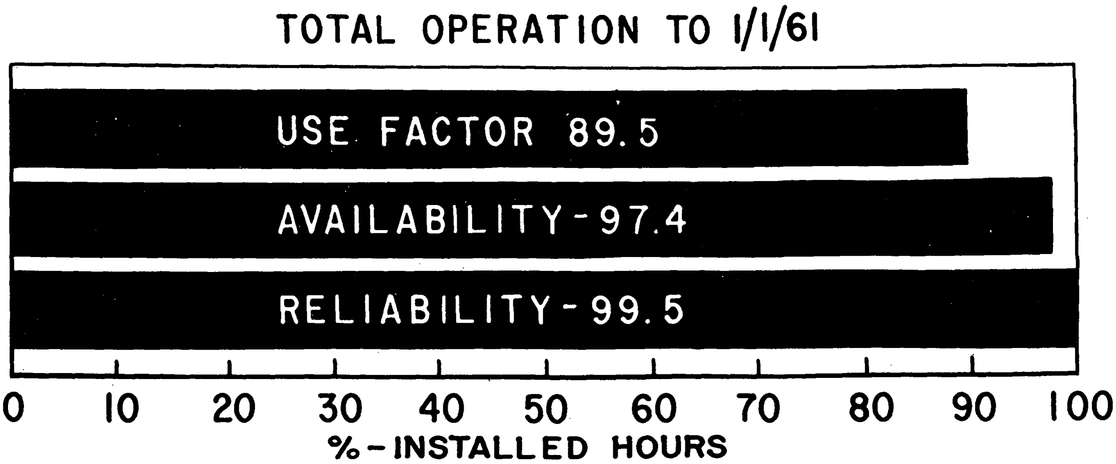
   c. marine reduction gear design based on operating experience on hundreds of main propulsion gears in all types of marine and navy service.

   d. three point support of the gas turbine base to maintain alignment.

   e. packaged approach of components and systems.

   f. shutdown of engine room systems when in port which provides more frequent opportunities to perform preventive maintenance.

Even though no failure rate of specific components is available, a mark of achievement of the industrial type gas turbine is shown by Fig. 11. This is an availability-reliability record achieved by 68 gas turbines which operate in a high-use factor service. These units have averaged 97.4 percent in availability and 99.5 percent in reliability. This includes the gas turbine accessories.



PRIOR YEARS DATA

	1954	1955	1956	1957	1958	1959	1960
NUMBER OF TURBINES	17	19	26	26	28	57	68
AVG. USE FACTOR	82.9	90.3	85.6	88.8	87.6	88.4	89.5
AVG. FIRED HOURS	—	—	—	—	—	27000	34400
CUMULATIVE AVAILABILITY	98.2	98.3	97.6	97.9	98.0	97.9	97.4
CUMULATIVE RELIABILITY	99.2	99.6	99.3	99.7	99.6	99.5	99.5

Figure 11. Availability-Reliability of Gas Turbines.

VIII. CONCLUSIONS

When this study is completed, an evaluation of the integrated gas turbine power plant will be made by the Maritime Administration and compared with other power plants in the study program. The information developed to date on this propulsion plant concept can be briefly summarized as:

a. A design which is readily automated with emphasis on self-regulating system.

b. A guaranteed all purpose fuel rate at design conditions of 0.510LBS/ FUEL/SHP-HR. An average of several units is expected to have an all purpose fuel rate of 0.500 LBS/SHP-HR or less. Also, the potential exists for an all purpose fuel rate on the initial gas turbine design of 0.489 at a firing temperature of 1550 F. The latter is based on successful completion of development work on fuel treatment systems and materials.

c. A power plant weight reduction of 284 tons compared to conventional steam turbine power plants of the same rating.

d. Final economic information on the initial design is expected to be comparable to existing power plants. However, subsequent units would show improved operating economy over other type power plants through improved efficiencies and higher firing temperatures.

An artist concept of the propulsion plant in reference ship PD 108 is shown on Fig. 12.

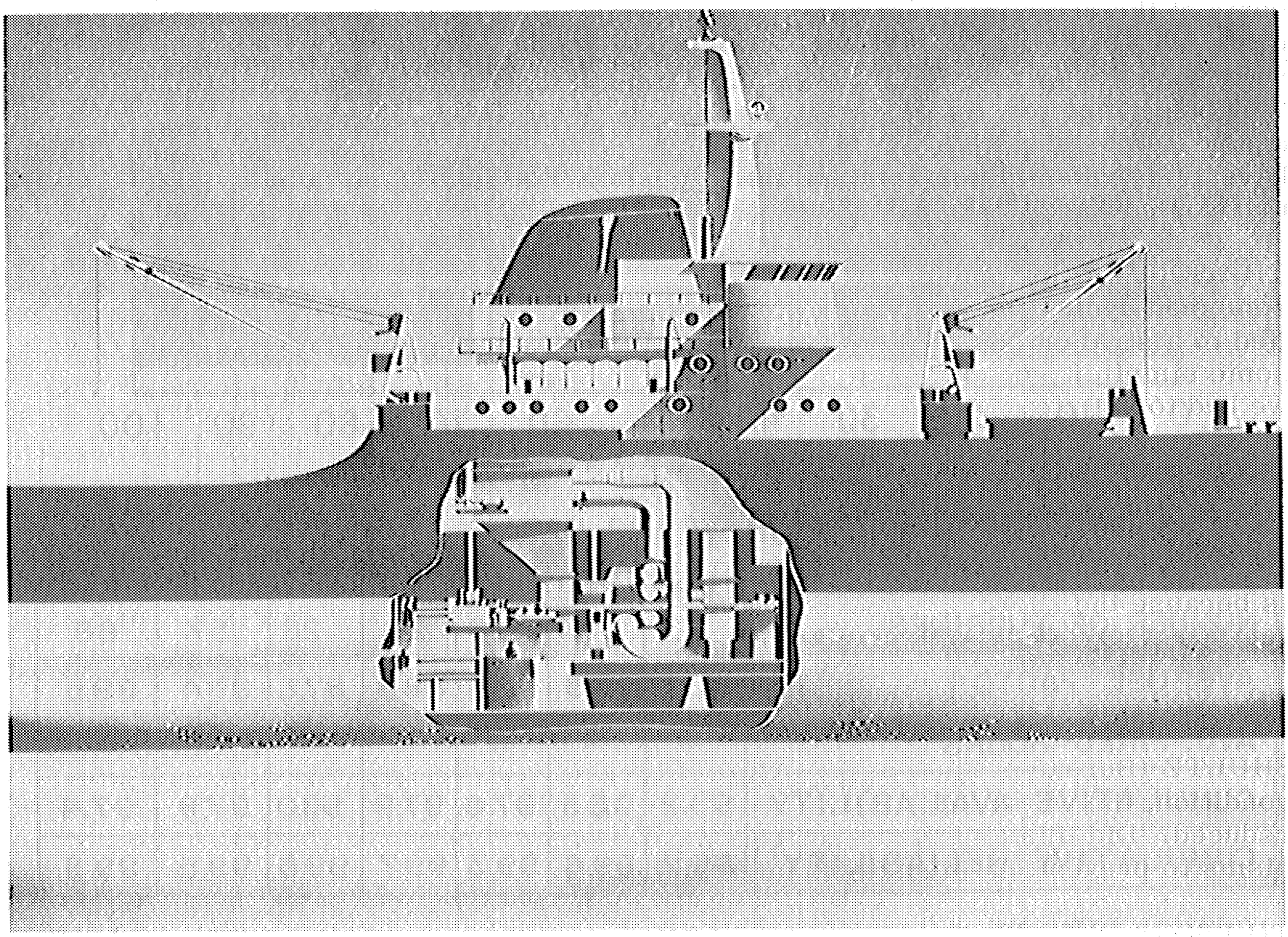


Figure 12. Artist Concept of Propulsion Plant in Ship.

DISCUSSION

HOLM (Webb Institute): I have one question regarding the steam atomization. This is a surprising amount of steam, roughly 8 times as much as we would expect on a steam atomizing burner on a boiler. Can you tell us why?

TILLSON: No, I can't tell you why. We use this steam flow in a certain ratio with the fuel flow, and it comes out to this amount. With about 10,000 pounds of fuel per hour, it's about an .8 to 1 ratio.

HOLM: You normally use about 1 pound of steam for each 1 pound of fuel oil?

TILLSON: Yes, we can go just slightly below 1 pound of steam.

BERMAN (Westinghouse): Since you stated that in order to get good efficiency out of this plant, you should recover your heat to the feed water temperature rather than compressor discharge temperature, and since you have a waste heat boiler, feed pump, a steam turbine and a condenser, and further you are apparently using much more steam than you really need for atomization, in fact you are getting some of the benefits of steam injection, and you are running your starting turbine continuously and I assume balancing out your power units with variable nozzles, what you have is not a regenerative system, but a regenerative combined cycle. Why not completely eliminate the regenerator with its high pressure air piping and associated losses, and simply enlarge the existing steam recovery system that you have, and make a plain combined cycle such as the — similar to your STAG system that you now are promoting for industrial use?

TILLSON: I think that is a very good question, and I'm one of the proponents of combined cycles for marine power plants. However, this study program was limited to a straight gas turbine. We felt that in using the starting turbine we achieved some simplicity by eliminating a clutch and letting it run at all times. As long as we had to put steam to this turbine, we felt that if we let it run all the time it would serve as a regulating device. There is a pressure regulation at the inlet of this turbine and also a bypass around this regulator so you will always have cooling steam.

