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Small Craft Engineering: Propulsion by the Internal Combustion Engine

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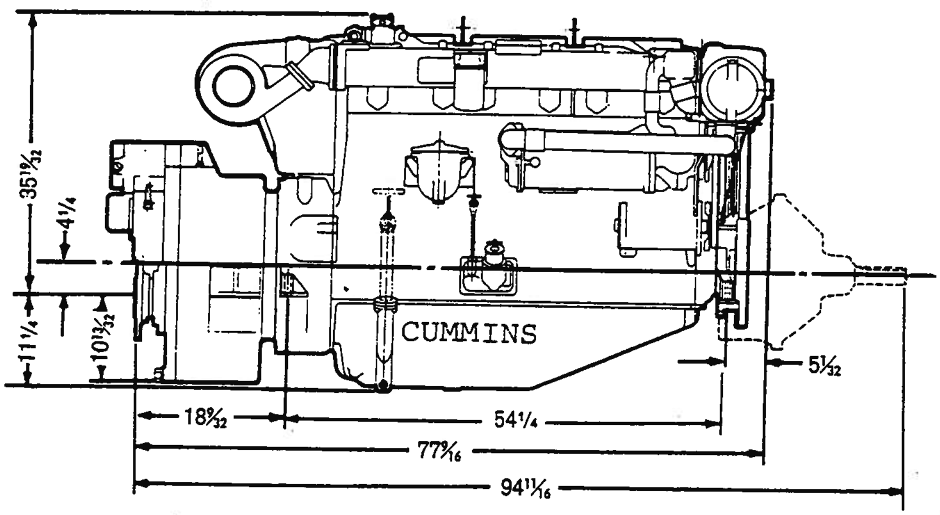

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PREFACE

This volume discusses the internal combustion engine as the source of propulsion power for small watercraft. The fundamental assumption is that the engine-propeller unit in a marine environment is a unique machine, quite different from the engine itself, and quite different from the engine connected to other types of load. Thus emphasis is on this marine application, rather than on description or theory relating to internal combustion engines. There are swarms of textbooks, technical papers, popular magazine articles, whatever your level of interest may be, that tell of engines in general and in most minute detail; I try to avoid repeating their messages, except as is necessary to give a coherent tale here.

Also, emphasis here is on principle of the application, rather than on practical detail. The reason is that installation details (for example) are expounded in handbook fashion by many engine builders, and are readily supplied to designers and builders confronted with a particular job or problem. Often, however, it is valuable to understand what is to be done in applying a marine engine; you just can't always do the right thing by blindly following rules, no matter how good the rules may be. To mention just one for instance, you may wish to innovate on occasion--to do something different--and handbook rules aren't likely to cover such situations. If you understand the principles, you will be able to estimate the effects of non-standard practice, and hopefully then make an intelligent innovation rather than a technical faux pas.

Of course, how-to-do-it details are also essential, and I have put them in to some degree. But where these can be found in other available publications, I have avoided repetition by merely citing the references.

This volume was prepared especially for short courses in small craft design at The University of Michigan. The intended reader is thus thought to be a practicing naval architect who wishes to upgrade his knowledge of small craft design principles. However, such a person may find some of the material too elementary for his level. The reason for this is that another potential readership is the regular student of naval architecture and marine engineering, probably at the sophomore or junior level, and he usually needs considerable introductory material.

The assistance of Detroit Diesel, Cummins Engine Company, Electromotive Division of General Motors, Caterpillar, Perkins Engine Company, Kielkhaefer Mercury, and Grafton Boat Company, was essential in preparing this work, and is acknowledged with appreciation.

CONTENTS

CHAPTER 1

INTRODUCTION

1.1 WHAT IS A MARINE ENGINE?

First, let's note that it is the common types of internal combustion engines that are treated here: the diesel, the spark-ignition ("gasoline"), and the gas turbine engines. Steam is omitted, for although it has a strong appeal to a small cult of nostaligiacs, and is sometimes mentioned as a future panacea for exhaust pollution, it has no real significance in the field. Wankel and Sterling engines, to name two examples, are other potential boat engines, but their trifling place at present does not outweigh the need for reasonable brevity here. So diesel, spark-ignition, and gas turbine.

Next, let's consider the attributes that you would expect of any engine in any application, and then look at the ones that seem especially appropriate in marine use. My list of these attributes is (no order of priority implied)

1. High efficiency
2. Low first cost
3. Quietness
4. Long life
5. Low maintenance cost
6. Reliability
7. Safety
8. Non-polluting wastes
9. Ease of starting
10. Speed and torque suitable for the load
11. Ease of control
12. Lightness and compactness

Some of these take on special significance for the general marine application, and even more significance for some particular marine uses. For example, lightness and compactness are obviously more important for boats than for stationary power plants, and yet more important for those boats requiring high speed and power. To take another example, efficiency is universally important because no one likes to pay more than necessary for fuel consumed, but it takes on special importance for a boat of long cruising range, because all fuel carried competes with payload. On the other hand, many short-range pleasure boats actually get very little use as compared to a municipal electric power plant, and in this comparison the efficiency of the marine engine is of much less importance.

Reliability if often cited as a prime marine attribute, and it is certainly to some degree more important at sea than in most shore uses.

Safety is important everywhere, but special problems arise with some engines in marine use. The foremost example is the gasoline-in-the-bilge explosion hazard that is present with engines that burn this fuel.

Each of the three engine types mentioned meets the general criteria to an adequate degree for many uses, including most marine uses, although each may have its special advantages and special problems in marine use. The gas turbine, for example, is outstanding in its lightness, but is relatively poor in efficiency, and requires special silencing. The explosion hazard with gasoline, used only by the spark-ignition engine, has been mentioned just above as an example of a special problem. But beyond these characteristics, which could be called inherent to the engines, there must be some special marine requirements, and marine problems.

Perhaps the first specifically marine consideration is the nature of the marine environment itself. This means several things to the engines, although the significance is not the same for each.

1. The corrosive nature of sea air. This doesn't mean much to the diesel and spark-ignition engines, but the gas turbine suffers considerably because air-borne sea salts coat its internals, consequently degrading performance as well as promoting corrosion.

2. The corrosive nature of the sea water. Since the surrounding water is the only practicable heat sink (air-cooled engines are used occasionally for small powers), its corrosive nature must be contended with in engine cooling systems. Even in fresh-water service, the water may be silt-laden, hard, or corrosive, and so cause similar problems.

3. The pervasive presence of water often means a danger of water in the fuel. This is especially true of ships that may bunker from barges, and that carry fuel in

tanks that must also carry water ballast at times. With small vessels that more typically fuel directly from shore, and carry fuel in an exclusive-use tank, the problem is essentially absent, save for the possibility of condensation in the tank.

The motions of a vessel, and the possibility of its operating at a list or trim, are also unique to marine use, although the problems arising from this source are minor.

The marine use is perhaps the only common one in which leaking fuel is trapped around the engine, and it is this feature that makes gasoline vapor a special hazard. In automotive or aircraft service, the vapors from leaking fuel are continuously blown away by the free flow of air under the vehicle. Not so with a boat, for the hull forms a gas-tight envelope below and around the engine, and in consequence careful measures must be taken in design and operation of fuel and ventilation systems to avoid the serious consequences.

A somewhat related problem exists even with non-volatile fuels: pumping oily bilge water overboard is becoming socially unacceptable, and even illegal.

The need for astern capability is shared with other mobile applications, though the means of achieving it are in some ways unique. For example, marine diesel engines, in particular, may be built to run in either direction. But this scheme is rare in contemporary small craft as a means of reversing; a reverse gear train or controllable-pitch propeller are much the more common today. In twin-screw vessels, however, it is desireable to have the screws turn in opposite directions, so that one engine of a pair may have a permanent "reverse" direction built in. But the same thing is often accomplished by a reverse gear train, or both engines are sometimes allowed to turn in the same direction. Consequently, the ability to run in both directions can't really be called an absolute requirement for marine use.

The load characteristics of the marine propeller, i.e. the relationships between torque and RPM, are unique (although much like those of other turbomachinery), but present only minor special problems to any of the internal combustion engines. However, the propeller is enough different from that most common of engine loads, the automobile, that the automotive transmission is not suitable for marine use*, and marine reduction gears with reverse train are available for all engines.

*Not completely hopeless, however, many a home-built boat has used an automobile engine with transmission attached.

You may have noticed by now the message that is taking shape: although there are special problems, and special attributes to be emphasized, nothing has been mentioned to suggest that there is a uniquely marine engine. If outboard motors and the monster "cathedral" engines of many foreign ocean-going ships are ruled out--and they are definitely outside the scope of this work--there isn't. Essentially all domestic engines are adaptations of engines that have been designed for some other use, or for a general range of uses. Automobile and truck engines account for most of the spark-ignition and diesel engines in small craft service. Gas turbines are still comparatively rare, so it is more difficult to make such a broad assertion, but those in use have originated in aircraft, truck, or industrial use.

The question of the section heading "what is a marine engine?" has now been answered in this fashion: it is an engine that has been adapted to marine use from any of the many other applications common to internal combustion engines. The obvious following question is "what is the adaptation process?", and this too has been answered, though in general terms. To summarize what has been said in this respect, adaptation requires protection of the engine from the marine environment, altering the cooling systems to use the marine cooling source, providing the marine transmission, and adding safety features (for spark-ignition engines). These steps, which are commonly referred to as "marinizing" (this is a gruesome word, but I can't supply a better one), are discussed in more detail at appropriate spots in later chapters. I should mention before going on, however, that some auxiliary systems, such as fuel system or exhaust system, may also have unique marine features. These have not been touched on so far, but will also be covered in later chapters.

1.2 INTRODUCTION TO THE ENGINES

1.21 Diesel Engines

Diesel engines for marine service are available in great variety over a power range from about 50 hp up to about 40,000 hp. The low end of this range is exemplified by propulsion engines for auxiliary sailboats, and by engines for electrical generators; the upper end obviously belongs to propulsion of the largest and most powerful ships. Naturally, in such an extreme range there are many subtypes of engines, classifiable in a number of different ways. Here, interest is limited to diesels that are commonly used to power small craft--pleasure boats of all kinds, and the smaller commercial craft, such as crewboats, and the smaller tugs, towboats, and fishing vessels. Most of these are powered by engines classifiable as "truck" or "high speed" types. Their external characteristics are typified by the following list:

1. Three, four, or six cylinders in-line, or eight, twelve, or sixteen cylinders in the V arrangement

2. Powers in the 50 to 1000 hp range

3. Rated speeds in the neighborhood of 2000 rpm

4. Specific weight in the neighborhood of 10 lb/hp

5. Specific fuel consumption of about 0.45 lb/hp hr

6. Either two-stroke or four-stroke cycle

7. Either turbocharged or naturally-aspirated

The "truck" appelation comes from their usual origin as truck engines, and the "high speed" from comparison with rated speeds of other diesels. Although the 2000 rpm cited is on the order of one-half that of a typical automotive spark-ignition engine, it is indeed high compared to the typical 400 to 900 rpm found in the diesel "medium speed" range, and the 100 to 150 rpm typical of the "low speed" range.

These engines are described in fine fashion, with profuse illustration, in the September 1965 issue of RUDDER magazine [1]. A complete listing of available models can be found in the current issue of the annual DIESEL & GAS TURBINE CATALOG [2].

Medium-speed engines also demand some attention here, for some of the larger "small" vessels, such as harbor tugs, commonly use engines that could be classified into this speed range. A typical example would turn about 800 rpm, produce about 800 hp, weigh 15 to 20 lb/hp, and have a slightly better fuel rate than that just quoted.

Both high-speed and medium-speed engines will nearly always be installed with a reduction gear between engine and propeller. Gear ratios typically fall in the range 2:1 to 6:1, and a reverse gear train is incorporated unless a controllable-pitch propeller is used.

1.22 Spark-Ignition Engines

Most people avoid wearing out their jawbones by calling these "gas" engines. Being a scholar, I cannot avail myself of this imprecise convenience; rather I will use hereafter the abbreviation "SI engine" when I weary of the full name.

These engines are sold under many trade names, but all originate as automotive or truck engines, and though considerable adaptation may have been applied, they remain basically the original engine. Their external characteristics are typified by the following list:

1. Four or six cylinders in-line, or eight or twelve cylinders in the V arrangement

2. Powers from about 50 to several hundred horsepower

3. Rated speeds in the 4000 to 5000 rpm neighborhood

4. Specific weight in the 5 lb/hp range

5. Specific fuel consumption of about 0.6 lb/hp-hr

6. Four-stroke cycle only

7. Naturally-aspirated only

The use of these engines is largely limited to pleasure craft, and so it is that the popular boating magazines are a good source of general information and description. In particular, the annual BOAT OWNER'S BUYING GUIDE [3] gives a complete listing of engines available, with power, RPM, weight, and sometimes price, being quoted for all engines.

1.23 Gas Turbines

The picture is somewhat different here because the gas turbine is a comparatively recent innovation, and one whose place in the marine propulsion picture is yet to be confirmed. Generalities are difficult, for rather than having a history of uncounted thousands of boats from which to draw them statements must be based on a few examples and many of the boats are experimental. A wide selection of engines is not available; what can be had can be discovered in the current issue of the DIESEL & GAS TURBINE CATALOG [2].

Several boats have been built using a gas turbine that is adapted from an aircraft jet engine. Power output is about 500 hp, weight one or two lb/hp, fuel consumption about 0.7 lb/hp-hr, and speed about 2000 rpm. Engines adapted from truck gas turbines have also been used, having about the same general characteristics.