BERMAN: In effect you are saying the reason you didn't go to a STAG system is because the scope of your contract limited you to mainly a pure gas turbine system.

TILLSON: Yes, sir, that is right.

SHULTZ (Bureau of Ships): In reducing the ship's speed from the head steaming condition, the windmilling speed of the propeller tends to increase as the pitch is reduced. Did this fact cause you to have to consider any unusual control mechanisms to prevent overspeeding?

TILLSON: We believe that in the final analysis of the control system, we will have to have some time delay in going from full pitch or full throttle back to zero pitch position. In other words in an integrated lever you can't slam this forward or slam it to the zero position and reduce your speed and your propeller pitch fast. We think this has to be timed.

I also would like to comment that this was no problem on the John Sergeant. I was talking with the former chief engineer on the Sergeant about this just the other day, and he indicated that the Sergeant's control dumped the pressure on the compression discharge so you took the energy off the gas turbine, but the problem, I assume, is the fact that the propeller is being windmilled due to the ship's motion through the water. What we have to do is prevent an overspeed signal, and we have to look into this problem. We're at this point now and investigating it. It hasn't been overlooked.

MCMULLEN (McMullen, Assoc.): Which two components do you consider the least reliable in your system?

TILLSON: That's a good question. Well, I believe that there is going to have to be a demonstration of a controllable pitch propeller of 20,000 horsepower. I believe it can be made into a reliable device. I might compare it to an aircraft propeller which is controllable pitch, I know that you don't have the forces involved, but this is quite a reliable device.

The next component — I would hesitate to say. Do you have any comments on that question, Mr. McCallig?

MCCALLIG: No.

MCMULLEN: The reason I asked for two, I know you would say the controllable pitch propeller, but I'm interested really in the one — do you consider that a regenerator is as reliable as the gas turbine itself?

TILLSON: Yes.

MCMULLEN: Has that been the experience to date?

TILLSON: It has been a very reliable piece of equipment. I will point out that those units that are operating in the field are operating on natural gas. I would also like to comment here — I believe that you noticed that the regenerator was mounted vertically above the unit. Now this was for a purpose. Generally you try to keep your weight low in a ship, but the regenerator will have a tendency to foul when you treated Bunker C fuel and you have some of the deposits coming out. This fouling rate depends upon the amount of additive you put in and the experience on the Sargeant indicated that they would have to occasionally clean the regenerator. We mounted the regenerator in our study in the vertical position so that openings could be provided in the top and the regenerator could be water washed occasionally and drained through the exhaust duct at the bottom. Most fuel deposits are water soluble and will wash off.

MCMULLEN: What was the experience with the regenerator in the Sargeant?

TILLSON: I explained that this had to be cleaned occasionally.

MCMULLEN: Were there any failures on it?

TILLSON: The regenerator on the Sergeant failed because the support to the regenerator buckled in a storm and the tube sheet cracked. I would also like to point out that this is a new regenerator design and it is a plate fin type.

MCMULLEN: The one on the Sergeant was a new design?

TILLSON: Yes, but a different manufacturer.

Name not given (Bureau of Ships): In your consideration of the reliability of the controllable pitch propeller did you also evaluate the electric drive as an alternate means of reversing?

TILLSON: Yes, we did. We looked at electric drive and we considered that it would be approximately 2% less efficient. It would cost more and it would weigh more. We didn't see any advantage of using an electric drive. A high electrical load was required in port and then the gas turbine would be operated. Also the sterm of this particular reference ship came up rather sharply and it would be rather difficult to get the propulsion motor all the way aft.

WELLING (Bureau of Ships): How have you handled salt air ingestion into the air supply?

TILLSON: Salt air ingestion is always a critical problem when operating a gas turbine. In this particular design we plan to utilize some features on the top side which would incorporate J or channel louvres arranged vertically and behind these we plan to use a device called a demister. These would be arranged at an angle, with the base slanting inward, and these in turn would be mounted in a trough so that the water could be carried off. The efficiency is over 99% and the manufacturer offers some very fine results from this device. We also are considering the possibility of operating this demister in a wet condition by running salt water over the front face which is expected to remove any dry salt particles in the air.

The demister would take out entrained moisture particles and by wetting the front face, we expect to take out the dry salt. I don't believe that there is any device that is going to take 100% of this salt out. We can use equipment that will take out a great deal of the salt, but eventually some salt will deposit on the compressor blading and when this happens these deposits must be removed by various means. The compressor can be water washed or some walnut shells can be run through the blading. We expect to operate for perhaps a month without any deposit problem as long as these features are designed into the inlet duct.

HAUSCHILDT (Bureau of Ships): Are you using Bunker C fuel on gas generators?

TILLSON: It will use either diesel or Bunker C fuels.

HAUSCHILDT: I'm not sure I understand this manning. You make the statement that it requires 3 less for engine room operation than for bridge control. Yet it would appear from the tabulation that you have, that it is the other way around. Could you explain that?

TILLSON: Eight men are used when operating from the bridge. This is a maneuvering station on the bridge. It doesn't take an engineer on the bridge to maneuver, but you have to have an engineer on the control console in the engine room.

HAUSCHILDT: You show 8 men for bridge control and 11 men for engine room control in your tabulation.

TILLSON: Yes.

HAUSCHILDT: In the text it is the other way around as I understand it.

TILLSON: It is?

MCCALLIG: The statement in the text in reference to the main control console is not directly related to either of the manning schedules shown in the tabulation. The statement referred to the main control console which includes all the instrumentation normally found on the main gage board. We don't believe deck officers are going to agree to monitor such parameters as bearing and exhaust gas temperatures. Therefore it would be necessary to station an engineer on the bridge to perform that duty. Such an arrangement was not recommended. The two manning schedules tabulated are predicated upon location of the main control console in the engine room under surveilance of an engineering watch-stander. Control of propeller speed and direction however may be from either the bridge or the engine room. The manning schedules shown in the tabulation refer to either arrangement. The 11 man schedule includes one extra man per watch in the engine room. If the engine room has to acknowledge and carry out telegraph orders from the bridge, we think two men are required in the engine room. The second man sees that the first man hasn't dropped dead or become incapacitated in any way. If the propeller control is on the bridge however, the second man in the engine room is not necessary. Neither of the two proposed manning schedules applies to an arrangement having the main control console on the bridge — the eight man schedule is for an arrangement having propeller control only on the bridge. Such an arrangement would be similar to the existing heading control system in which the rudder angle is controlled on the bridge but monitoring of the cylinder ram pressures must be done in the steering gear room.

HIRSCHKOWITZ (U.S. Merchant Marine Academy): I'm just wondering if this schedule is somewhat misleading probably to the disadvantage of your proposal. This is not a serious problem if you need an extra man when you are maneuvering. The overtime rates are not really that serious. You're only maneuvering for a very short period of time, so it seems to me that you should lay out a program for normal operation. It's been very customary for the first assistant to come down to the steam plant during maneuvering periods. This is not a serious problem. I think it is strictly a matter of company policy.

TILLSON: I think you will find a different idea with whomever you talk with on the manning arrangement. We feel that the gas turbine doesn't need anybody to watch it.

In fact there are a great many operating unattended, remotely, and automatically. However, there are other parts of the power plant that we aren't going to automate to the degree that you can take all the men off.

KIN (Esso International): With the use of residual fuel for this engine, how many hours can you run on this engine without doing the usual cleaning? How long does it take at that point?

TILLSON: The hours between cleaning are going to depend upon the fuel you are burning and the characteristics of the fuel. For instance, you may have a very high vanadium fuel. The present additives that are being used are magnesium sulfate that is epsom salts. If you use an additive like magnesium sulfate, the higher the vanadium content, the more of the additive you would have to use, and this would mean the more possibility of cleaning more frequently. Now how frequently, I can't tell you.

KIN: How much of a job is it to clean the nozzles?

TILLSON: We propose to incorporate a cleaning system in our final study, that would permit cleaning under partial load. I would say that it would get quite complicated to clean under full load. I know you can clean under part load or at idle conditions by injecting a cleaning compound. If you are willing to shut down and you have the opportunity of shutting down, I would say that the best way of cleaning would be by hot water and steam, and this shouldn't take very long, because we plan to incorporate openings to facilitate cleaning.

KAUFMAN (NBTL): Do you have any idea how much salt you could tolerate in your air coming in?

TILLSON: No, I don't know what this amount is. Maybe someone else can give that figure.

MCMULLEN: Have you an estimate percentage wise of how much it costs to take care of this treatment in the residual oil manning?

TILLSON: No, we haven't these figures yet.

HIRSCHKOWITZ: Is this oil clean-up setup going to be a batch system where you would have a tank and clean up say for a day's run or several day's run, or would it be running continuously?

TILLSON: We're working with the company that manufactures this equipment, and they are reasonably confident that they can automate most of these functions. For example a slurry can be made and pumped to where it is needed. I'm not too confident that this can be fully automated. I feel that the problem is getting the additive from the storage area to where you are going to use it. Also I think that there is a lot of work that has to be done in trying to get an analytical system — that is a direct monitoring system so you know the fuel characteristics at all times. These are problem areas that we have to face up to.

FRANCIS (Boston Naval Shipyard): What are the relative space requirements for this plant versus a conventional plant?

TILLSON: That was included in the paper. Ed, do you remember those figures?

MCCALLIG: No, not the exact numbers. The total space is about the same however for either plant.

THE APPLICATION OF RELIABILITY ENGINEERING TO THE INTEGRATED STEAM POWER PLANT

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Reliability engineering is a new and useful engineering tool that has not been applied extensively in Marine Engineering. Reliability engineering, as used herein, refers to the statistical evaluation of component and system reliability. It will be the intent of this paper to present some reliability data and maintenance and repair costs obtained from ship repair records, and to illustrate the application of this data in the evaluation of a steam power plant (Integrated Steam Power Plant) designed for reducing manning, maintenance, and costs.

The potential rewards available to the marine industry through the use of this new engineering tool are many and it is hoped that this paper will bring an awareness of some of the possible applications in the industry. The primary limitation of reliability engineering today is the lack of factual component performance histories. To be of value, it is essential that the basic tools, mean time between failure (M.T.B.F.) and failure rate (λ), be based on sound and accurate data and that the user is well aware of these limitations. Numerous factors enter into a discussion of why this data is not available, how it can be obtained, and the problems that will be encountered in obtaining it; these are included in the latter part of this paper along with general reliability considerations from the shipbuilder's viewpoint.

Component Reliability Data

There has been some recently published statistical component reliability data; however, this data has been determined under ideal conditions and primarily in connection with the electronic and aircraft industries as a result of the urgent need in the field of missile development. It is believed that to be useful in Marine Engineering applications, failure rates must be determined by tests simulating operating conditions or from recorded data of actual shipboard installations. No formal attempt is known to have been made to collect, analyze, and publish marine component failure rates.

In an attempt to provide at least some small portion of input for the vast amounts of quantitative reliability data that are required, the author investigated ship repair records for several ships. The detail records were analyzed for two machinery plants and the results of one of these is presented in Figs. 1 and 2. The other records investigated showed the same general repair cost trends as those presented but were not included due to the short operating period represented. The data that is presented is for eleven consecutive dry docking periods over 10 1/2 years (1951-1961) of ship operation. Fig. 1 presents the estimated total operating hours, mean time between failures, failure rates, and reliabilities for the various machinery plant components considered. "Failure" for this specific use was defined as any operating malfunction of the unit to the extent overhaul or repairs were considered desirable or necessary for continued service of the unit during the scheduled drydocking period; defects such as bearing replacement, leaking

MACHINERY PLANT COMPONENT FAILURE RATES AND MEAN TIME BETWEEN FAILURES

COMPONENT	NO. OF UNITS INCLUDED	TOTAL NO. OF FAILURES	TOTAL HOURS OPERATION	M.T.B.F.	FAILURE PER 100,000 HOURS	FAILURE RATE (PER HR.) λ	RELIABILITY* $R=e^{-\lambda t}$
PUMPS-MAIN FEED	2	4	85,680	21,400	4.7	.000047	.7542
MAIN CONDSTE.	2	4	80,600	20,150	5.0	.000050	.7407
AUX. CONDSTE.	2	3	85,680	28,600	3.5	.000035	.8104
MAIN CIRC.	2	8	80,600	10,080	10.0	.000100	.5489
AUX. CIRC.	2	7	85,680	12,250	8.2	.000082	.6113
OTHER S.W.	6	7	51,400	7,340	13.6	.000136	.4421
LUBE OIL	2	3	80,600	26,900	3.7	.000037	.8010
F.O. SERV.	2	2	85,680	42,800	2.3	.000023	.8711
F.O. TRANS.	2	0	13,700	13,700	7.3	.000073	.6452
MAIN BOILER -	2						
TUBES		6	128,500	21,400	4.7	.000047	.7542
REFRACTORY		14	128,500	9,200	10.9	.000109	.5200
S.H. TUBE SUPPORTS		6	128,500	21,400	4.7	.000047	.7542
SAFETY VALVES		13	128,500	9,900	10.1	.000101	.5456
SOOT BLOWERS		17	128,500	7,560	13.2	.000132	.4505
DRUM DESUPERHEATER		3	128,500	42,900	2.3	.000023	.8711
SUPERHEAT TEMP. CONTROL		5	128,500	25,700	3.9	.000039	.7914
FEED REG. VALVE		9	128,500	14,300	7.0	.000070	.5711
GENERATORS	2	1	171,200	171,200	0.6	.000006	.9646
MAIN TURBINES	2	0	161,200	161,200	0.6	.000006	.9646
MAIN RED. GEAR	1	1	80,600	80,600	1.2	.000012	.9305
D.F.T.	1	1	85,680	85,680	1.2	.000012	.9305
H.P. FEED HEATER	1	1	85,680	85,680	1.2	.000012	.9305
L.P. FEED HEATER	1	0	85,680	85,680	1.2	.000012	.9305
F.W. EVAPORATOR	2	0	85,680	85,680	1.2	.000012	.9305
AIR EJECTOR-MAIN	1	0	80,600	80,600	1.2	.000012	.9305
AUX.	2	0	85,680	85,680	1.2	.000012	.9305
EVAP.	2	1	80,600	80,600	1.2	.000012	.9305
CONDENSER-MAIN	1	0	80,600	80,600	1.2	.000012	.9305
AUX.	2	0	85,680	85,680	1.2	.000012	.9305
GAS AIR HEATERS	2	5	128,500	25,700	3.9	.000039	.7914
FORCED DRAFT BLOWER	2	0	85,680	85,680	1.2	.000012	.9305

*BASED ON OPERATION FOR 6000 HOURS

Figure 1.

glands, etc., have not been included. Extreme caution must be exhibited in the use of the data and in obtaining such data from repair records for several reasons. Failures which may have occurred between annual dry dockings may not appear in the tabulation.

Repair yard records do not appear to be the most logical source for reliability data since most operators have repair work done on a competitive bid basis and thus consecutive annual repair records for a useful period of time are almost impossible to obtain from one yard. The data presented represents, to the best of our knowledge, complete shipyard repairs for the 10 1/2 year period noted. To establish the operating hours, it was necessary to assume certain "use factors" for individual components and operating days for the plant. The ship was assumed to be at sea 320 days per year, in port 20 days per year, and unavailable 25 days per year. Realistic "use factors" on the basis of Marine Engineering experience were also assumed.

The reliability of each component (R) in Fig. 1 has been computed on the basis of an estimated total ship operating time of 6000 hours and is the probability of the component surviving the 6000 hours operating period without failure as defined. It should be noted that the reliability can be computed for any operating period and that 6000 hours was selected only for convenience as an average number of annual operating hours for a marine steam plant in cargo service. The reliabilities shown in Fig. 1, as with all other relationships used herein, are based on equations from reference 1 and were computed as follows:

MACHINERY PLANT COMPONENT MAINTENANCE, INSPECTION, AND REPAIR COST

COMPONENT	TOTAL 10 1/2 YEAR REPAIR COST - $	AVG. DAILY REPAIR COST-$	PERCENT OF TOTAL REPAIR COST
MAIN BOILERS	211,320	55.14	30.84
GAS AIR HEATERS	79,350	20.71	11.57
SOOT BLOWERS	6,345	1.66	.93
FORCED DRAFT BLOWERS	8,335	2.18	1.21
MAIN TURBINES	57,975	15.13	8.49
MAIN REDUCTION GEAR	10,895	2.84	1.58
MAIN THRUST BEARING	695	.18	.10
LINE SHAFT BEARINGS	795	.21	.12
GENERATORS	32,870	8.58	4.79
MISCELLANEOUS ELECTRICAL	17,225	4.49	2.50
CONDENSERS	17,380	4.54	2.53
OTHER HEAT EXCHANGERS	18,290	4.77	2.66
VALVES	97,200	25.36	14.17
PIPING	42,790	11.17	6.24
PUMPS-MAIN FEED	13,290	3.47	1.93
PUMPS-CIRCULATING	15,865	4.14	2.31
PUMPS-SEA WATER	7,440	1.94	1.10
PUMPS-FRESH WATER	4,790	1.25	.70
PUMPS-ROTARY	8,490	2.22	1.23
MISCELLANEOUS	34,450	8.99	5.00
TOTAL	685,790	178.97	100.00

Figure 2.

$$R = e^{-\lambda t}$$

Where: R = reliability or chance probability for survival, 1.0 being complete reliability

e = the base of the natural logarithm (2.71828)

λ = constant chance failure rate, failures per hour

t = time period for which the reliability is desired.

No attempt will be made here to derive or justify the statistical reliability relationships used; an excellent treatise on the subject may be found in reference 1.

It is acknowledged that the value of the presented data is limited since it represents only one ship of one operator; however, the other ship repair records investigated indicated similar trends and to the first order of magnitude and in a relative sense the reliability of the individual components is considered satisfactory for the application.

Repair Costs

Fig. 2 is a tabulation of shipyard maintenance, inspection, and repair costs to the owners for various individual machinery plant components and was extracted from the same repair records previously noted. This data provides some understanding of the relative reliability of machinery plant components and indicates where the high maintenance and repair (M & R) costs are. The actual costs to the owner for the 10 1/2 year period have been prorated and are shown on the basis of equivalent November, 1962, dollars; the cost per day is based on 365 days per year. Additional machinery components such as propellers and deck machinery showed some repair requirements also but were not included in this investigation as they were not of concern for the intended use; also, dry docking charges are not included. The costs shown include opening and inspecting equipment as well as performance of actual repairs; in the case of the main boilers and main turbines, inspection represents approximately 20% and 46.5%, respectively, of the total cost shown. The highest M & R cost, as might be expected is for the main boilers with a total cost per day including appurtenances of $77.51 or approximately 43% of the total bill. It is of interest to note that the second highest repair cost is taken by valves and piping which show a total cost per day of $36.53 or about 21%. Pumps show a total cost per day of only $13.02 or about 7% of the total M & R bill, which is perhaps lower than expected by some.