1.3 FACTORS OF CHOICE AMONG THE ENGINE TYPES

Most small craft could be powered by either a diesel or an SI engine, and also perhaps by gas turbine if a suitable unit were available at the particular power needed. Let's neglect the gas turbine for the moment, however, and look at the factors of choice between the two old-timers. I will outline the general situation by quoting a summary by Wyland [4] following:

> Gasoline engines have a lower first cost, a higher specific fuel consumption, and burn a much more expensive fuel, and a fuel which is far more hazardous because of the explosive possibilities. They have the added complication of an ignition system.
>
> Diesel engines have a higher first cost, low specific fuel consumption, and burn a much cheaper fuel which does not present an explosion hazard. There is no ignition system, and they are reputedly easier to trouble shoot insofar as failure to start is concerned.
>
> There is as a rule no saving in cost due to the fuel economy of a diesel on the average yacht for the simple reason that few yachts are used in the continuous sense, as is the case with commercial vessels, and the savings in fuel would seldom offset the additional first cost. There is a real advantage in the reduced weight and smaller volume of fuel to be carried. On small craft the volume required for fuel often becomes of equal importance to the weight factor.
>
> Gasoline engines, as mentioned, are more complicated due to the addition of an ignition system, and starting or other troubles may be more difficult to track down or correct. Diesels should start promptly if fuel is available. However, leaks in fuel lines or components which will cause air in the system will not only prevent starting but can be laborious and difficult to track down. A good fuel line installation by the builder as well as a good integral system itself on the engine is of maximum importance.
>
> Nearly all diesels have a higher noise level than a gasoline engine, and most modern yacht engines are of the high revolution and high piston speed variety. As a result, a much more effective acoustical installation is required.
>
> Lubricating oil consumption is considerably greater than for a gasoline engine.
>
> On larger cruising craft today there is little choice except for diesels as they are about the only type available.

When a unique design is being made, each choice factor should be examined in light of its significance to the situation at hand. Wyland's summary, however, should give you a good idea of why the choice should go the way it does in most cases.

As previously noted, the gas turbine is still something of an outsider. The basic reason is its comparatively high first cost, as well as its poorer fuel consumption. It also is somewhat handicapped by the salt ingestion problem, and by the need for comparatively large inlet and exhaust ducting. Its great advantage is its outstanding lightness and compactness, an advantage that is most meaningful in a light-weight boat requiring high power. In craft where high speed is essential, such as military vessels, hydrofoil boats, and offshore crew boats, this advantage may offset the drawbacks.

1.4 WHAT THE DESIGNER SHOULD KNOW

It might truly be said that the designer of a motor vessel of any kind should know the greatest amount possible about all aspects of its machinery. Fine if he does, but it's impracticable to ask that everyone be a universal expert. He might indeed get by with knowing absolutely nothing about machinery if he could always work closely with a machinery specialist whose knowledge would fill the gap. But let's aim more at the middle, and speak of the designer who will be responsible for the entire design--he is the person at whom this work is principally aimed. He is not going to be that universal expert, for he is not going to design the engine, and he will use the advice of specialists as necessary; on the other hand, he is not going to buy the engine as a Black Box and say "put it in". Let's see.

This designer needs the knowledge to cope with three broad problems:

1. He must select the engine. On the face of it, this sounds simple, but it involves several things that may not be: he must always determine the most appropriate rating for the engine, and he may need to make careful evaluation of the relative merits of two or more engines of different type or manufacture.

2. He must match the engine to the propeller. This involves best choice of RPM, the assurance that off-design conditions will be satisfactory for engine and propeller, and possibly the specification of operating programs (especially if a controllable-pitch propeller is used) for the propulsion plant.

3. He must design the supporting functions. Exhaust and intake will always be an important responsibility, as well as the drive-line components between engine and propeller. Likewise, parts of the fuel, lube oil, cooling, and control systems may not be furnished with the engine, and so require his attention.

This volume is intended to treat these problems, all of which might be broadly described as application of the internal combustion engine to small craft propulsion. Application it is, throughout, but the message is largely one of principles of application, rather than of details. Much of the application data that will be needed for a particular job is of a handbook nature, and is much better supplied by the application instructions of the engine builder. The designer is advised to make full use of that source, but to base his use on the understanding that the following chapters hope to give him.

1.5 REFERENCES

1. The RUDDER Magazine, September 1965

2. DIESEL & GAS TURBINE CATALOG (annual)

3. BOAT OWNER'S BUYERS GUIDE (annual), Yachting Publishing Co.

4. Gilbert Wyland "Selection of Power Plants for Yachts," MARINE TECHNOLOGY, July 1968

CHAPTER 2

PROPELLER CHARACTERISTICS

2.1 INTRODUCTION

To understand the marine propulsion engine you must also understand the propeller, and vice versa, for these disparate units must work together in an intimate driver-load relationship. They form a single machine whose input is fuel and whose output is thrust. The behavior of this machine is determined by the interaction of propeller characteristics and engine characteristics. To cite an example, it might be asked "what happens to power and RPM if growth of barnacles increases hull resistance coefficient by X percent?" To answer, a response function must be constructed from propeller coefficients of thrust, torque, and advance, all in combination with the torque characteristic of the engine; the question cannot be answered from consideration of propeller behavior alone, nor of engine behavior alone.

The engine and propeller must be properly matched in the design process so that optimal performance will be obtained from the engine-propeller combination. In simplest terms this means selecting propeller pitch so that the engine can turn at its rated RPM, and hence develop its rated power. There is sometimes a conflict here with the concurrent desire for highest possible propeller efficiency, but the availability of a wide range of reduction gear ratios usually provides an easy solution to the possible dilemna. But every possible gear ratio that might be wanted is not available in stock gear sets, so that sometimes matching involves trade-offs between what seems best for the engine and what seems best for the propeller.

Somewhat more subtle factors are also involved in matching. For example, deterioration of hull surfaces (barnacles, you know), bad weather, changes in hull draft, etc, cause the designer's intended point of matching to shift. He may therefore find it advisable to make some compromises in his original choice so that the conditions of matching will be the best possible in service.

Whatever the engine-propeller problem to be analyzed, propeller characteristics and engine characteristics both must be understood by the designer. Propeller characteristics might properly be left to other sources, for after all, the topic here is engines. However, the characteristics referred to are those seen by the engine; as will be developed in the next section, this means fundamentally the torque-RPM characteristics. Unfortunately, most texts that you might consult for this information neglect it altogether. PRINCIPLES OF NAVAL ARCHITECTURE [1], our standard text in naval architecture, is an example. Therefore, the present chapter is included in a book on engines, its purpose being to expound what needs to be known about propellers to understand their interaction with the engines that drive them.

2.2 DRIVER-LOAD RELATIONSHIPS

This section sketches the fundamental relationships between drivers and loads in order to form a simple theoretical background for the more particular discussions of engine and propeller interactions.

The driver supplies energy to the load. You may equally well speak of it as supplying power, power being the rate at which the energy is supplied. According to the conservation of energy principle, the energy imparted by the driver must equal that absorbed by the load. (If there are energy losses along the transmission link between the two, the principle states driver energy = load energy + loss energy. Further discussion in a subsequent paragraph.) Further, you will find that any form of energy transmission is characterized by two factors, an intensity and a flux (= flow or current). One familiar example is electrical energy, for which these factors are voltage and current. The great majority of marine propulsion transmissions are mechanical--by rotating shaft. Here the two factors are torque and rotational speed (usually called "RPM" after its common units, revolutions per minute). This can be expressed in equation form by

$$P = \frac{2\pi\ Q\ N}{33000} \tag{2.1}$$

where

P = horsepower

Q = torque, ft-lbf

N = RPM

It can further be said that a rotating shaft must have the same RPM at both ends [2]. If power must be the same at both ends, then equation (2.1) shows that torque likewise must be the same for both driver and load. (What about reduction gears? See later paragraph.)

An immediate conclusion is that the vital characteristic for analysis of engine-propeller interactions is the torque-RPM characteristic of each unit. Consider this: none of the many properties, parameters, dimensions, etc, that characterize a propeller can mean a thing to the engine save as they affect the torque required to turn the propeller at a chosen RPM. And likewise looking in the opposite direction.

Equation (2.1) shows that if torque and RPM are known, or indeed any two of the three factors determines the third. Thus it is that a power-RPM relationship is equivalent to a torque-RPM relationship. Because it is often more convenient or meaningful to speak of power rather than torque, the power-RPM alternative is often used in lieu of torque-RPM, here and elsewhere.

Figure 2.1 shows an example plot of a driver torque-RPM relationship and a load torque-RPM relationship. If RPM must be the same for both units, and torque the same for both, then their common operating point--the only place where they are in equilibrium--is at the intersection of the curves. Here is introduced a seemingly trite, but nonetheless vital, instruction for finding the operating point of driver-load pairs: <u>plot the respective curves</u> on a common plane. If they are linked mechanically, plot the torque-RPM (or power-RPM) curves; if electrically, plot the current-voltage curves; and so on for the other possibilities. Plot the curves. I emphasize this message only because so many students seem to resist it.

The intersection in Figure 2.1 is an example of <u>stable</u> equilibrium. If a momentary influence upsets equilibrium (e.g. pitching of the hull causes the propeller to over speed), equilibrium will be restored at the same point once that influence is removed. It is also possible to have <u>unstable</u> equilibrium; this can be pictured simply by reversing the labels on the two curves of the figure. In this case, equilibrium would not be restored if it were ever lost. It is left to the reader to ponder the figure a moment to see the why of both these cases.

Now for more complex situations. The assembly may consist of more than a single driver and a single load. An example that might be found in marine propulsion is electric drive: an engine drives a generator that supplies electrical power to a motor that drives the propeller. The generator, for instance, is a load as seen by the engine and a driver as seen by the motor. Another common situation finds parallel drivers handling a single load. Two engines geared to the same propeller shaft is an example. Whatever the combination, and no matter how complicated, the principles previously stated still apply; energy is, of course, still conserved, and there will be an equilibrium point found by the intersection of a driver curve and a load curve. Finding this point in a complex situation is a process of step-by-step reduction of the complexity by combining the curves representing adjacent units. If two engines drive the same propeller, then combine their torque curves by addition, since it is apparent that they must have the same

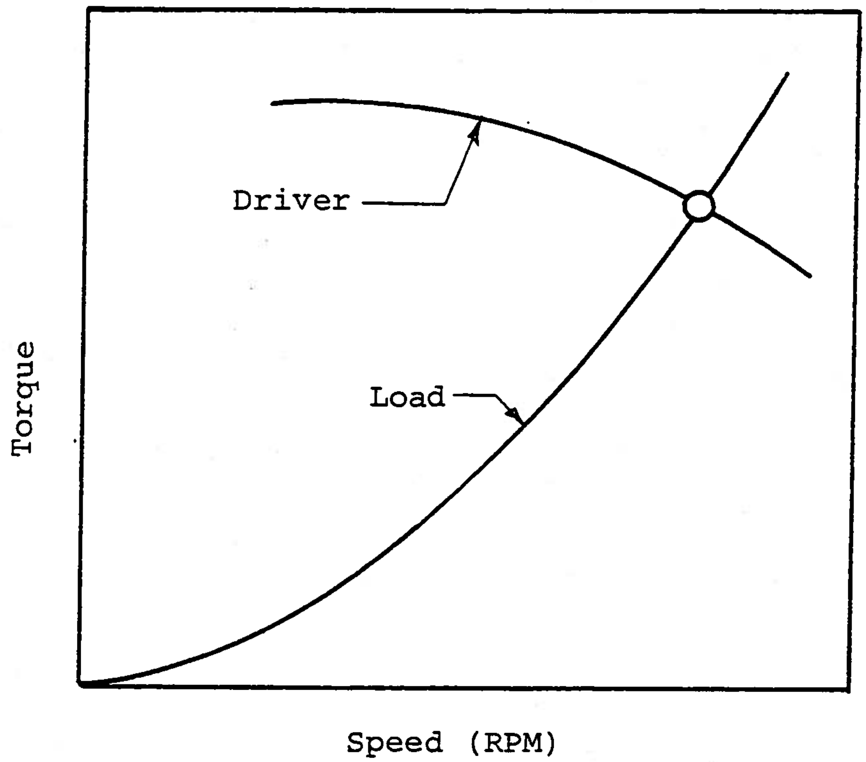


FIGURE 2.1 Driver and Load Equilibrium

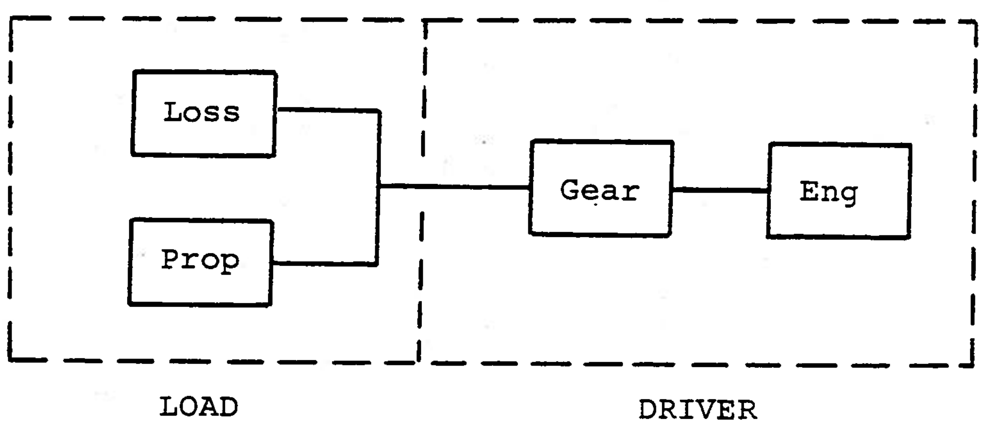


FIGURE 2.2 Driver and Load Combinations

RPM with respect to the propeller, and that the torque received by the propeller is the sum of the two engine torques. Likewise if an engine drives two loads, say a propeller and an attached generator. The two loads must have the same RPM with respect to the engine, and the sum of their torques must be the torque supplied by the engine. So add torques at common RPMs and plot the resulting sum curve against the engine curve. More detailed examples of the technique appear in later chapters.

The preceding paragraph suggests how to treat a reduction gear and friction losses in the typical marine drive train. Look at Figure 2.2. It represents in block form an engine and propeller with these two additions. I have arbitrarily lumped losses with the propeller as part of the load, and arbitrarily lumped the gear with the engine as driver. The output RPM of the gear is engine RPM divided by the gear ratio; its output torque-RPM curve of the combined engine-gear driver is readily constructed from the engine torque curve. Likewise the combined load curve is constructed by subtracting friction torques from propeller torque. At the arbitrary boundary between driver and load, torque must equal torque and RPM must equal RPM as before. The nuisance complication of different speeds for engine and propeller is neatly disposed of by the prior combination of engine and gear into a single unit.

2.3 DEFINITIONS

As further groundwork for subsequent material, this section defines terms frequently used in discussions of marine propulsion.

Effective horsepower (EHP or P_E) is the power required to move the vessel at speed V against resistance R:

$$P_E = \frac{R\ V}{550} \tag{2.2}$$

for R in lbf, V in ft/sec. It is sometimes referred to as towrope horsepower, since it is the power required to tow, and thus is the power predicted by a model resistance test.

Thrust horsepower (THP or P_T) is the power delivered to the water by a propeller producing thrust T and moving at speed of advance V_A:

$$P_T = \frac{T\ V_A}{550} \tag{2.3}$$

for T in lbf, V_A in ft/sec.

The relationship between P_T and P_E is conventionally represented as follows:

$$R = (1 - t)T \tag{2.4}$$

$$V_A = V(1 - w) \tag{2.5}$$

where t is the thrust deduction, and w is the wake fraction.

Therefore $$P_T = \frac{R\ V}{550}\frac{(1 - w)}{(1 - t)} = P_E\frac{1 - w}{1 - t} = \frac{P_E}{\eta_H} \tag{2.6}$$

The definition of hull efficiency (η_H) is implicit in (2.6)

Delivered horsepower (P_D or DHP) is the power delivered to the propeller by its shaft. It is, of course, greater than P_T because of propeller inefficiency. Thus

$$\eta_P = \frac{P_T}{P_D} \tag{2.7}$$

Propeller efficiency (η_P) is usually regarded as being the product of two terms, the open water efficiency (η_O) and the relative rotative efficiency (η_R), the former being efficiency predicted for the propeller operating far from the disturbing effects of a hull, and the latter being a correction factor to account the hull influence.

Now relating P_E to P_D via (2.6), (2.7), and the preceding discussion:

$$P_E = \eta_H\eta_O\eta_R P_D \tag{2.8}$$

The product $\eta_H\eta_O\eta_R$ is called the quasi-propulsive coefficient (QPC) (sometimes quasi-propulsive efficiency).

Shaft horsepower (P_S or SHP) is frequently used almost synonymously with P_D. It is, however, power delivered to the propeller as measured inboard of the stern tube, and so is larger than P_D by the amount of stern tube losses. The engine builders' definition of P_S is the power as measured at the reduction gear output coupling, and thereby agrees with the standard definition only if there are no intermediate bearings. When the propeller is driven through an inboard-outboard drive, P_S is defined as the power delivered to the propeller, and in this case is, therefore, the same as P_D.

The ratio of P_E to P_S is known as the propulsive coefficient (PC) (or propulsive efficiency).

Brake horsepower (P_B or BHP) is the power delivered by engine, usually thought of as being measured at the engine flywheel.

A word about notation: although P_B, P_E, etc. are convenient symbols for formula use, BHP, EHP, etc. are generally used hereafter to conform to more common usage.

2.4 FIXED-PITCH PROPELLER CHARACTERISTICS

Design or selection of a propeller for a particular application has many ramifications that are well beyond the scope of this work. I shall have to assume that the reader is acquainted with propellers, or can avail himself of another source, such as PRINCIPLES OF NAVAL ARCHITECTURE [1]. The problem of interest here is choosing the rated RPM, which usually depends on choice of propeller pitch, and this is to be touched on in Chapter 6. For the moment it is assumed that all is settled concerning the propeller, and that it is its behavior that is of interest; as developed in Section 2.2, "behavior" is to be interpreted as the relationship between torque and RPM.

A good starting point for this development is the traditional K_Q, K_T versus J representation of propeller performance. These are nondimensional coefficients, the torque coefficient (K_Q), the thrust coefficient (k_T), and the advance coefficient (J). They are defined by

$$K_Q = \frac{Q}{\rho\ D^5 N^2} \tag{2.9}$$

$$K_T = \frac{T}{\rho\ D^4 N^2} \tag{2.10}$$

$$J = \frac{V_A}{N\ D} \tag{2.11}$$

where Q is torque, T is thrust, ρ is density of water, D is propeller diameter, N is rotational speed, and V_A is propeller speed of advance. Units are chosen to make the coefficients non-dimensional.

Open water efficiency (η_O) is usually presented with these parameters, and is related to them by

$$\eta_O = \frac{J\ K_T}{2\pi\ K_Q} \qquad (2.12)$$

A typical plot of K_T, K_Q, and versus J is given in Figure 2.3.

The properties of a range of propellers of a particular series are frequently represented by B_P-δ and B_U-δ charts, and you may have used these in picking the principal properties of a propeller. If so, it is convenient to calculate K_Q, K_T, and J by the following relations:

$$K_Q = \frac{1.89 \times 10^7}{\frac{\rho}{g_o}} \frac{B_P^{\ 2}}{\delta^5} \qquad (2.13)$$

$$K_T = \frac{1.173 \times 10^6}{\frac{\rho}{g_o}} \frac{B_U^{\ 2}}{\delta^4} \qquad (2.14)$$

$$J = \frac{101.3}{\delta} \qquad (2.15)$$

where ρ is lbm/ft^3 and g_o is 32.17 lbm·ft/lbf sec^2. In using these, B_P-δ or B_U-δ combinations are read along a constant pitch ratio line on the appropriate chart.

Development from these coefficients to the torque-RPM or power-RPM characteristics depends in part on assumptions about the hull that the propeller is driving. A convenient and frequently used set of assumptions forms the basis of the so-called "propeller law." They are

1. Resistance of hull is proportional to the square of its speed through the water.

2. Wake fraction and thrust deduction are constant over the vessel's speed range so that thrust is also proportional to speed squared.

If thrust is proportional to speed squared, and D is constant, as it must be for a particular propeller, then from equations (2.10) and (2.11).

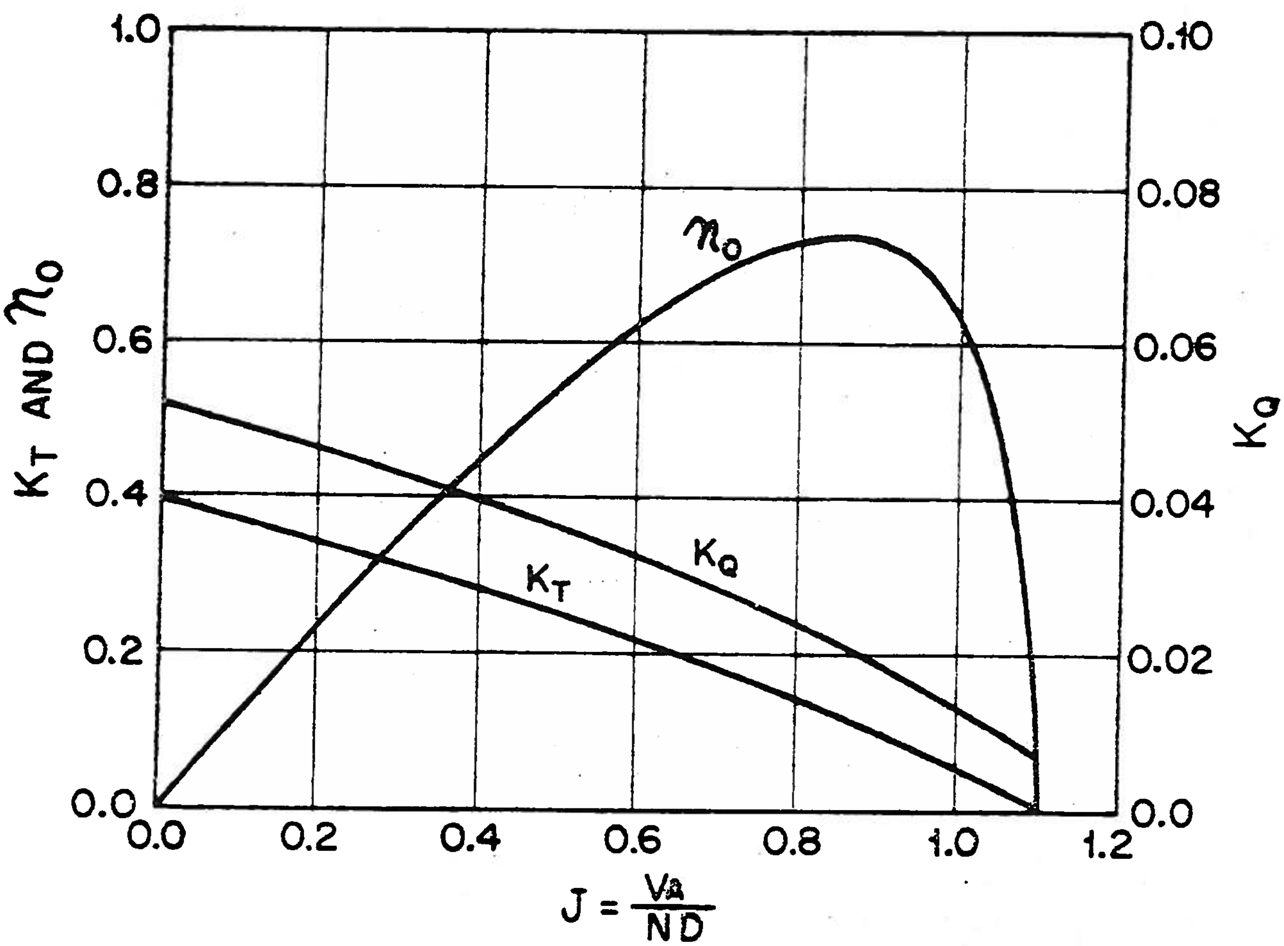


FIGURE 2.3 Typical Relations of Thrust Coefficient, Torque Coefficient, and Open-Water Efficiency to Advance Coefficient, Fixed-Pitch Propeller

$$K_T = C_1 \frac{V_A{}^2}{N^2} \qquad (2.14)$$

$$J = C_2 \frac{V_A}{N} \qquad (2.15)$$

where C_1 and C_2 are convenient constants. These can be combined into

$$K_T = C_3 J^2 \qquad (2.16)$$

where C_3 is another constant.

A glance at Figure 2.3 shows that the K_T - J line is described approximately by $K_T = a + b\,J$ (a and b are constants). Since the propeller can operate only along this line, it follows that $K_T = C_3 J^2$ can be satisfied only at a single point. Therefore, K_T, K_Q, and J all remain constant as vessel speed varies, and equations (2.14) and (2.15) show that RPM must be proportional to vessel speed. Equation (2.9) also shows that torque must vary in proportion to RPM squared. Since power is proportional to the product of torque and RPM, it must be proportional to RPM cubed. This, then is the "propeller law," which can be summarized as follows:

1. Propeller RPM is proportional to vessel speed.
2. Propeller torque is proportional to RPM squared.
3. Propeller power, both absorbed and delivered, is proportional to RPM cubed.

If all were as simple as the propeller law would pretend, the deed would be done right here--it has now been established that the propeller torque-RPM relationship is a parabola. Unfortunately, the propeller law is a good approximation only for displacement vessels, and for them not especially good at high speed-length ratios (i.e. above 1, speaking roughly). For a planing boat, it is usually quite erroneous, mostly because the assumption of resistance proportional to square of speed does not hold through the planing region. Figure 2.4 (from [1]) illustrates with resistance curves for several boat types. The "hump" that occurs typically as planing begins is apparent. Figure 2.5 (from [3]) gives thrust, SHP, RPM, and torque curves for an esample planing boat. Again, the hump and the distinct departure from propeller law are evident.