Application of Component Reliability Data

It will be shown how reliability data such as has been presented can be applied to the selection of machinery components for a marine steam power plant to evaluate several possible systems and to determine the most desirable combination. A hypothetical steam plant cycle will be used in the evaluation.

Fig. 3 illustrates in block diagram form the basic components of a steam power plant without any duplicate equipment. The steam plant represented thus is a series system with each component dependent on the other; that is, if any one component fails, the entire system fails. For simplicity and due to the lack of reliability data available, some actual components and subsystems essential to plant operation have been omitted. This should not affect the results materially as it is intended to consider reliability of various possible systems comparatively only. The reliability of each component as indicated on each respective block is taken from Fig. 1. The system reliability for a series system is the product of all the individual system component reliabilities, thus:

$$R_s = (.2515)\,(.6690)\,(.9305)\,(.7407)\,(.9305)\,(.9305)\,(.9305)\,(.9305)\,(.7542) = .0656$$

This means that the basic system considered has a 6.56% probability of surviving 6000 hours of operation without extensive repairs being required for a component. From this estimated reliability, it can be shown that the mean time between failures for the system is approximately 1400 hours.

RELIABILITY BLOCK DIAGRAM FOR A MARINE STEAM POWER PLANT

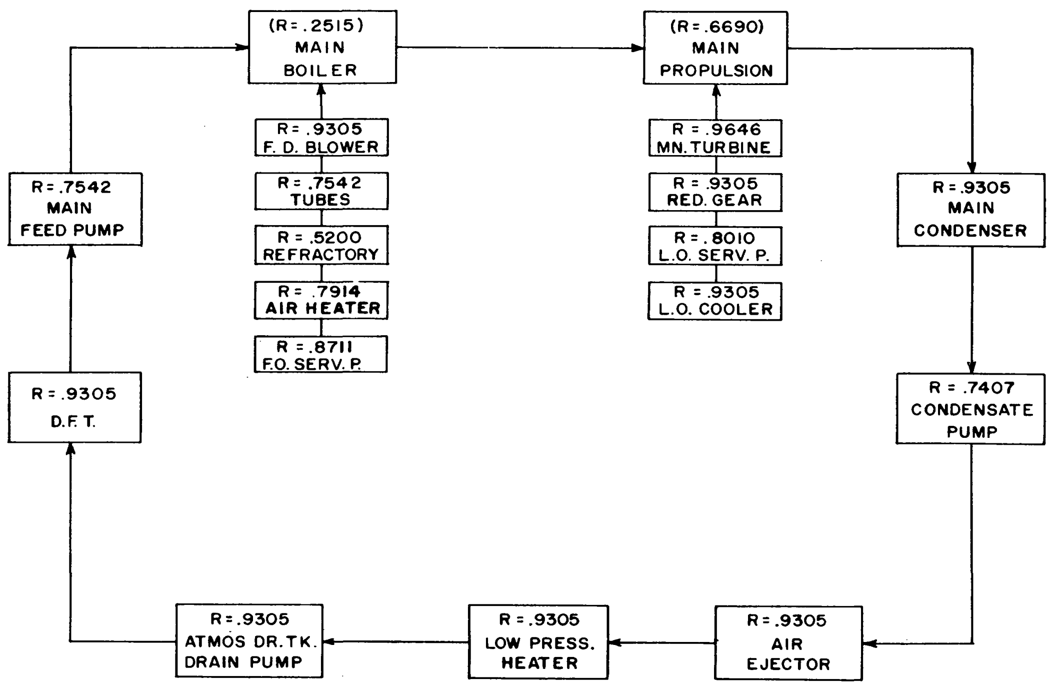


Figure 3.

In any reliability analysis, it is essential to keep in mind the definition of "failure" upon which the failure rates are based since it is the probability of such an event occurring that is being predicted. In this particular analysis, "failure" has been interpreted as extensive repairs, but it should be noted that these failures as recorded were not of such a sudden nature as to require immediate shipyard repairs. Generally, component failures which require shipyard repairs occur over a considerable period of time, give ample warning, and can be repaired at the first opportunity. The boiler refractory, for instance, has an inherently low reliability as indicated in Fig. 1; however, only in an extreme case would failure of the refractory render the boiler useless. In fact, the condition of the refractory is not usually known until the boiler is opened for inspection. A failure as defined here means that some special precautions will be necessary such as more positive temperature control, reduced power operation, etc. One now begins to appreciate the necessity for good reliable data and the necessity that the user be aware of the limitations of the data.

The apparently low plant survival probability of 6.56% is based on the limited data available and on 6000 hours of operation and further, the failure probability of 93.44% (100 - 6.56) does not represent the chance of the plant going dead-in-the-water — only the chance that repairs will be required for a component. For one voyage of say 100 hours, the survival probability for the same basic plant would be 95.56%. It may, therefore, be reasoned that if sufficient time and manpower are available after each voyage to perform repairs and preventative maintenance on the plant, a simple plant may be satisfactory. This is not usually the case however, and means must be found to improve reliability as it is usually desired to operate the plant from one annual dry docking period to the next without any extensive repairs.

Fig. 4 presents a comparison of reliabilities for variations of the basic plant with the intention of illustrating the effect of component duplication on plant reliability. The reliability of a system or subsystem having a component of equal reliability in standby, such as the main feed pump where one unit is in operation and a second unit is on standby to be put in operation upon failure of the operating unit, can be described by:

$$R = e^{-\lambda t}(1 + \lambda t)$$

MARINE STEAM PLANT PROPULSION SYSTEM 6000 HOUR RELIABILITY COMPARISON

SYSTEM	A BASIC	B	C	D	E	F
NO. OF COMPONENTS:						
MAIN BOILER	1	1	1	2	2	1
FORCED DRAFT BLOWER	1	1	1	2	2	1
FUEL OIL SERVICE PUMP	1	1	1	2	2	1
MAIN PROPULSION SYSTEM	1	1	1	1	1	1
LUBE OIL SERVICE PUMP	1	1	1	1	2	1
MAIN CONDENSER	1	1	1	1	1	1
CONDENSATE PUMP	1	1	2	1	2	1
AIR EJECTOR	1	1	1	1	1	1
LOW PRESSURE HEATER	1	1	1	1	1	1
ATMOS. DR. TK. DRAIN PUMP	1	1	1	1	1	1
DEAERATING FEED TANK	1	1	1	1	1	1
MAIN FEED PUMP	1	2	1	1	2	1
SEPARATE EMERG. PROPULSION	0	0	0	0	0	1
SYSTEM RELIABILITY, %	6.56	8.41	8.53	12.77	26.01	77.34

Figure 4.

Thus, if a standby feedpump having the same reliability is added to the basic system as in system "B" of Fig. 4, the feedpump subsystem reliability is:

$$.7542(1 + .282) = .9669$$

and the reliability for the entire system with the standby feedpump becomes:

$$R_s = (.2515)\ (.6690)\ (.9305)\ (.7407)\ (.9305)\ (.9305)\ (.9305)\ (.9669) = .0841$$

By the same method, the reliability of system "C" with a standby condensate pump was computed.

For systems "D" and "E", the two main boilers have been assumed to be in parallel. This is normally the case; both boilers are usually in operation and if one must be shut down the second one is already in operation to take over. The reliability of a system or subsystem in parallel operation, with both units of equal reliability, can be described by:

$$R = 2e^{-\lambda t} - e^{-2\lambda t}$$

Thus, the reliability of the boiler subsystem, with two boilers of equal reliability in operation is:

$$2(.2515) - \frac{1}{15.8} = .4896$$

and the reliability for the entire system becomes:

$$R_s = (.4896)\ (.6690)\ (.9305)\ (.7407)\ (.9305)\ (.9305)\ (.9305)\ (.7542) = .1277$$

The estimated reliability for system "E" was computed using the previously described methods for standby and parallel operating units. System "E", for comparison purposes, simulates modern marine steam plant practice with standby equipment for many of the plant components. While the system reliability has been increased considerably, 6.56% to 26.01%, over the basic single component system, it must be noted that the system has become quite complex with interconnecting piping, valves, etc., for the dual components. An indication of the reliability of piping and valves has been given by the high repair costs noted for these items in Fig. 2. It can be concluded that the added complexity for the interconnecting piping, and controls, although not included in the numerical reliability evaluation, would limit the system reliability to a somewhat lower value than is shown in Fig. 4. Therefore, duplicate equipment is an undesirable and expensive means of improving plant reliability.