So the propeller law isn't much good for the planing powerboat. Nonetheless, it is useful for discussion purposes because of its simplicity. Most <u>principles</u> to be discussed

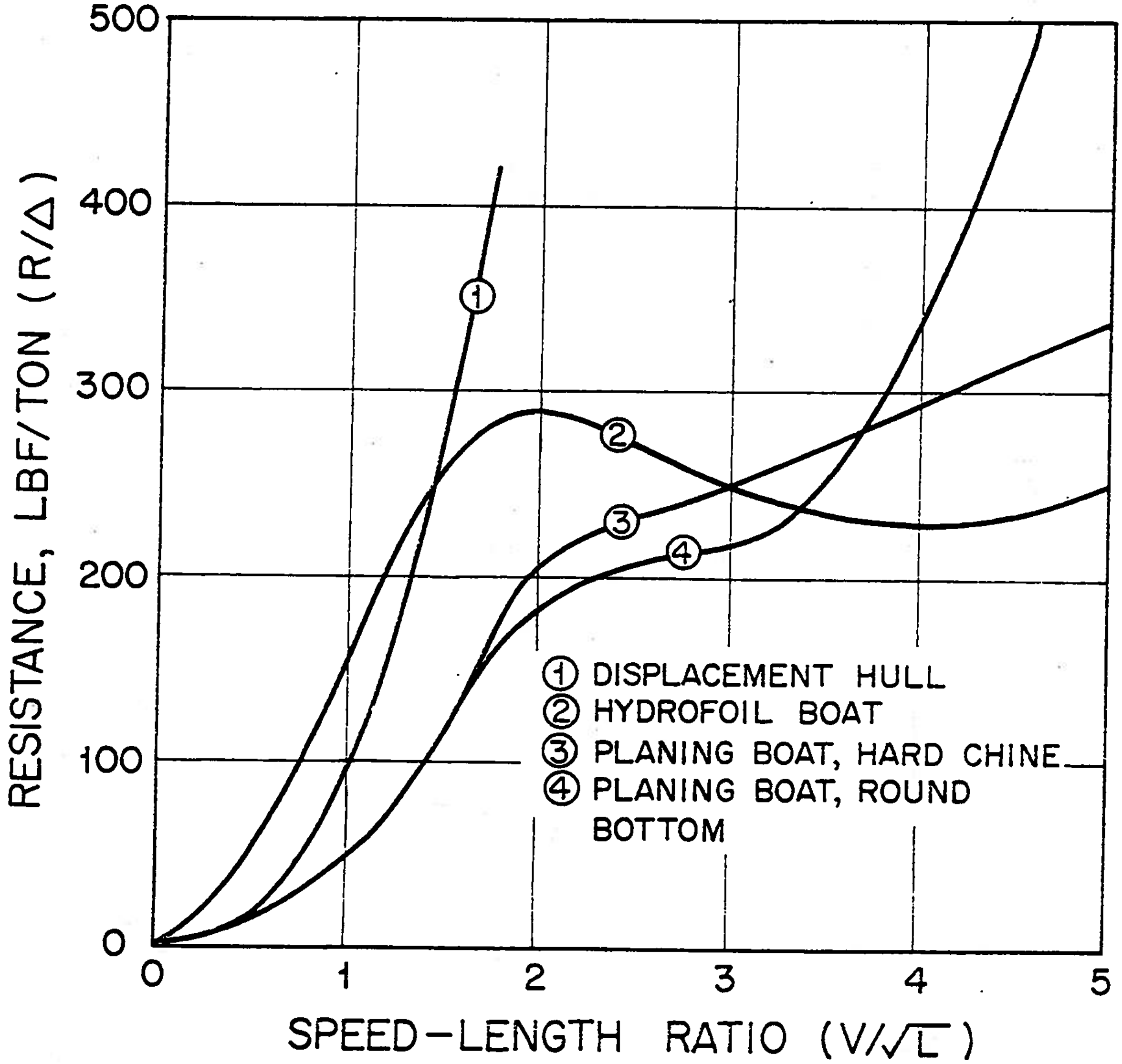


FIGURE 2.4 Typical Resistance Curves for Several Hull Types

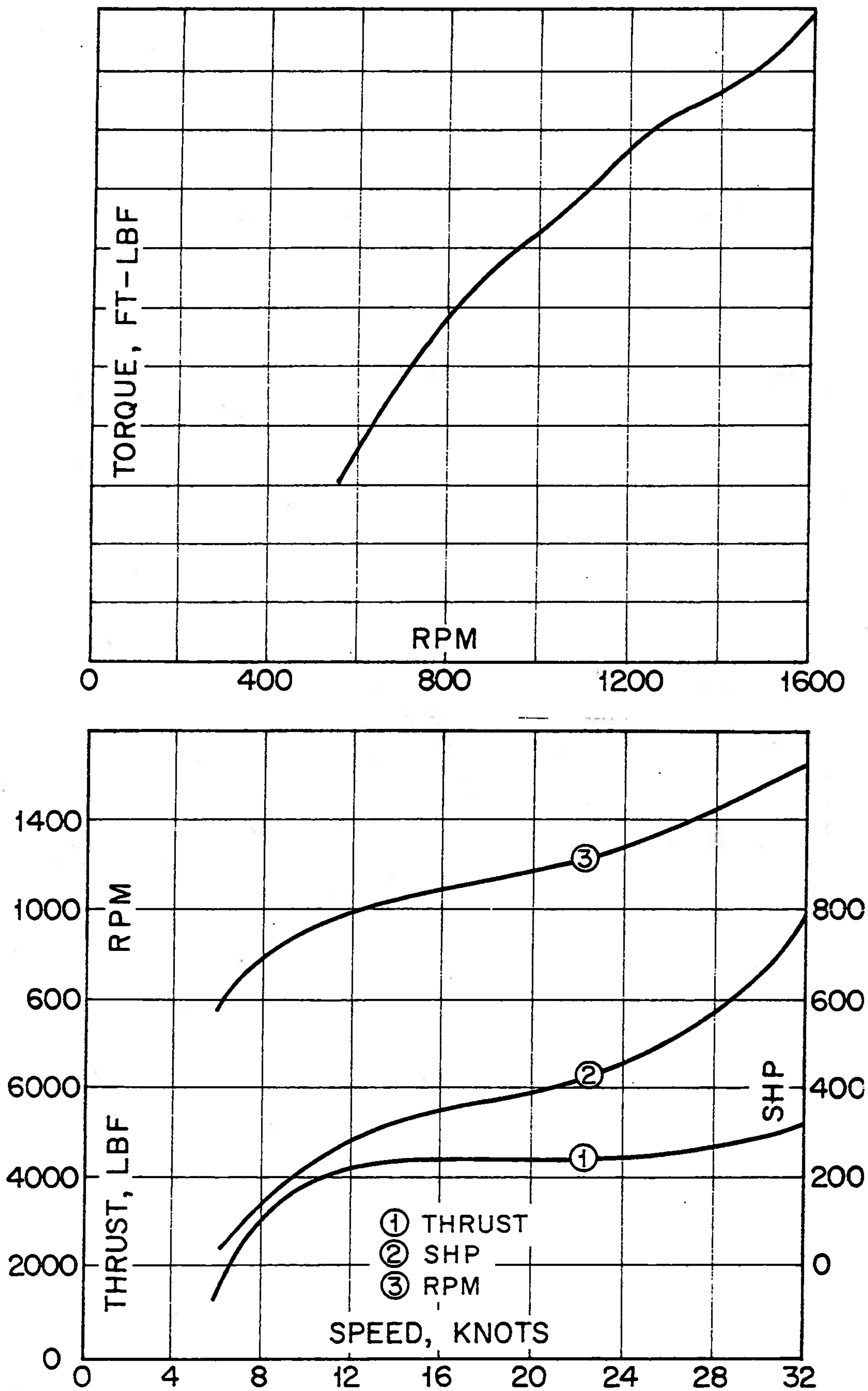


FIGURE 2.5 Typical Thrust, Power, and RPM Curves for a Planing Boat (from Reference[3])

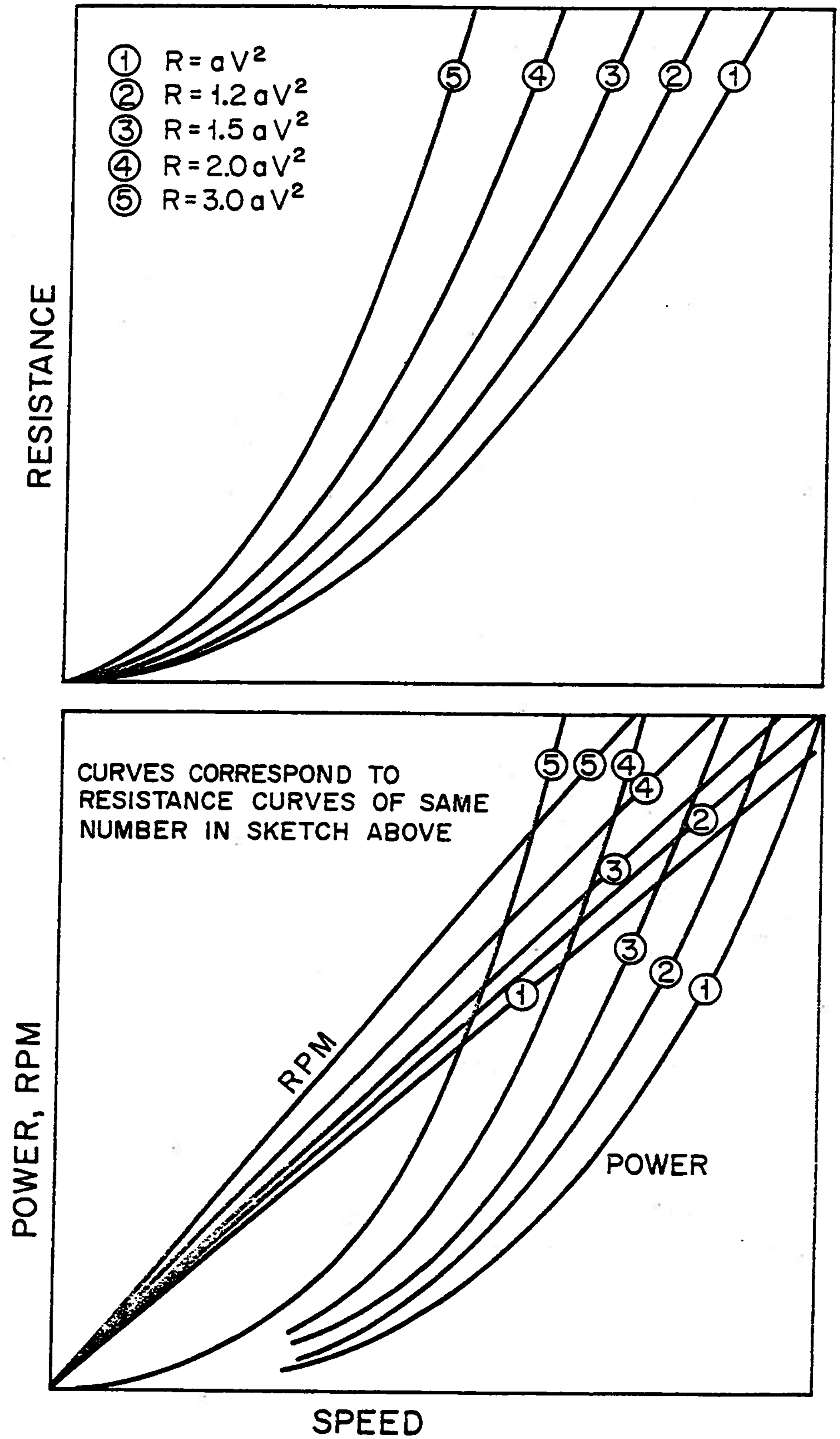
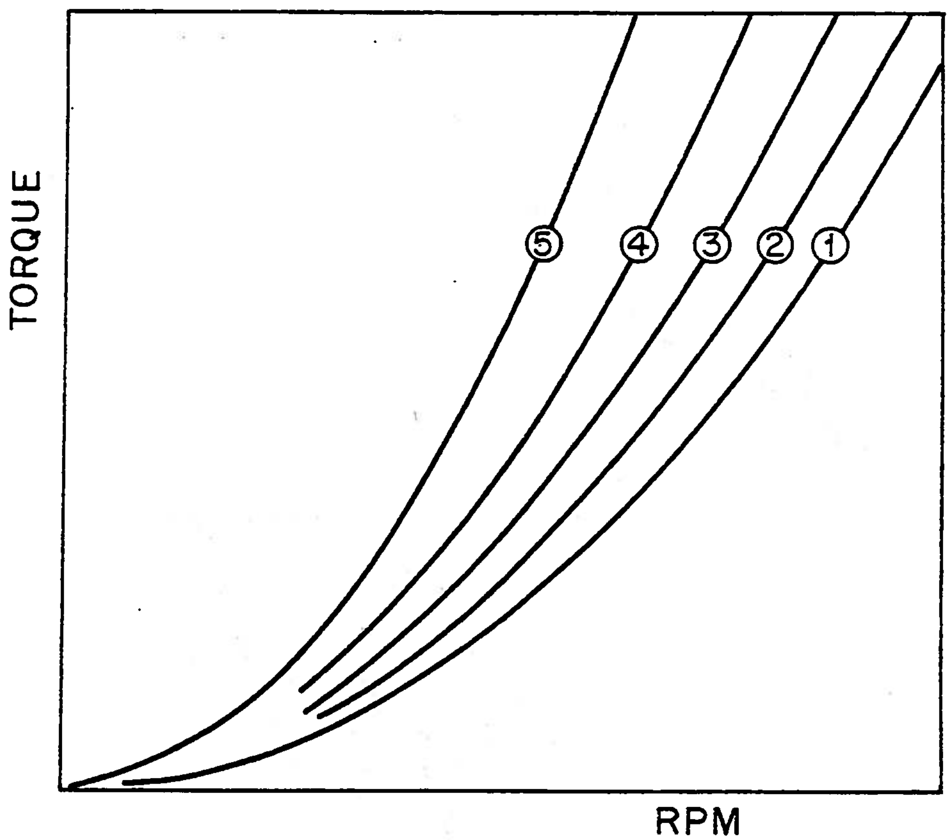


FIGURE 2.6 Resistance, Power, and RPM Curves According to Propeller Law



Numbers on the Curves Match Those at Figure 2.6

FIGURE 2.7 Torque-RPM Curves Corresponding to the Power Curves of the Preceding Figure subsequently are not dependent on the shapes of the curves used to illustrate them. You are thus to find the square and cubic curves frequently used in the following chapters, with modifications required with more realistic curves pointed out as appropriate.

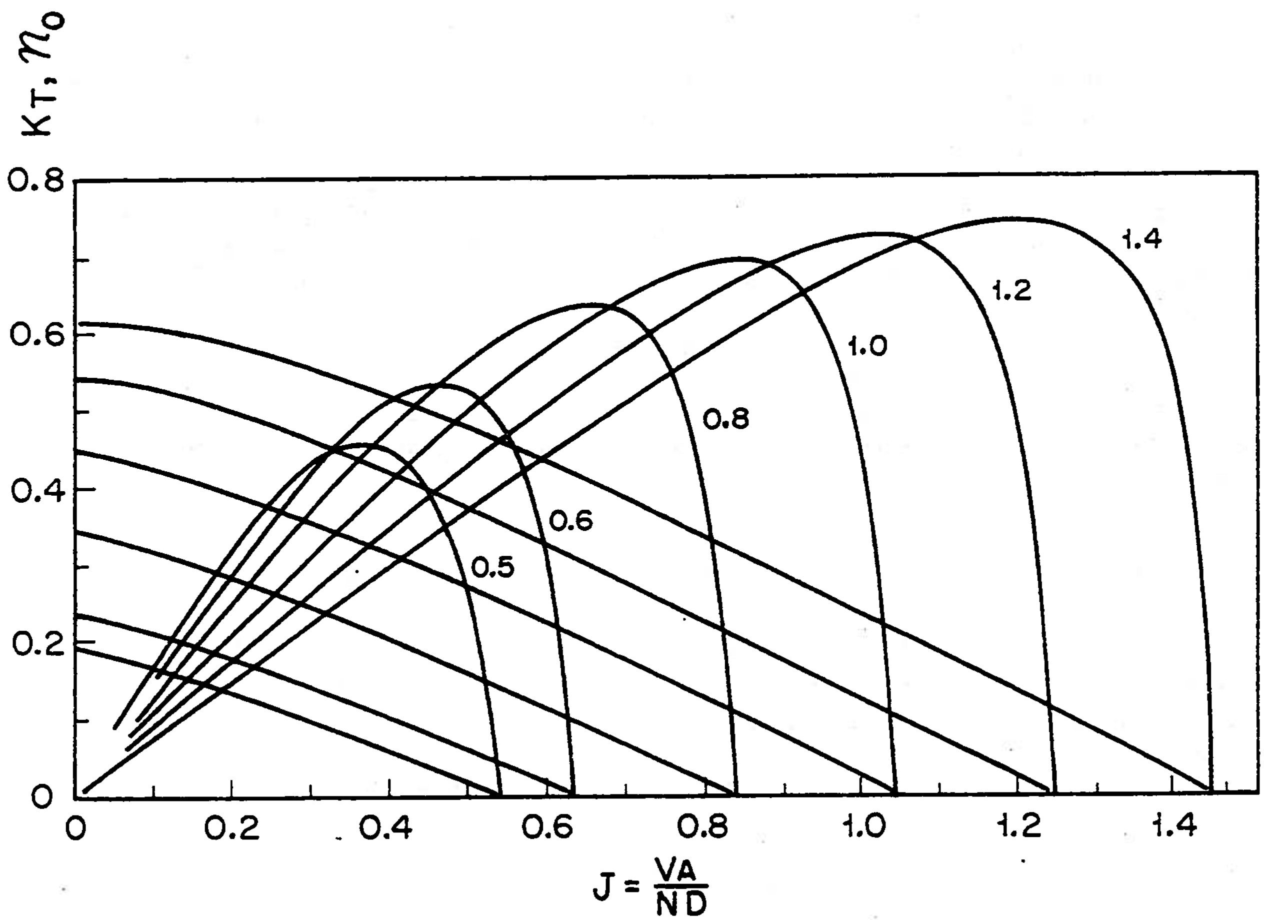
It should be emphasized that the curves of Figure 2.4 and 2.5 do not show the changes of resistance et al that occur when speed variations are due to external factors such as draft, trim, sea state, wind, or hull surface roughness. If one or more of these factors acts to increase resistance at a particular speed, the resistance-speed relation, for instance, remains parabolic, but the factor of proportionality changes so that the parabola becomes a different parabola. Torque and power are likewise different in the same fashion; the curves are the same shape, but these curves are, in effect, shifted to new positons. Figures 2.6 and 2.7 illustrate this point, using the propeller law. In the first sketch, parabolic resistance curves are shown for five different resistance coefficients. Corresponding power, RPM, and torque curves follow.

2.5 CONTROLLABLE-PITCH PROPELLER CHARACTERISTICS

At any particular pitch, the K_T, K_Q, and η_O versus J curves for a controllable-pitch propeller resemble those of a fixed-pitch propeller of the same pitch ratio. The complete plot, therefore, resembles a superposition of enough of these curves on the same sheet to cover the pitch ratios of interest. A typical example is given in Figure 2.8. The K_Q curves are omitted for clarity.

If the propeller law can be assumed to hold, then for each pitch ratio there is a cubic power versus RPM curve and a parabolic torque versus RPM curve. A set of such power versus RPM characteristics, constructed from Figure 2.8, is given in Figure 2.9.

A set of actual power-RPM characteristics for a displacement vessel is given in Figure 2.10, principally to show the slight difference in efficiency contours from the propeller-law version of Figure 2.9.



K_Q Curves are Omitted to Avoid Confusion from Too Many Lines. Note that K_Q can be Calculated from K_T and η_o.

FIGURE 2.8 Typical Characteristic Curves for a Controllable Pitch Propeller

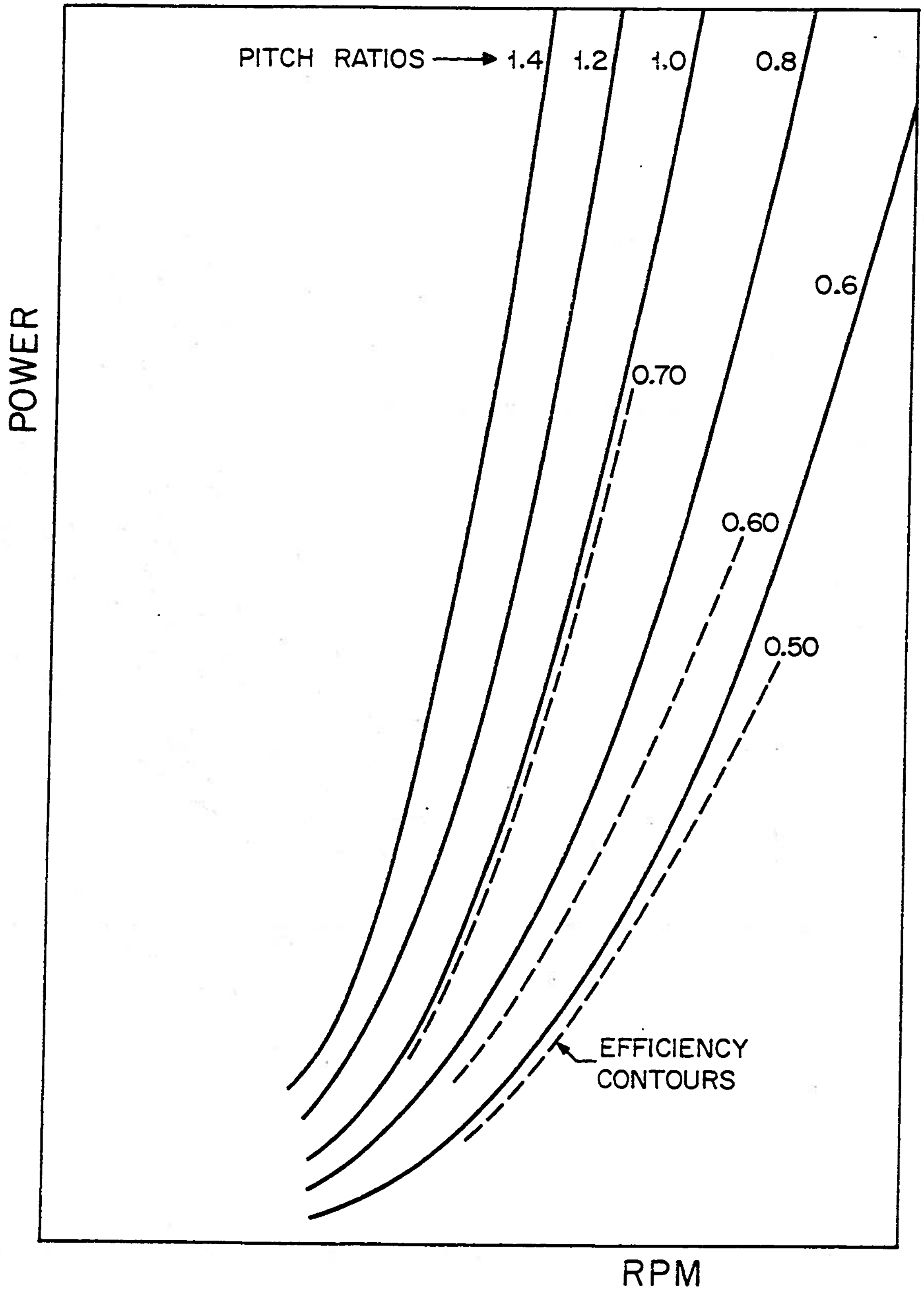


FIGURE 2.9 Power-RPM Characteristics of a Controllable-Pitch Propeller as Calculated from the Curves of Figure 2.8

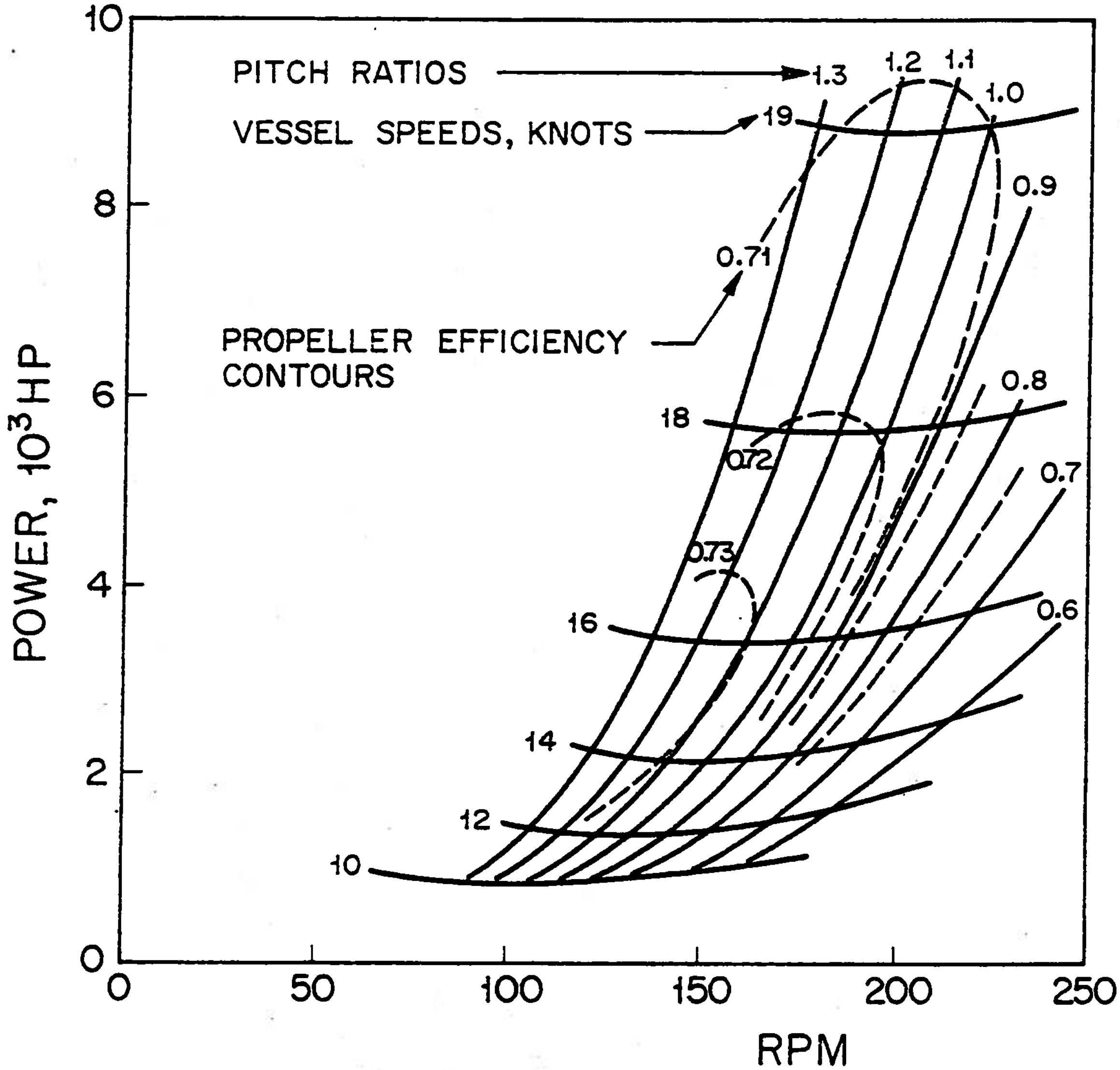


FIGURE 2.10 Power-RPM Characteristics of an Actual Controllable-Pitch Propeller

2.6 REFERENCES

1. Society of Naval Architects and Marine Engineers, Principles of Naval Architecture, 1967

2. Arnold Superheat, Assorted Lecture Notes, Department of Naval Architecture and Marine Engineering, University of Michigan

3. Donald L. Blount, "Smooth Water Powering Characteristics of an Experimental LDVP(T)", Report 2283, Naval Ship Research and Development Center

CHAPTER 3

THE MARINE DIESEL ENGINE

3.1 INTRODUCTION

The diesel engine receives attention first. This is not to imply that it is more important than the spark-ignition engine; each is more important in its own place, and the same might be said for the gas turbine in its presently rather limited corner of the marine field. But two factors justify putting the diesel first, and giving it more attention that the SI engine throughout:

1. Most contemporary readers are likely to be familiar with the SI engine because of the ubiquitous presence of the automobile, but there is no such every-day close-at-hand acquaintanceship with the diesel.

2. The builders of marine diesel engines are much more liberal with engineering data on their products. Several publish installation guide books with details of exhaust flows and temperatures, cooling water requirements, mean effective pressures, piston speeds, specific fuel consumptions, etc. The marine SI engine industry just doesn't follow this policy. Naturally enough, I put most of my attention to the areas where the information is available.