System "F" illustrates the effect on plant reliability of providing a completely separate emergency propulsion system. It has been assumed, for the purpose of illustration, that an emergency propulsion system can be obtained that will have a reliability at least equal to that of the steam turbine propulsion unit shown, or .6690. The reliability engineering evaluation of a system with another system in standby, but of unequal reliability, is described by:

$$R = e^{-\lambda_1 t} + \frac{\lambda_1}{\lambda_2 - \lambda_1}\left(e^{-\lambda_1 t} - e^{-\lambda_2 t}\right)$$

The reliability of the basic system with the emergency propulsion feature included would then be:

$$R_s = .0656 + (-1.173)(-.6034) = .7734$$

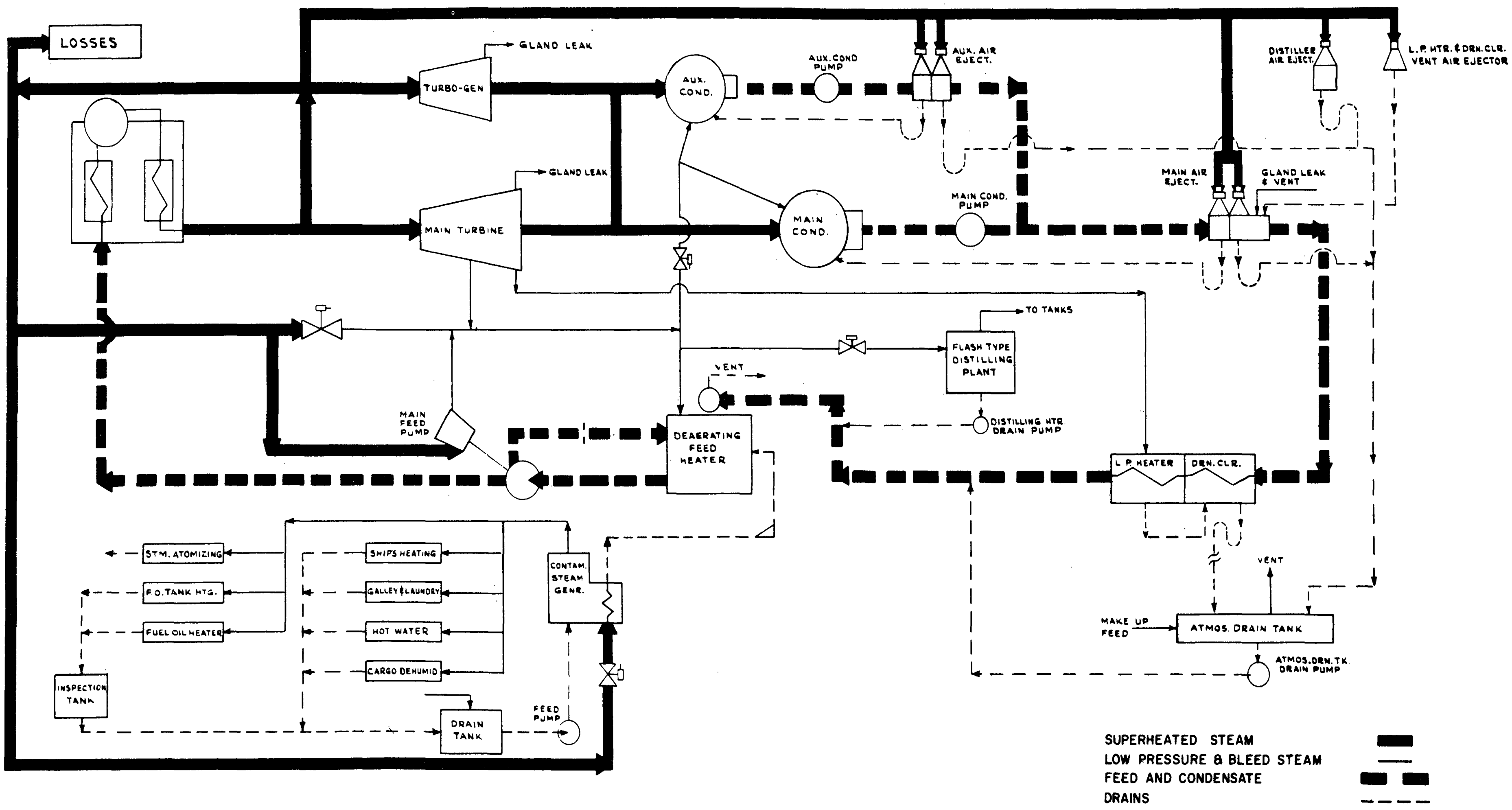
as shown in system "F" of Fig. 4. The apparent increase in plant reliability from 6.56% to 77.34% is quite significant. It is only fair to note that these figures represent the probability of the plant completing 6000 hours of operation without repairs at no specified performance level. It is unlikely that the emergency propulsion feature would have the power capability of the normally used propulsion unit. Likewise, one boiler operation with systems "D" and "E" would not propel the ship as fast, or give the same performance as two boiler operation. This situation has not been evaluated in this analysis and can only be done so by the vessel operator. To illustrate this, the calculated reliability of system "F" may be considered as including three reliabilities. The system has a probability of 6.56% of successfully completing 6000 hours operation at the steam power plant performance level, a probability of 70.78% of successfully completing 6000 hours of operation at the emergency propulsion performance level, and a probability of 77.34% of successfully completing 6000 hours of operation at any performance level. The acceptability of such an arrangement must consequently be the decision of the operator who must factor in the significance of possibly completing a voyage at reduced power or at least the chance of a delay due to reduced power operation while making repairs.

Evaluation of the Integrated Steam Power Plant

The Newport News Shipbuilding and Dry Dock Company entered into a contract with the Maritime Administration in January, 1961 for a design study of an integrated steam power plant for marine use. The entire study team consisted of Newport News, Babcock and Wilcox Company, DeLaval Turbine Inc., Bailey Meter Company and Diamond Power Specialty Corporation. The project had as its stated objective to develop, through the application of technological advancements and modern power plant techniques, a steam turbine power plant in the 20,000 SHP range for a cargo ship in maritime service which would have a low overall cost, a high degree of simplicity, reliability, safety, and minimal maintenance.

On the basis that system reliability is inversely proportional to the number of system components, the study was begun with a simplified steam power plant cycle including only the least number of components essential for a workable steam plant. Additional components were added to the simplified plant to attain optimum overall plant cost after carefully considering the overall effect on the study objectives noted. The resulting recommended cycle is shown schematically in Fig. 5. A few of the features of this plant recommended by the project team representing over 300 years of combined experience in the power industry, are as follows:

1. 600 psig and 875°F rated superheater outlet steam conditions.
2. One single furnace, two drum, bent-tube, natural circulation main boiler utilizing an economizer and downcomer saturated water air heater as heat recovery devices and also incorporating a multi-pass, walk-in, convection type superheater and four Y-jet, automated, retractable, steam atomizing burners.
3. One single-casing, impulse type main turbine including astern elements in the low pressure end and with bar-lift type throttle valves.
4. A locked-train, double-helical, double-reduction gear connected to the main turbine by a flexible mechanical coupling with provisions for coupling the standby ship's service gas turbine generator for direct mechanical emergency propulsion.
5. Two turbine-driven, single-stage main feed pumps and two motor driven



INTEGRATED STEAM POWER PLANT CYCLE

Figure 5.

lube oil service pumps but with a minimum of the other usual redundant auxiliary equipment.

6. Basically a solid state main plant control system.

7. A centralized engineer's operating station for single operator control of the plant.

Numerical reliability evaluation of the integrated steam power plant has not been made due to lack of data and due to lack of plant component design details.

The Newport News project team has based the design of the integrated, automated power plant on the thesis of simplicity and reliability. Much of the duplicate stand-by equipment considered necessary for reliability in today's marine steam power plants has been eliminated on the basis that it does not provide optimum reliability but rather only adds to the complexity, cost, and vulnerability of the plant.

It must be emphasized that the team's desire was to obtain "optimum reliability" — not maximum reliability as is often proclaimed for mechanical equipment. This point is illustrated in Fig. 6 which is a theoretical plot and can be applied to any component or even an entire machinery plant. It can be reasoned that service costs on a piece of equipment to the owner will be high with low reliability and will diminish with increasing reliability; this is represented by curve "S". Similarly, it can be reasoned that the initial cost of the piece of equipment will be low with low reliability and will increase as its reliability is increased; this is represented by curve "I". The sum of these costs is the total cost to the owner and is represented by the curve labeled "S + I." From this simple plot, it is evident that "optimum reliability" is the point where total cost is lowest and it would be indeed unusual if this occurred at maximum reliability for a marine steam plant.

Recognizing the economic factors involved, the team approach was to apply existing off the shelf designs with improved reliability to obtain a realistic plant without resorting to developmental designs. Thus, designing to produce a simple and reliable plant has led to an integrated power plant that is composed of proven equipment and, therefore, equipment that requires little or no development. The integrated power plant design that is proposed can be put into the hardware stage immediately.

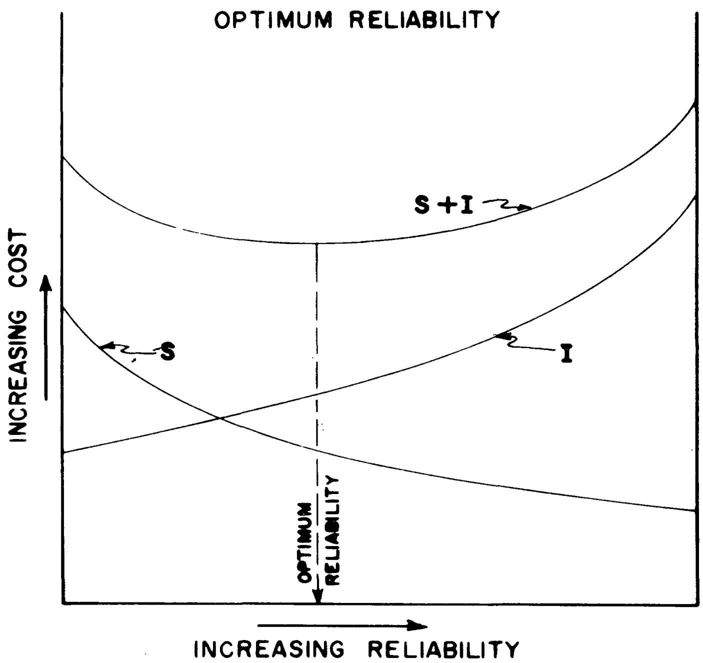


Figure 6.