Now I repeat that the purpose here is not description of engines, nor treatment of engine theory or design principles. All of these things are thoroughly covered elsewhere. Try [1], [2] for description of marine diesels, with [2] being especially devoted to small craft engines. There are many general textbooks on internal combustion engines and diesel engines--so many that I will not try to give an exhaustive list, but will only mention two [3], [4] that I have found useful. The general theme here is marine application, so what <u>is</u> found in this chapter is a modicum of basic description material, definitions of essential terms, some generalities that help to understand the marine application of the diesel engine, and some particular things that are needed in a discussion of applications.

3.2 FUNDAMENTAL CONSIDERATIONS

The diesel is a reciprocating internal combustion engine. It is distinguished from other engine types of this class by the method of introducing and igniting the fuel: the compression ratio is high enough that the corresponding compression temperature will ignite a spray of liquid or gaseous fuel. Fuel is thus sprayed into the cylinder by an <u>injector</u> (the underlining serves here to call attention to important terms being introduced) when the piston is near top center. It ignites almost instantaneously, thereby releasing energy to accomplish the power stroke of the piston. This characteristic feature leads to the name compression ignition engine being used as an alternative to diesel, but it is an obsolescent term.

The engine will operate on either a two-stroke cycle or a four-stroke cycle. If the former, the spent cylinder gas is exhausted as the piston nears the bottom of the power stroke, hurried along by a blast of air (the scavenging process) from a scavenging blower. The exhaust openings and scavenging inlets are closed near the beginning of the upward stroke of the piston. The air thus trapped in the cylinder constitutes a fresh charge that is compressed on the ensuing compression stroke. Near the end of the compression stroke, the fuel is injected, and the power stroke begins again. The whole cycle is completed in two strokes of the piston. If the four-stroke cycle is used, exhaust is accomplished by the piston on an upward stroke (the exhaust stroke) following the power stroke, and intake of a fresh charge is accomplished by the piston on a downward stroke (the intake stroke) following the exhaust stroke. The compression stroke and power stroke then follow. No scavenging blower is required, since the piston itself provides the pumping action.

The engine may be either naturally aspirated or supercharged (or turbocharged; see subsequent explanation). If the former, intake or air to the cylinders is accomplished as described in the preceding paragraph; the pressure of the air trapped in the cylinders is essentially atmospheric. If supercharged, some form of compressor furnishes high pressure air to the cylinders during the intake process (though the action of the piston itself is the same as previously described). High pressure means high density which means more fuel can be burned per cycle which means more power from a cylinder of given size. The compressor is nearly always driven by a gas turbine whose working fluid is the hot exhaust gas from the engine. The turbine-plus-compressor is a turbocharger, hence the term turbocharging, which is more commonly used than supercharging.

If the turbocharged engine is two-stroke, the turbocharger may replace the scavenging blower, or the blower may be retained to provide scavenging when exhaust gas energy is too low to run the turbocharger adequately (e.g. low speed running). The blower is always driven mechanically from the engine.

The four-stroke engine cylinder is provided with one or more intake valves and one or more exhaust valves, thus functioning to admit and exhaust the cylinder charge. Some two-stroke engines use no valves: exhaust is accomplished through ports in the cylinder wall that are uncovered by the piston at the proper time, and intake is accomplished through similar ports that are also uncovered by the piston shortly after the exhaust process begins. Other engines substitute an exhaust valve for the exhaust ports. The valve is located in the cylinder head, and is driven by a camshaft, just as are the valves of the four-stroke engine.

Fuel is sprayed into the cylinder by an injector, which is essentially a very fine nozzle, combined with a valve and fuel pump (sometimes). If the fuel pump for the cylinder is not part of the injector, it will be contained in a separate housing with pumps for the other cylinders. Or a single pump may serve all cylinders via a distributor.

The engine must be provided with several auxiliary systems for successful operation. Cooling water must be circulated through jackets around the cylinders, through the cylinder head, and sometimes through the pistons. Lubricating oil must be provided under pressure to many points, and it must be cooled. A governor is always provided to limit speed to a safe value, and often it controls speed in the working range as well. Starting must be accomplished from an external source of power; an attached motor (electric, hydraulic, or pneumatic) is typically used, though the larger engines are cranked by injection of compressed air into the cylinders.

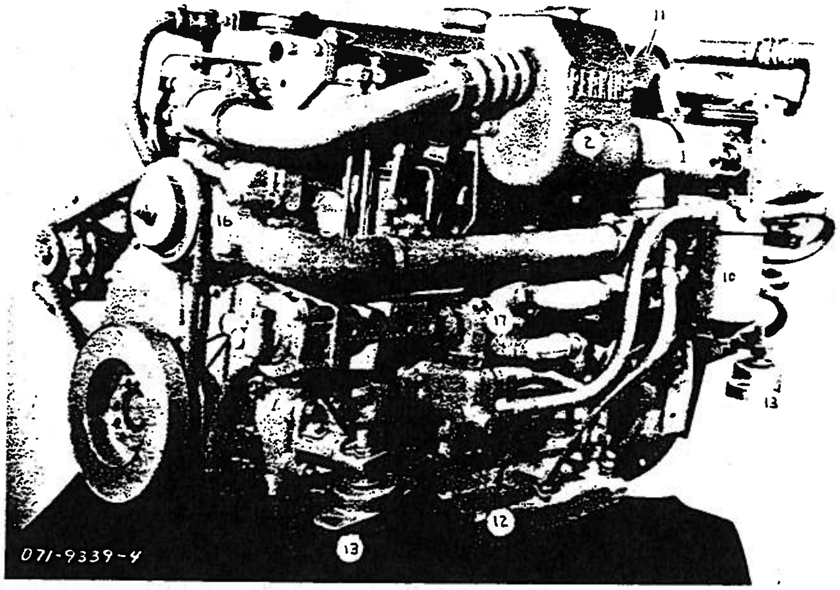
Diesel engines may be classified in a number of ways, according to certain important feature in their construction or operation.

An engine is either four-stroke or two-stroke; either turbocharged or naturally aspirated. Both these pairs of terms have been explained just above.

It is described by its number of cylinders, and their arrangement. Single-cylinder engines are rare, though they are sometimes used as small sail yacht auxiliary engines. The in-line arrangement for the more usual multi-cylinder engines is used for two to six cylinders (as many as twelve in ship-size engines). The V arrangement is used for from six and twenty cylinders. Other possible arrangements, such as radial and delta, are used, but are not common, and are not found among American-built small-craft engines.

In addition to its number of cylinders, the size of an engine is described by its bore and stroke, these being respectively the diameter of the cylinder and the length of piston travel. The volume swept out by the piston on one stroke, times the number of cylinders, is the engine displacement.

An engine is either a trunk-piston, or a crosshead engine, this distinction referring to how the pistons are



1	HEAT EXCHANGER	7	INTERCOOLER	12	OIL PAN
2	COOLANT HEAD TANK	8	REVERSE GEAR	13	MOUNTING
3	EXHAUST MANIFOLD	9	COOLANT OUTLET, EXHAUST MANIFOLD	14	TURBO CHARGER
4	EXHAUST	10	LUBE OIL FILTER	15	INLET MANIFOLD
5	FUEL LIFT PUMP	11	AIR INLET	16	WATER PUMP
6	ALTERNATOR			17	RAW WATER PUMP

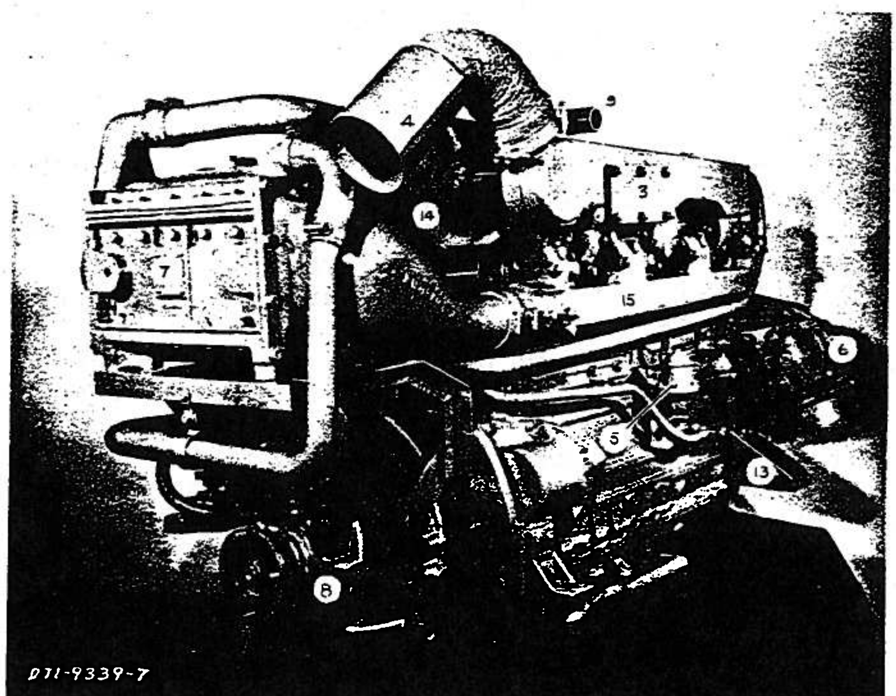


FIGURE 3.1 Perkins T6.354 MGT - 185 bhp, 2400 rpm

connected to the crankshaft. The trunk arrangement is the simple one found in the familiar automotive engine. In the other, the piston is connected to a crosshead that slides in guides below the cylinder, and the crosshead is in turn connected via the connecting rod to the crankshaft. Crossheads, however, are found only in the large ship-size engines.

Some engines are of the opposed piston arrangement; two pistons, traveling in opposite directions, operate in the same cylinder.

Engines are often classified as either low-speed, medium-speed, or high-speed. The low-speed engines are generally those that are designed for direct connection to a ship propeller; a rated RPM of around 120 is typical. Medium-speed engines must drive through a reduction gear in most installations; rated RPM typically lies in the range 400 to 800. High-speed engines likewise use the reduction gear, and usually turn at well over 1000 RPM; a rated RPM anywhere in the range 1500-2500 would be typical. As mentioned in Chapter 1, small-craft engines usually fall in the high-speed category, with medium-speed engines being used in some of the larger vessels of the "small" class.

Figure 3.1 depicts a typical marine diesel engine of the high-speed truck type.

3.3 SIGNIFICANT DIESEL ENGINE PARAMETERS

Several significant parameters can be introduced in the context of a brief discussion of what happens during the cycle taking place in a diesel cylinder. Begin with the compression stroke: air trapped in the cylinder is compressed by the upward stroke of the piston. While the piston is in the vicinity of top center, the air charge is heated by combustion of fuel that is injected at this point. With its pressure and temperature both increased by this addition of energy, the air then expands to drive the piston downward. The expanded, but still hot, air and burnt-fuel mixture is then expelled and replaced by a fresh charge so that the cycle can be repeated. (This last sentence conceals the major differences between four-stroke and two-stroke, and between supercharged and naturally-aspirated, engines, but the differences are unimportant in the present discussion.)

The obvious function of the cylinder is to produce mechanical work from this cycle of processes. The amount of work produced per cycle can be found from a record of instantaneous pressure in the cylinder, since

$$P \times A = \text{force}$$

$$P \times A \times L = \text{work}$$

where P is pressure, A is piston face area, and L is the stroke length. A and L are, of course, constant for a particular engine, but P varies drastically between firing pressure and exhaust pressure.

Figure 3.2 shows a representative pressure-volume diagram for one cycle of an engine cylinder. Such a diagram is often called an indicator diagram, since it can be traced from an operating engine by an instantaneous recorder known as an indicator.

The work done by a process shown on a pressure-volume plane is given by

$$ {}_1W_2 = \int_1^2 P \, dV \qquad (3.1) $$

which is the area, in pressure-volume units, of the area under curve that describes the process between points 1 and 2. If there are several processes that form a closed cycle, it is easy to extend this principle to the rule that the cycle net work is equal to the area enclosed within the cycle on the P-V plane.

Now let the cycle be replaced by one having equal area between the same volume limits, but with a constant pressure difference, i.e., a rectangular cycle such as shown by the dashed lines in Figure 3.1. If this imaginary cycle actually occurred, it would produce the same work at the crankshaft as the real cycle. Its advantage for discussion is that it is conveniently characterized by the constant value of a single parameter, the height of the cycle or its pressure difference. This idea is borrowed to characterize the actual cycle, since its value can easily be found (if the indicator diagram is available) by measuring the area of the cycle, and then dividing by the volume difference. The parameter is known as the indicated mean effective pressure (IMEP, imep, P_{mi} for symbols). Note again that it is directly proportional to the net work done per cycle.

The formula for indicated horsepower (IHP) can now be constructed. The product P L A has earlier been noted as a work quantity. If P is the indicated mean effective pressure, P L A is the indicated work per cycle. Multiplying by N cycles per unit time gives work rate, or power. Thus

$$ IHP = k \, P \, L \, A \, N \qquad (3.2) $$

where k is the conversion factor from work rate to horsepower. Typically N is given in cycles per minute and the product P L A is in ft-lbf, requiring k to be 1/33000.

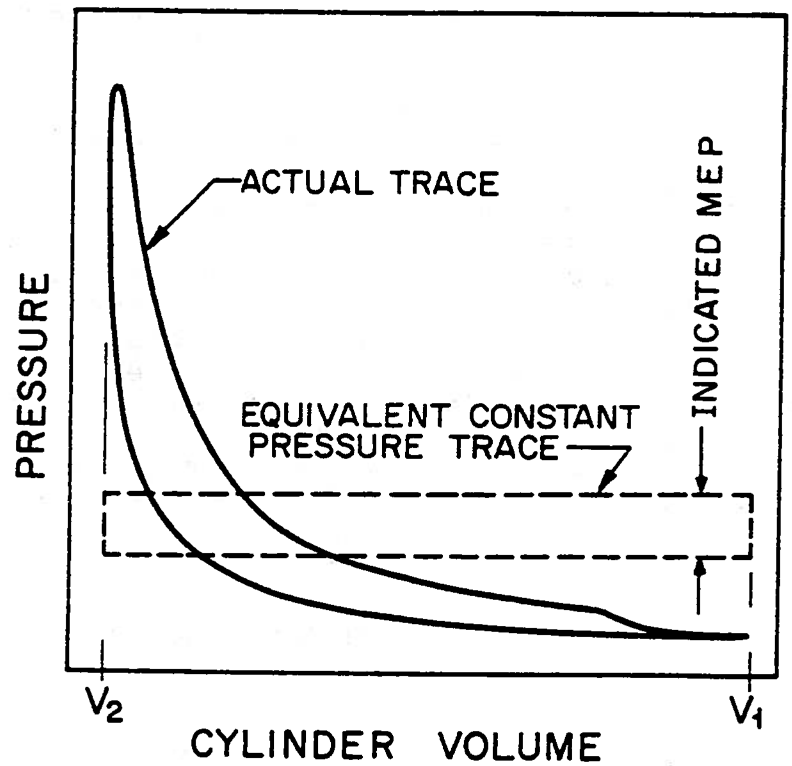


FIGURE 3.2 Pressure-Volume Diagram for a Diesel Cylinder

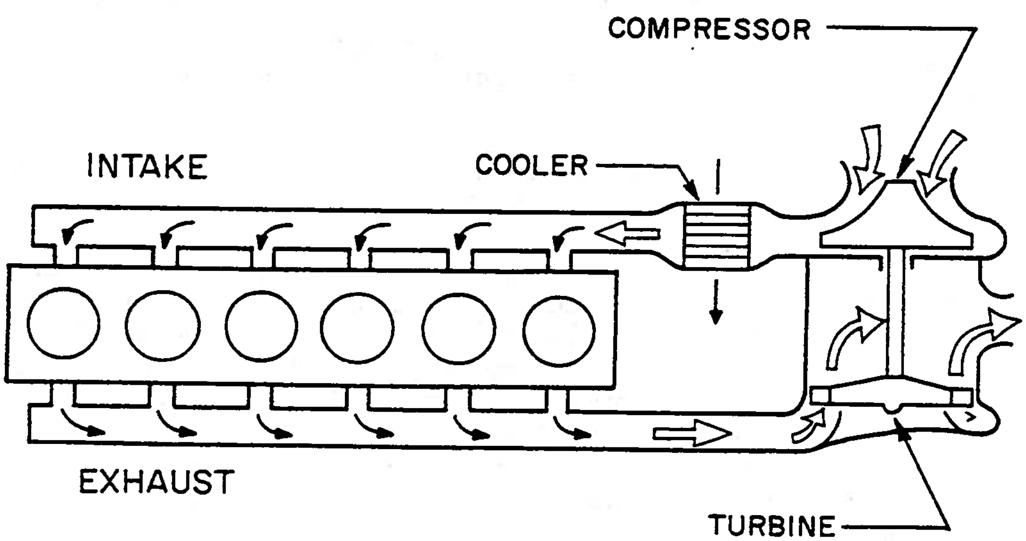


FIGURE 3.3 Turbocharging

For the two-stroke engine, N is equal to RPM, since each cylinder completes a cycle in one turn of the crankshaft. For the four-stroke engine, N is 1/2 of RPM.

Note that for multicylinder engines, A must be the total piston area for all cylinders if total engine power is wanted.

IHP is the power produced in the engine cylinders, and is greater than the power that can be applied to an external load since friction and parasitic loads (water pumps, fuel pumps, etc.) subtract from cylinder power. The power that is available for the external load is the brake horsepower, so-called because it is traditionally measured with the aid of a brake on the flywheel (an electromagnetic dynamometer is more commonly used today). The brake or dynamometer measures the torque exerted by the engine. The product of torque and rotational speed is proportional to power, hence

$$BHP = k'Q\ N \tag{3.3}$$

For torque in lbf-ft and N in RPM, the conversion factor k' is $2\pi/33000$.

The ratio of BHP to IHP is the mechanical efficiency (η_m) of the engine.

$$\eta_m = \frac{BHP}{IHP} \tag{3.4}$$

Combining the last with equation (3.2) gives

$$BHP = k(\eta_m P)\ L\ A\ N \tag{3.5}$$

The quantity in parentheses, $\eta_m P$, is the brake mean effective pressure (BMEP), i.e., a pressure proportional to work per cycle as measured on the output shaft. Note that also

$$\eta_m = \frac{BMEP}{IMEP} \tag{3.6}$$

Two other significant parameters that can be extracted from the P L A N formula are the displacement and the piston speed. Displacement is the volume swept out by the pistons in one stroke, and so is simply the product A L. The linear speed of a piston obviously varies during its stroke since it must come to rest at top and bottom center while reaching a maximum speed in between. The speed conventionally referred to is the average speed during the stroke, and this is the product 2 L(RPM).

Compression ratio is the ratio of the volume swept by the piston to the clearance volume above the piston at top center, or V_1/V_2 in Figure 3.1.

Several efficiencies are appropriate to this discussion. One, mechanical efficiency, has already been defined. Another is the brake thermal efficiency (η_{bt}), which is the ratio of engine brake output to its fuel energy input. This efficiency is the one most likely to be of interest to the engine user. If both input and output are expressed in Btu,

$$\eta_{bt} = \frac{2545(BHP)}{m_f\ h_f} \qquad (3.7)$$

m_f = fuel consumption, lb/hr h_f = heating value, Btu/lb

Volumetric efficiency is the ratio of air mass flow rate to the engine to the air mass flow rate found by the product of inlet density and cylinder swept volume per unit time, or stated otherwise, it is the ratio of air mass received by the cylinders to that ideally received. It is significant since the mass of air present in a cylinder determines the amount of fuel that can be burned there, and so determines the possible output of the cylinder. Strictly speaking, the definition does not fit the two-stroke engine, since in the scavenging process air is blown through the cylinder, meaning that more air is furnished than actually remains in the cylinder once the exhaust valve or ports close. The concept of a volumetric efficiency is the same, however, and the definition can be elaborated to suit.

Volumetric efficiency is not of direct interest in the application of an engine, and values are never quoted in application literature. It is mentioned here because it forms a background for the later discussion of supercharging.

Specific weight, the engine weight per unit output (BHP), might be called the weight efficiency of the engine. Although seldom quoted by engine builders (they usually give the weight itself) it is a parameter frequently used by marine designers in judging the merits of competing engines. The same parameter is conveniently applied to the entire machinery plant. This suggests a note of caution: when encountering a value of specific weight of a marine propulsion plant, check to see if the entire outfit is meant, of just the propulsion engine itself.

3.4 SPECIFIC OUTPUT

Specific output is the power output per unit piston area, or in symbols, BHP/A. Its significance can be seen from the P L A N formula (equation 3.5):

$$BHP/A \propto (BMEP)\ N\ L \tag{3.8}$$

Recall that average piston speed is proportional to N L, so that specific output is proportional to the product of BMEP and piston speed. These two factors are generally the limiting ones in engine design, thereby making BHP/A a quite meaningful measure relating engine unit output to the state of engine design technology.

Piston speed is related to accelerative loads on bearing sufaces, and their limited ability to absorb loads limits the piston speeds that can be used. An excessive (i.e. excessive as compared with recognized values for similar engines) BHP/A may indicate an engine being operated beyond a reasonable speed capability. Of course, it may otherwise indicate an excessive BMEP. Or yet again, it may indicate an engine of superior design.

In addition to mechanical stresses caused by accelerative loads, there are those related to BMEP, for the maximum pressures that occur in the cylinders are a function of this parameter. Thermally caused stresses, those caused by temperature gradients in the cylinder walls, cylinder heads, and pistons, are also related to BMEP--the higher the output from a cylinder, the higher will be its internal pressures, and the steeper will be the thermal gradients. The stationary parts of the cylinder assembly are always cooled by water passages, but the piston presents a difficult cooling problem because of its motion. The piston cooling problem becomes more severe as the size of engines increases, because as the pistons get larger, the heat flow path from center of piston to the cylinder wall cooling passages via the piston rings becomes longer. The really large low-speed engines all have water-cooled pistons, in spite of the difficulty this entails in getting water to and from the moving piston. Medium-speed engines (i.e. medium size) often have oil-cooled pistons. Their cooling problem is less severe because of the smaller piston size, so that the poorer cooling qualities of oil can be tolerated (oil is otherwise preferred to water because its leaks won't contaminate the crankcase). With the still smaller pistons of the high-speed engines typically found in small craft, conductive cooling via the piston rings to the cylinder walls is usually sufficient. In fact, although thermal loading if often the principal limitation on BMEP in large engines, in small engines the limitation is more usually air supply to the cylinders. As the inlet and exhaust passages become smaller, it becomes more difficult to cram in the air required to sustain efficient combustion of all the fuel that might be injected. But whichever the fundamental cause, the result is a limitation of BMEP.

Thus it is that limitations on engine output can be categorized as either a BMEP limitation, or piston speed limitation, making BHP/A a handy "goodness measure" because it is proportional to the product of these factors, and because it is easy to calculate from information that is usually available on any engine. It is not, however, a tell-all. The closing sentence two paragraphs back hints at the missing feature: if a BHP/A is excessive, how do you know whether the engine is being overloaded, or whether it is of superior design? Just from BHP/A you can't; it is, after all, a very simple parameter. You need to know more. Nonetheless, specific output is a useful concept.

3.5 TYPICAL DATA

Several significant engine parameters having been introduced and discussed qualitatively, it is perhaps time to supply a few numerical values. Table 3.1 lists data for several popular small craft engines, all as taken from builders' literature.

3.6 TURBOCHARGING

Marine diesel engines can gain significant benefits from supercharging; consequently most engine builders offer engines with this feature. The supercharging equipment is always an integral component of the engine, so that the vessel designer need not concern himself with this item any more than he does with any other individual component. He does, however, often have the alternative of buying a naturally-aspirated (non-supercharged) or supercharged engine, and the characteristics of the supercharger can affect engine performance. These factors justify the brief discussion given here.

An engine is supercharged when the air is furnished to the cylinders at a pressure higher than the ambient pressure. This implies the use of a compressor, which can be driven mechanically from the engine itself, by a gas turbine running on engine exhaust gas, or by some means independent of the engine. The exhaust gas turbine is by far the most common method; the turbine/compressor combination is known as a turbocharger, and the engine is said to be turbocharged. In fact, this scheme is so popular that "supercharging" and "turbocharging" are almost synonymous terms. Figure 3.3 sketches the typical arrangement.

The definition is a little ambiguous when applied to a two-stroke engine, since it inherently requires a blower to furnish scavenging air. When the blower pressure is just high enough to accomplish cylinder scavenging (that's why it's a "blower" rather than a "compressor"), the engine is considered to be naturally aspirated. A practical dis-

TABLE 3.1 Principal Data on Several Small-Craft Diesel Engines

Builder	Model	No. Cyl	BHP	RPM	BMEP	Bore x Stroke	Displ. in^3	Weight lb
Perkins	6.354	6	130	2800	100	$3\frac{7}{8} \times 5$	354	1300
Cummins	VT8-370	8	370	3000	120	$5\frac{1}{2} \times 4\frac{1}{8}$	785	2775
Caterpillar	D330T	4	165	2200	135	$4\frac{3}{4} \times 6$	425	2150
Detroit Diesel	6V-71 (7062N)	6	265	2300	102	$4\frac{1}{4} \times 5$	426	2570

Data is from the 1971 Diesel and Gas Turbine Catalog, except for BMEP, which is calculated from power, RPM, and displacement.

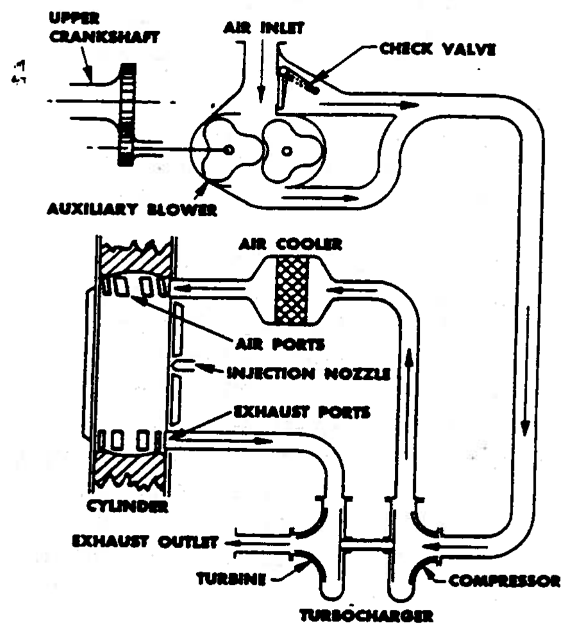


FIGURE 3.4 Scavenging and Turbocharging of a Two-Stroke Diesel

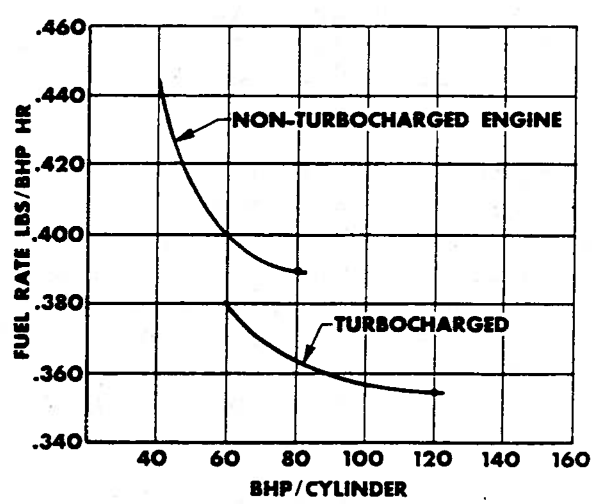


FIGURE 3.5 Example of Improvement in Fuel Rate after Addition of Turbocharger (from [5])

tinction can be made according to the driving arrangement. The blower of a naturally-aspirated engine is always mechanically driven from the engine, while the supercharging compressor is exhaust turbine driven. The turbocharged engine may retain the scavenging blower in series or parallel with the turbocharger, or the latter machine may perform the scavenging and charging function alone. Figure 4 shows an opposed-piston two-stroke engine with a mechanically-driven blower in series with the turbocharger.