The reliability of conventional marine steam plants has been maintained at an acceptable level by utilizing duplicate equipment; two main boilers, two and sometimes three main feed pumps, two condensate pumps, two fuel oil pumps, two lube oil pumps, and so on. With the integrated plant, that is, the plant where components have been selected on the basis of compatibility with the remainder of the plant and to complement other components in the plant, only two components are duplicated. Two main feed pumps and two lube oil service pumps will be provided. The elimination of the usual redundant equipment is made possible, to a large degree, by the application of a split shaft gas turbine prime mover in the dual role of stand-by electrical generator drive (its normal function) and for emergency ship propulsion.

The 750 KW capacity of the gas turbine generator set will be sufficient to drive the ship at about seven knots and simultaneously to develop about 250 KW for other ship electrical needs. As previously noted, it is impractical to provide emergency propulsion capabilities equal to the main propulsion system. The basis for the seven knot emergency propulsion is that most of the failures that occur at sea can be remedied in less that a day's time and the emergency unit would be used only until the steam plant could be put back on the line. Both the stand-by gas turbine and the steam turbine propulsion plants are dependent on the main reduction gear, main shafting, and propeller system. Other modes of emergency propulsion independent of these components were considered, but rejected due to prohibitive cost and with the knowledge that excellent reliability has been obtained from reduction gears, shafting, and propellers.

It can be seen in Fig. 3 that the main boiler has the lowest reliability of any component considered and it will be noted that for a series system, the system reliability can be no greater than the most unreliable unit in the system. Realizing this and the fact that providing duplicate equipment is an undesirable means of optimizing reliability, the Newport News team devoted considerable effort to improve boiler reliability.

A requisite of the design was that the boiler burn Bunker "C" oil; further, it was desired that the boiler be capable of satisfactory operation with any grade oil within the vast range of Bunker "C" available. The comparatively conservative selection of steam conditions of 600 psig and 875°F was made primarily on the basis of obtaining acceptable plant fuel consumption while affording optimum main boiler reliability without selective bunkering. Higher steam conditions, in fact, up to 1200 psig and 1050°F, have been used for marine plants; however, experience has shown that with high temperature installations accelerated boiler tube corrosion and excessive slagging will occur and result in high boiler maintenance and low boiler reliability. Repair records that were investigated for two vessels in similar service, one with a superheater outlet temperature of over 1000°F and the other with 850°F, showed average annual boiler repair bills of $27,700 and $17,000, respectively. The selected maximum superheater outlet temperature of 875 F will permit a boiler design with ample margin on a maximum superheater metal temperature of 1000°F which is well below the accelerated corrosion zone and which, when combined with other boiler features, will provide a reliability considerably higher than that shown in Fig. 3.

In selecting the steam plant cycle, a choice had to be made between the use of a gas air heater or an economizer as boiler heat recovery equipment. The air heater versus economizer question has been argued since the inception of modern steam plants and the decision still remains a matter of personal preference. For the integrated plant, an economizer has been selected.

The data in Figs. 1 and 2 show gas air heaters to have a relatively low reliability and high maintenance cost in relation to other plant components. Five air heater failures (complete tube replacement) in the 10 1/2 year operating period were noted. Other repair records investigated for another vessel fitted with both gas air heaters and economizers showed six complete air heater retubings while only one instance of economizer tube repairs. Data on this subject has been presented by Messrs. Shulters and Van Riper in reference 2. Although their findings indicated a 40% higher replacement rate for economizers than for air heaters, 9 air heater fires were noted while no fires were noted with economizer installations. Details of the individual economizer difficulties were not available and it may well have been that the high economizer replacement rates were the result of too low a feed temperature which will definitely cause excessive corrosion. With the integrated plant, the feed temperature to the economizer will be pegged at 280°F which has been found to produce long economizer life with minimum corrosion. The preference for one over the other has been based on personal experience and it is in this area of design that good reliability data can be applied with success. Unfortunately, neither of the cases cited could be transformed to failure rates due to lack of operating time data. The very design of the gas air heater, i.e., gas to air heat transfer promotes the possibility of stack fires which would be wholly unacceptable in an automated plant. Regenerative type air heaters and tube inserts have helped to extend air heater life; however, it is not considered that the unattended service desired will tolerate the still existing problem of air heater fires.

For the remainder of the feed heating cycle a low pressure feed heater with combined drain cooler and a deaerating feed tank, both operating with bleed steam, have been selected. The first thought of the team was to have the feed heating cycle remain as simple as possible by not utilizing regenerative feed heating or the low pressure heater. Further investigation indicated that the slight decrease in plant complexity and increase in reliability could not completely justify the fuel savings possible with the extraction cycle. An estimated 25 year fuel savings of $482,000 is afforded by the cycle shown in Fig. 5 over a non-regenerative cycle and it is not believed that the maintenance of the additional equipment would be sufficiently high to warrant use of the more simple non-extraction cycle. It is evident that a fully developed reliability engineering program could complete the evaluation of the extraction cycle. To assure the reliability of the proposed cycle, several unusual features have been included. The turbine extractions will be fitted with motor operated, remote actuated, positive closing valves to insure against loss of condenser vacuum and turbine overheating during astern operation. The low pressure feed heater vent which is normally orificed to the main condenser will be replaced by a separate air ejector to prevent loss of condenser vacuum. Vacuum connections to the main condenser have been minimized and those that are used will be fitted with welded connections and steam or water sealed valves. The low pressure heater and drain cooler will be located sufficiently high in the ship so that the condensate will drain by gravity to the atmospheric drain tank thus eliminating the necessity for a drain pump or vacuum drain line to the condenser. Fig. 1 indicates that the low pressure heater itself has a relatively high reliability. Marine Engineering experience bears this out.

The selection of a single main boiler was based on system and component simplification (increased reliability), improved operating procedures, the previously noted independent emergency propulsion system, and certain quality features of the proposed boiler. An important improved operating procedure is that the entire plant has been so designed that the boiler will be steaming continuously from one annual repair period to the next. This will eliminate the detrimental effects of thermal transients experienced with conventional installations and increase considerably the reliability shown for boiler refractory in Fig. 1. It can be seen from Fig. 4 that a significant improvement in plant reliability can be made by the addition of a second main boiler of equal reliability. Experience with multi-boiler installations however, indicates that boiler failures are usually the result of maloperation or failure of a supporting auxiliary and, therefore, a boiler casualty that results in sudden shutdown is generally not restricted to only one boiler; i.e., the second main boiler cannot be considered as providing 100% standby service. Further, with the second boiler, must be added additional supporting equipment. While the elementary reliability analysis that has been made does not take these additional considerations into account, it is certain that a more detailed analysis including all controls and equipment necessary to operate the second boiler would show an improvement of somewhat less than that indicated in Fig. 4. A more logical approach to improved reliability is to maintain maximum simplicity and to direct efforts to improve reliability of the boiler proper. Improved operating procedures, quality features, and a strict program of annual preventative maintenance will greatly reduce boiler troubles currently experienced. Further, it has been estimated that most failures that have been experienced can, if necessary be corrected at sea within 10 hours time.

It is essential that the boiler be provided with heated combustion air to improve combustion efficiency and ensure good combustion during low load operation. To supply this need, the boiler will be fitted with a newly designed air heater which will utilize the heat of the saturated water in the downcomers to heat the combustion air. This new design has a comparatively "flat" air outlet temperature characteristic and will provide relatively high temperature air at low loads which is necessary for extended reliable boiler operation. The design was chosen over the conventional steam air heater on the basis of expected higher reliability and system simplification; but cannot be completely evaluated from a reliability standpoint until operating experience is gained.

The boiler will be fitted with eight steam, motor operated, automatic soot blowers. There will be two vertical retractable units in the superheater, three rotating head units in the superheater convection section, and three rotating head units in the economizer. An in-line superheater tube arrangement in conjunction with care in positioning the soot blowers will result in greater boiler tube and refractory life than has been indicated in Fig. 1. Retractable soot blowers have been selected for the superheater as experience has indicated this type to be more effective and to have a greater life than the stationary type.

Fully automatic, steam atomizing, Y-jet oil burners were selected with due regard to the higher reliability they will afford. It is estimated that the boiler load range with the four steam atomizing burners will be at least 16:1. This will permit boiler operation under most all conditions without retracting any burners from service. Automatic burner features are to be provided to permit the operator to remotely withdraw, secure, or restart burners as may be needed in an emergency and for low load steaming for an extended period of time. Thus, for normal maneuvering all burners may be left in service which will provide a higher reliability than an installation in which burners must be withdrawn and inserted due to a low turn down ratio. Each burner will have an individual spark ignited diesel oil lighter with remote operation to permit remote manual or automatic restarts with safety. Normal at-sea burner cleaning and maintenance will be essentially eliminated with the steam atomizing burners which have clean burning characteristics in addition to automatic steam purging and cooling sequence when removing burners. Individual burner barrels and tips will be completely withdrawn from the furnace when not in use providing additional assurance of maintenance-free operation. The oil burners, the heart of automation, have been selected to complement the quality designed boiler and promote the desired extended reliable boiler operation.

A fully water-cooled furnace and water-cooled oil burner throats will greatly improve the refractory reliability indicated in Fig. 1.

Additional boiler quality features which it is predicted will result in improved reliability are a large 60-inch steam drum, special cyclone steam separator baffles, downcomer tubes to all circuits, improved accessibility to pressure parts, and replaceable superheater supports. It is indeed unfortunate that the boiler features noted cannot be evaluated quantitatively. As more reliability data becomes available and the shipbuilder gains experience in the field of reliability engineering it will perhaps be possible to estimate the value of such improvements numerically. Nevertheless, it is certain that the features noted will materially improve boiler reliabilities currently being experienced.

The aforementioned quality features apply to the main boiler design; however, the entire integrated plant has been selected to provide component compatibility. It was considered desirable, to insure boiler reliability, to provide a separate contaminated steam system. This will eliminate any possibility of boiler feedwater contamination from oil. The main condensate and feed system will be fitted with salinity indicators at critical locations to detect leakage of seawater into the system. A deaerating feed tank will be provided to remove gross oxygen in the feed. An attempt was made to design a feasible automated feedwater chemistry control and treatment system; it was reasoned that an automatic system would provide higher reliability than existing manual methods by eliminating the human error element. An automated system was designed; however, it was found that the requirements for manually checking the instrumentation could not be eliminated, with the result that essentially as much human assistance with the automatic system would be required as with the manual system. A non-automated treatment system utilizing hydrazine for oxygen scavenging and tri-sodium and di-sodium phosphate for boiler water treatment was recommended. Hydrazine will be introduced in the atmospheric drain tank; thus, eliminating the need for pumping equipment and the phosphate will be introduced in the feed by means of a by-pass pot feeder. It was concluded that automation of this system would result in decreased reliability and that conventional feed treatment methods, when conscientiously applied, would be adequate.

Only one fuel oil service pump is proposed; however, the fuel oil transfer pump will be arranged to back-up the service pump. Approximately 60% rated power can be made when supplying the boiler with the transfer pump.

Main turbine inspection was previously noted to be about 46.5% of the main turbine repair bill of $15.13 per day. The single casing turbine in lieu of the more conventional cross-compound will reduce the M & R costs by reducing the inspection requirements.

The cross-compound turbine arrangement has been thought of for many years as a means of emergency propulsion. A close inspection reveals however, that either turbine is only a standby for the other and not for the entire plant. Any casualty to a critical steam plant component, which results in the inability to produce steam, renders the entire conventional propulsion plant useless. The proposed integrated plant with a single casing main turbine and the previously noted emergency propulsion system will unquestionably provide 100% standby propulsion and simultaneously reduce maintenance, weight, space, and initial cost requirements. The single casing turbine will incorporate proven quality design including a stiffened rotor for the increased length, automatic casing drains, an automatic gland seal system, vibration monitoring equipment, remote bearing temperature indication, as well as conventional overspeed protection.

Fig. 1 shows a comparatively high reliability of .9646 for steam turbine driven electrical generators. This illustrates the high reliability that can be obtained from an automated piece of equipment. It also confirms the team's belief that high reliability can be assigned the S.S.T.G. and thus the decision to include only one steam driven generator along with the gas turbine unit to supply the ship's electrical needs.

The main plant control system, including automatic burner control, combustion control, and feedwater control, are primarily solid state electronic. A conventional hydraulic control system for the main turbine and S.S.T.G. set is proposed.

The solid state system was selected due to the requirement for eventually moving the entire plant control function to the bridge. The controls will be in module form with self checking and fault detection circuits included. All components will be located in nine steel cabinets with easy access in the air-conditioned engineer's operation station. Solid state control equipment, it was estimated, will have a higher reliability than conventionally used pneumatic equipment and it has been proved by tests that higher reliability is obtained with solid state electronic equipment than with the inherently unreliable vacuum tube type. Conservatively derated components with at least 3:1 power dissipation derating at 80 to 100° F ambient temperature and no moving parts will be employed thus assuring good reliability.

Individual control subsystems have been selected with considerable thought to reliability. The feedwater control system will be of the three element type (steam flow, feed flow, and drum level) which will provide the maximum protection against loss of feedwater and carry over. Features will be incorporated to automatically shut off fuel supply if the drum level drops to a predetermined level or if the combustion air supply is lost. The necessity for a conventional feed water regulating valve, which shows a low reliability in Fig. 1, has been eliminated by the use of a single main boiler and feed flow will be controlled by controlling the main feed pump turbine speed. A metered flow tie-back combustion control system has been selected; this scheme is independent of the number of burners in use, will maintain satisfactory fuel air ratio over the entire boiler operating range, and will insure air flow leading oil flow on increasing loads and oil flow leading air flow on decreasing loads. The electric control system power supply will be provided with a motor generator-battery set which will provide complete standby in case of electrical power failure.

Throughout the study it was reasoned that by reducing the requirements for manual operator functions to a minimum, the entire plant reliability would be improved. The engineer on duty in the enclosed engineer's operating station will have only one manual function to perform for normal operation including maneuvering the ship; this is to position the turbine speed control level — all other operations are automatic. To enable manual operations to be minimized, a data information system is proposed. The system will include automatic scanning and annunciation of vital points as well as automatic periodic log printout. In addition to the usually measured pressures, temperatures, and levels, vibration pick-ups will be utilized to inform the operator of critical rotating machinery performance.

An attempt has been made in this paper to illustrate one valuable role that reliability engineering may play in ship design. Many other possibilities such as an aid in establishing realistic preventative maintenance schedules and a means of reducing or justifying design factors of safety must have also become apparent to the reader. A word of warning is extended however, along with the newly found engineering tool: the results obtained from its use can be no better than the data upon which it is based and no greater than the respect which the user has for its limitations.

A warning must also be given to those ardent proponents of reliability engineering inexperienced with the ways of the sea: A wealth of data is available (providing a suitable means can be established to collect it) from the many vessels now in service; however, estreme caution must be exercised in analyzing it due to the many possible mishaps that may be called failures and also due to the unusual circumstances which may accompany a failure.

General Reliability Considerations from the Shipbuilder's Viewpoint

Marine industry personnel have always been cognizant of the value of applying machinery performance data to design work; however, the potential rewards to the individual ship operators and ship designers for establishing an organized system to collect, analyze, and disseminate reliability data appear to be far outweighed by the requirement to increase an already too high overhead. Additional reasons for the lack of performance data in the Marine industry may be cited as follows:

1. The actual machinery plant reliability obtained with existing plants, although not determined numerically, has been considered acceptable; thus, there has been little incentive to instigate a formal reliability program.

2. Although most ship operators proclaim the cost of machinery plant maintenance, repair, and replacement to be too high; it does not approach the reported $2 annual maintenance cost per $1 investment the Armed Services experienced with electronic equipment which led to formal reliability engineering in this country. A comparable figure for a marine steam plant is estimated to be about 2 1/2 cents per year per dollar invested; this figure is based solely on the owner's repair bill for the machinery plant; when maintenance crew wages are included the figure would be in the order of 5 cents per year per dollar of machinery plant initial cost. Nor has there been any great manifestation of equipment failures such as the often quoted 70% inoperative electronic equipment experienced by the Navy in 1949.

3. Chief engineers and operating crews are quite adept at jury rigs and have become accustomed to makeshift operation. Further, it is almost impossible to determine the necessary environment and operating conditions which accompany component failures.

4. Operators are reluctant to make public their operating problems due to keen competition in the shipping field.

5. It is also interesting to note that all the bodies concerned appear to think the other either has in his possession or has access to the desired statistics. Cognizant Government agencies think the shipbuilder has the data and the shipbuilders think the manufacturers and operators have it; the truth appears to be that nobody has it in an orderly usable form!

A form of reliability engineering has been employed in the marine industry by using only those components which have provided proven reliable service in past installations. A word of caution is in order when following this philosophy; namely the acceptance of ideas and improvements are slower, and the designer's knowledge of component performance is generally limited to the operator's complaints and the misleading "birth defects" that occur prior to the end of the guarantee period. The reliability engineers call this "intelligent guesswork" and warn that such reasoning may be, and frequently is, wrong by large orders of magnitude. On this subject, Lord Kelvin said, "When you can measure what you are speaking about, and express it in numbers, you know something about it; but when you cannot express it in numbers, your knowledge is of a meager and unsatisfactory kind; it may be the beginning of knowledge, but you have scarcely, in your thoughts, advanced to the stage of science, whatever the matter may be." Statistics have provided the means for numerical reliability evaluation; it is now the charge of the marine industry to provide the essential data and make use of the new tool to reduce ship construction and operating costs and aid in the ever increasing U.S. Maritime industry fight for survival.

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DISCUSSION

BOATWRIGHT (Bureau of Ships): The cycles are getting simpler apparently in the proposals for merchant ships, somewhat comparable to those that we have on combatant Navy Ships. It is enlightening. The air ejectors, just to get to one little simple item. I note that you appear to be using full six hundred pound superheated steam on the air ejectors.

The Navy's experience has been the higher the pressure, the smaller the nozzle in the air ejectors, and more likely to get stopped up because of the small size nozzles.

I would like to hear about the considerations that you give the air ejectors when you decided to use a full 600 pounds pressure on it.

RIDDICK: We have no experience in this field. We have, however, discussed this with a very reliable manufacturer, C. H. Wheeler, and they assure us that operation on full pressure with the small nozzle is possible.

This is an excellent instance where reliability engineering data could be applied. However, we need experience to draw this data from to tell us that full boiler pressure operation of the air injectors is satisfactory.

SALING (Pratt & Whitney): Did you make any attempts to apply failure rates to your simplified system for comparison with the other system you presented earlier? Was sufficient information available to you?

RIDDICK: No, sir, we definitely do not have sufficient data to come up with an entire system reliability. As reliability engineering becomes popular and known in the marine engineering field through symposiums such as this, I hope that we will be able to collect enough data so that, at least for this type of application, the marine engineer will have at his disposal enough failure rate data to use to make this type of analysis. I am not sure at this particular time what the value would be of the absolute reliability of an entire system.

FRANKEL (M.I.T.): First of all I believe we are all grateful to Mr. Riddick for making possible this most important point of this conference by showing us that complexity definitely reduces the reliability and not necessarily increases performance parameters, and we should really aim at simplicity in every respect.

This is particularly important with regard to the well presented point of redundancy to increase reliability, and I believe that the point of Dual boiler and the compound turbine is something all ship owners and operators should really consider very carefully. A very minor point is reliability of the stand-by main propulsion system. I simply would like to point out that theoretically it is impossible that two redundant systems have a reliability larger than the sum of the reliability of each individual system.

You had a main plant with reliability of less than 10% and stand-by plant of 66%, both giving us 77% which is impossible.

RIDDICK: Perhaps we can discuss this later. It sounds like mathematics to me.

FRANKEL: My question actually is on your repair costs and failure rate data. I didn't see any control equipment, and also the big item, valves, comprising 50% of the total repair costs, possibly does include some of the boiler mountings or at least control equipment for the boiler. I believe it might be quite enlightening to see where control reliability, particularly with regard to safety control devices, does come in.

RIDDICK: I agree with you and without being facetious would like to say that if you have data on control equipment, I would be very much interested in obtaining it. The repair records that I investigated were not of such a nature that I could readily obtain control systems reliability data from it.

DUNN (Electric Boat): I think that in the last few days, excellently summarized by yourself, we must have come to the realization that our future progress in this area is dependent upon the data that we are able to obtain. Unfortunately and perhaps fortunately at the same time, we have to depend on the customer and the user for the collection of this data.

I would make a point then that if we are careful with our reliability figures, probability, and always in addition to the probability figure assign a confidence level, we can talk in terms of confidence levels associated with those reliability figures as a function of the input data, and call the users attention to the fact that our calculations are dependent upon his input.

RIDDICK: A point that is brought out in the paper and I hope is clear from our discussion is that I acknowledge the limitation of the data and caution you on the use of it. It is taken from one ship's repair records only, and is not by any means considered satisfactory in the standards of reliability engineering. However, I have used this data to illustrate a point, and believe it to be satisfactory for the application. Other data that has been collected substantiate the data presented; however, the population was not great enough to calculate the confidence level or distribution.

HIRSCHKOWITZ (U.S. Merchant Marine Academy): This reliability is coming at an extreme cost one way or the other, even to the point of collecting data, simulating it, studying it, perhaps building more expensive and more exotic materials in our equipment. Have the insurance companies or agencies who extract penalties on us due to the fact that perhaps we appear unreliable showed any evidence that they may give us an incentive to invest in this direction? In other words, you talk about this turbine, often a big expense to open it up, would they conceivably agree to require us to open it up less frequently if you incorporate into it more reliability and design, and help you to invest in this direction?

RIDDICK: The regulatory bodies have not been approached on their feeling on the use of reliability engineering.

MCINTOSH (USCG): Anything I say here can be used against me. I must be careful. I think that all regulatory bodies should certainly become interested in this science of reliability engineering and should take cognizance of it in their regulations. We in the Coast Guard attempt to be flexible and reasonable. If we can be shown that the intent of a regulation is served by an arrangement not in literal compliance, we would, at least on a trial basis, allow operation of it.

MCCALLIG (Geo. G. Sharp, Inc.): I would like to add a comment on the difficulty of obtaining the type of data the author has presented here. I was formerly employed by one of the largest ship yards in the country with an extensive design staff and everytime we attempted any analysis of this sort, we could obtain data only from one source and that source was available to us only because the parent company owned the shipping line which supplied us with a year's reduced-data log sheets. Other than that we would have had nothing at all.

THAYER (Lykes Bros.): I'm a little confused here. We've for the last couple of days learned that progressive maintenance and preventive maintenance are all part of reliability, and now you are telling us that we should take one boiler and put it on a line for a year where we can't inspect it, we can't clean it. Now I know you are going to say that you are going to make a boiler that you don't have to clean and don't have to inspect, but let us face the facts — there isn't one, and isn't it going to increase the maintenance cost rather than decrease it?

RIDDICK: We believe that one of the primary causes of boiler maintenance and repair is due to subjecting the boiler to thermal transients by alternately raising steam and then securing the boiler. Our single boiler installation is designed to be continuously fired from one annual repair period to the next.

I would like to call on Mr. John Banker to say a few words about this subject.

BANKER (Babcock & Wilcox): We have built units that have been on the line as long as three years without cleaning. So it is obviously possible to do. We have a number of other ships that have been on the line for a year, and while they were cleaned, the people that did the job observed that it was not always necessary. Now there are many other jobs that have gone three months and probably couldn't have gone any further without cleaning. What we are trying to do here is to produce a unit which this type of action is unnecessary. Furthermore, I might observe that part of the wear and tear on boilers is probably due to too much water washing, because this can promote a rather gradual degree of corrosion which may not show up in the first few years, but in about 10 years you can get decided signs of this.

Now boilers that are overwashed are going to get this down at the lower part of the boiler where you tend to have the soot and the water combine, and while it is a necessary operation, we don't know of any other way to get rid of this stuff other than water wash. If you run into this problem, it should be done with about as minimum scheduling as possible.

RIDDICK: Is that satisfactory, Mr. Thayer?

THAYER: I might add one thing to that. I've talked to a lot of other ship owners and I know of very few who are ready to go one boiler at this time.

KIN (Esso International): We have been considering a one boiler installation, and we feel that from our analysis of boiler casualties on our domestic fleet there really haven't been too many troubles that we have had which would warrant a two boiler installation. As far as washing, we have boilers today that can go one year without being touched.

RIDDICK: Thank you sir.

MCMULLEN (McMullen Assoc.): In regard to this discussion concerning the length of time a boiler can be kept upon the line, I would like to mention that this is dependent upon the quality of the fuel oil. If you burn a distillate fuel in a boiler you can keep it on the line for a very long period of time, but this isn't the case of the present ship owner operating with Bunker C. He actually takes on fuel of an unknown specification. I would like to ask you how you intend to control this in your single boiler plant. If an operator burns certain Bunker C oil, as he does at present, your one boiler would clog up in three months.

BANKER: The ships that I mentioned are burning residual Bunker oil.

MCMULLEN: To which specification?

BANKER: Well, here is an oil man (Mr. Kin) and I would say that he hasn't been burning oil that I would consider really excellent.

MCMULLEN: What ash content?

BANKER: 0.1; 0.15 percent.

MCMULLEN: Total ash content?

BANKER: That's right.

MCMULLEN: You mean 150 parts per million?

BANKER: I'm talking about 0.15% ash now.

MCMULLEN: Total ash?

BANKER: That's right, which is pretty much run of the mine for bunker residual oil.

MCMULLEN: You're saying 150 parts per million.

BANKER: 1,500 parts per million.

MCMULLEN: Actually it was the experience of the United States Lines operating the *America* that when they were taking bunkers in New York just as any other ship operator might, they were forced to clean their boilers on the average of every three voyages, and it wasn't until they were able to specify a total ash content of less than 600 parts per million that they were able to hold off their cleanings for periods up to nine months. It is the total ash content which controls. If you put 1,500 parts per million of total ash content into a boiler you're going to clean it every three months; if you're that lucky.

BANKER: You can't argue with success though. This has been done. I would say this in answer to what you are talking about. I will agree that on a ship like you have just cited that this is probably very true. Where does this collection occur? Well, the superheater is one serious part. In fact this can block the draft to a point where you have to shut down and clean. I've seen superheaters that were 90% bridged over and aside from the fouling problem, it doesn't do the superheater tubes that are still taking gas any good. With the Lance blowers, and that is what Mr. Kin is talking about, we have had a considerable amount of success removing this slag. If anybody else at this meeting has this type of blower on any of his boilers, he would have probably observed the same things.

Now I say it is only on jobs that have the Lance blowers that this can be done. The other side of the coin is the gas outlet end of the boiler. This is a place where if you start chasing too many BTU's you're talking about a low sink temperature and you can get wet soot which is a very obnoxious thing. One of the features of the design that has been picked is an economizer with a sink temperature of about 280° F. We have found that the soot at this temperature is drier. It doesn't collect like it does when you use a metal temperature, say in an air heater, where it might be 160° F near the air inlet and you are well within the dew point range.

In fact I'll say this. This 280°F cycle started about 10 years ago, and to date there has been no failure of any economizer tube due to external corrosion on any job that had this temperature. The only failures, and these have been few, have been in welds.

MCMULLEN: May I ask one question? Are you saying that your company is prepared to guarantee a boiler that will go a year without being cleaned, burning fuel oil with 1,500 parts per million of total ash content?

BANKER: I don't know what we could guarantee. I couldn't say this. All I can tell you is that it is being done.

MCMULLEN: I debate your answer.

ADVANCED MARINE ENGINEERING CONCEPTS FOR INCREASED RELIABILITY

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