The principal justification of supercharging is the increase of output from a given engine displacement, and hence from a given weight or linear dimension. Put simply, the more air that can be crammed into a cylinder, the more fuel that can be burned therein. The increased energy release points toward higher pressures and temperatures, and so tends to cause higher mechanical and thermal stresses. The increase in output thus must be kept within limits, but a turbocharged engine may produce as much as twice the output per unit volume as a naturally aspirated counterpart. The supercharging components add something to the weight and dimensions of the engine, but this deficit is minor.

Turbocharging is accomplished largely by energy that would otherwise be wasted, so that there is no significant increase in fuel rate to be expected. Indeed, a secondary benefit is an improved efficiency, so that where comparisions can be made, a supercharged engine typically shows a distinctly superior fuel rate. The causes of improved efficiency are (a) higher η_m because the power absorbed by friction is proportionally lower, (b) higher excess air, which improves cycle efficiency, and (c) smaller fraction of energy input absorbed by jacket water. Figure 3.5 illustrates the fuel rate improvement that was obtained by turbocharging a well-established naturally-aspirated two-stroke engine.

Turbocharged engines are allegedly slower to accelerate than naturally aspirated engines because of the necessity to accelerate the turbocharger before additional air is available at the cylinders. Documentary evidence for this is slender, however, and would seem to depend on the features of a particular engine more than on the general type. Use of turbocharged engines in rail traction service and in trucks indicates that this problem is a minor consideration to the application engineer.

Turbocharged engines are less sensitive to changes in atmospheric density than the naturally aspirated, but more sensitive to high exhaust pressure.

The air compressed by the supercharger is usually passed through a cooler before reaching the cylinders, the benefits being a greater air density and generally lower temperatures throughout the cycle. The process is referred to as intercooling (inter- = between compressor and cylinder) or aftercooling (after- = after compressor). The cooling medium is either engine jacket water, or water from a separate source such as the vessel's raw water supply. The point of interest is that the engine rating is affected by the temperature to which the air is cooled, and so may be affected by the temperature of the coolant arranged by the vessel designer.

Since the performance of an engine depends on its ability to convert fuel energy into mechanical energy, and this in turn depends on the oxygen that is made available for combustion, the characteristics of the turbocharger distinctly affect those of the engine. And since the engine supplies the energy to run the turbocharger, there is strong influence in the other direction also. Hence an important task for the engine designer is the optimal match between engine and turbocharger. His choices may be seen by the user, for example, in the shape of the torque-RPM curve, it being likely to have its peak near the engine RPM that matches turbocharger highest efficiency conditions.

In adopting the turbocharger, the engine designer is usually attempting to push the engine rating as high as possible, but he must also ensure adequate air supply at off-design points. This leads to some of his biggest problems, particularly with the two-stroke engine at light loads. Since this engine type is completely dependent on an external air supplier of some kind, and since at light loads its exhaust may not contain enough energy to sustain the turbocharger, complications to avoid the problem are inevitable. Figure 3.4 shows an auxiliary pump in series with the turbocharger for just this reason. The blower found on the naturally-aspirated version of the engine is retained to assist at light loads. Note the check valve that bypasses the auxiliary blower. It opens at high turbocharger speeds, unloading the auxiliary blower, and thus relieving the engine of the power that would be absorbed by supercharging mechanically.

All of the above discussion is but a qualitative introduction to the problems of designing the engine-turbocharger combination. Many textbooks on internal-combustion engines provide further details. I found References 5, 6, 7 useful in this area, even though one of them, [7], is specifically aimed at the big-ship diesel.

3.7 ENGINE SMOKE

The diesel has a reputation for smoky exhaust. It is partly deserved; everyone who has observed the noxious trail of diesel trucks on the highway knows that diesel engines do produce smoke. On the other hand, smoking is not at all necessary, at least in steady running; many of the trucks with clear exhaust stacks are also diesels.

There are numerous causes of diesel smoke, and even several types of smoke. For example, a whitish smoke sometimes appears during cold starting, in contrast to the more familiar dark greasy stuff. The white smoke is said to be due to cooling of the fuel spray before it ignites, perhaps from impinging on an overcooled spot of the cylinder wall. Unless it is the result of a serious design defect, this kind of smoke clears up as the engine warms. But it may persist if the engine remains lightly loaded.

The gruesome brown-black smoke seen from a steadily-running loaded diesel is due to incomplete combustion, which in turn can be caused by several different factors. The engine may simply be overloaded; the "smoke limit" is often used to determine the maximum output that should be taken from an engine. Or perhaps the injectors are worn, so that the fuel spray does not form properly. Or injection timing may be off.

A momentary smoking is sometimes seen when load is increased rapidly because the air supply fails to keep up with fuel supply during the transient period. This may be especially true with a turbocharged engine, since the turbocharger must accelerate before the air flow rate can be made to increase.

Brief smoking during startup and rapid load changes may be difficult to avoid, but the smoking during steady running is definitely avoidable by choice of engine rating to prevent overloads, and by proper maintenance during the service life of the vessel.

Reference 8 can provide further details on the diesel smoke situation.

3.8 TWO-STROKE VS FOUR-STROKE ENGINES

Both two-stroke and four-stroke engines are available throughout the power ranges covered by high-speed and medium-speed engines. Hence in selecting for a particular application, a designer nearly always has a choice between the two types. Is there any reason for preferring one over the other? If one were distinctly superior, that one would long ago have pushed the other out of the market, so the differences are either minor or there is a balance of advantages and disadvantages.

I prefer the "external" approach to evaluating engines. In contemporary jargon it might also be called an "interface" approach, meaning that evaluation is made solely on the basis of factors that are significant to the application, e.g. weight, fuel rate, first cost, maintenance cost, and several others. The details of the engine itself are ignored. Under this scheme one need not consider whether an engine is two-stroke or four-stroke, naturally-aspirated or turbocharged, single-cylinder or multicylinder, etc. The underlying assumption is that these internal considerations, if significant, are accounted for by the influence they have on the external factors. A recommendation of this attitude toward engine selection is an answer, albeit a "don't ask that question" answer to the two-stroke vs. four-stroke question. But of course, you may still like to know what external differences are to be expected, and there are indeed some.

The two-stroke engine has a power stroke on every revolution, compared to a power stroke on every other revolution in the four-stroke design. In consequence, it might be expected to have twice the specific output, and perhaps one-half the specific weight, of the four-stroke. However, it's not quite this simple, for there are complicating influences in both factors of specific output, piston speed and mean effective pressure, that tend to reduce the potential advantage of the two-stroke. Piston speed can be somewhat higher in a four-stroke engine because certain essential lubrication tasks are easier; for example, the absence of inlet ports in the cylinder wall improves lubricating conditions between wall and piston; in addition to allowing more cylinder oil to be used since there is no danger of clogging ports. The use of an intake stroke and an exhaust stroke greatly increases the mass of air that can be made available per cycle in the four-stroke cylinder, thus increasing the amount of fuel that can be burned, and hence increasing the BMEP. The four-stroke mean effective pressure is typically about 50-60% higher than for a comparable two-stroke engine. Both factors that make up BHP/A thus have four-stroke elements that tend to offset the seeming times-two advantages of the two-stroke. Nonetheless, the two-stroke design retains enough of its potential advantage that is usually does show a lower specific weight than its rival.

One should realize in discussions such as this that generalities are often obscured in a particular instance because the designers of engines may choose to emphasize one feature at the expense of others. For example, if a certain designer chose to emphasize lightness at the expense of other features, he could doubtless design a four-stroke engine that was lighter than a comparable two-stroke engine whose designer had put his emphasis elsewhere. The four-stroke design should be expected to have contain sacrifices in other areas, but they might be so artfully spread around that a detailed analysis of the competing designs would be needed to pinpoint them.

Generalities are thus often difficult to construct from published data on engines. Fortunately for our discussions, attempts have been made to sort out the factors in four-stroke and two-stroke comparisons by designing and building pairs of engines that were as nearly alike as possible, with one of each type. In one such case reported recently [9], the engines were put through a rigorous test program to illuminate as many differences as could be made evident. The engines were of medium speed, with the two-stroke version being of the valve-in-head uniflow-scavenging type that is typical of our two-stroke small-craft engines. The conclusions that were reached in this study are these:

Fuel consumption at rated load is essentially the same for both. Part-load consumption is distinctly better for the four-stroke. For example, at 30% of rated BMEP in the case of the two test engines, it is 25 grams/hp-hr better.

Air consumption is higher for two-stroke engines. Acceleration performance is better for four-stroke engines, since this type is not so much dependent on the acceleration of a turbocharger to supply additional air (this conclusion assumes that both engines are turbocharged).

Because of the generally lower piston speeds of two-stroke engines for equal specific outputs, their RPM tends to be lower, making it possible to match propeller speed with a cheaper reduction gear.

3.9 FUEL

Marine diesels burn a variety of fuels. The great majority of them burn either "marine diesel oil" or "heavy oil". The former is more properly designated grade 2 diesel fuel, as defined by the appropriate ASTM standard [10], and is the fuel usually supplied for both shore and marine diesels of the truck type. The heavy stuff is burned by the large ship diesels, and is usually a mixture of grade 2 oil with residual oil (the gunky stuff left over from the petroleum refining process). It is of poorer quality, and hence cheaper, than the grade 2 oil, and can be used successfully in large engines because their slow RPM makes sufficient time available for combustion.

The most distinctive property of fuel oils is their viscosity. Viscosity has a precise scientific definition, which I here leave to any fluid mechanics textbook to supply, but its method of measurement also gives a good idea of its meaning and significance. The common commercial method is to measure the time for a standard amount of oil to flow through a standard hole; the more viscous the sample, the longer will be the time, (which explains why commercial units are usually expressed in seconds). The point of significance to the diesel is that high viscosity makes the essential process of atomization difficult; the upper limit for atomization is about 100 SSU (SSU = saybolt seconds universal). The grade 2 oil typically has a viscosity of about 50 SSU at a temperature of 100F. The heavy oil may have a viscosity at this temperature in the range 1000-2000 SSU, so that heating to reduce viscosity is required before it is pumped to the engine. On the other hand, heating is seldom required when grade 2 oil is to be used.

The ignition quality of diesel fuel is also significant. It is mostly a measure of the time lag between start of fuel injection and the start of combustion. Obviously it must be a short time, and the higher the speed of the engine, the shorter this time must be. Because of the difficulty of actually measuring this lag, ignition quality is based on a comparative scale (i.e. different fuels are compared for performance in a test engine) whose units are the cetane number. The higher this number, the better the ignition quality.

Contaminants in the fuel, or produced by it during combustion, must be considered. Sulfur is present to some degree in all fuel oils, and is undesireable because its combustion products lead to sulfuric acid. Carbon residue is measured by evaporating a standard sample of an oil, and is a measure of its potential for fouling the engine. Ash includes gritty stuff that is bad for the fine passages of injectors, as well as dissolved inorganic compounds that can act as catalysts promoting the conversion of sulfur to sulfuric acid.

Flash point measures the safety of a fuel. The low flash point of gasoline is the property that makes it an explosion hazard; the comparatively high flash point of diesel fuels make them relatively immune to explosions from stray bilge vapors.

Table 3.2 gives the ASTM specifications for grade 2 diesel oil.

At the time this is written, the Navy is in the process of abandoning the traditional concept of unique fuels for steam, diesel, and gas turbine machinery, and developing a single fuel specification for all types. The properties of this fuel are described in Chapter 5.

3.10 TORQUE-RPM AND POWER-RPM CHARACTERISTICS

It has been said in Chapter 2 that the characteristic common to both engine and propeller is the torque-RPM characteristic of each, since the constituent factors of the mechanical power transmitted from one to the other are torque and RPM. It has also been noted that a power-RPM characteristic conveys the same information, and is sometimes a useful alternative.

The word characteristic denotes the graphical (usually) relationship between the two factors named, e.g. the change of torque as RPM changes with all other factors internal to the engine held constant. The diesel engine torque-RPM characteristic, at least in its ideal form, is found to be the simplest possible such relationship; if a load change causes the RPM to vary, the torque tends to remain constant if no adjustment is fuel supply is made to compensate for the load change. The diesel is often called, therefore, a constant torque machine.

Circumstances will modify this behavior; for example, if the engine is governed to a constant speed, the torque will be forced to change to suit this requirement. Also, certain features of the engine or of its turbocharger will modify constant-torque behavior. Nonetheless, the assumption of constant torque is realistic enough to make it useful in many problems involving engine-propeller interactions.

The explanation of the constant-torque behavior begins with a comparison of equations (2.1) and (3.5), the first being an expression for power in terms of torque and RPM, and the second being the same in terms of RPM and several appropriate engine parameters. They are repeated here with constants omitted.

$$BHP = Q\ N \qquad (3.9)$$

$$BHP = P_b L\ A\ N \qquad (3.10)$$

Note that Q (torque) is equivalent to $P_b L$ A. Since L A is fixed for a particular engine, this can be shortened to

$$Q \propto P_b$$

i.e. torque is proportional to brake mean effective pressure.

If mechanical efficiency is a constant, as actually it is over a reasonable load range, then torque is also proportional to indicated mean effective pressure. IMEP is proportional to the energy released by combution in the cylinder, and the latter in turn is proportional to the amount of fuel injected for each power stroke. Fuel is injected by a positive-displacement pump, so that the amount injected per power stroke should be the same, irrespective of speed. The net result is that torque is independent of speed, and is determined only by the fuel setting.

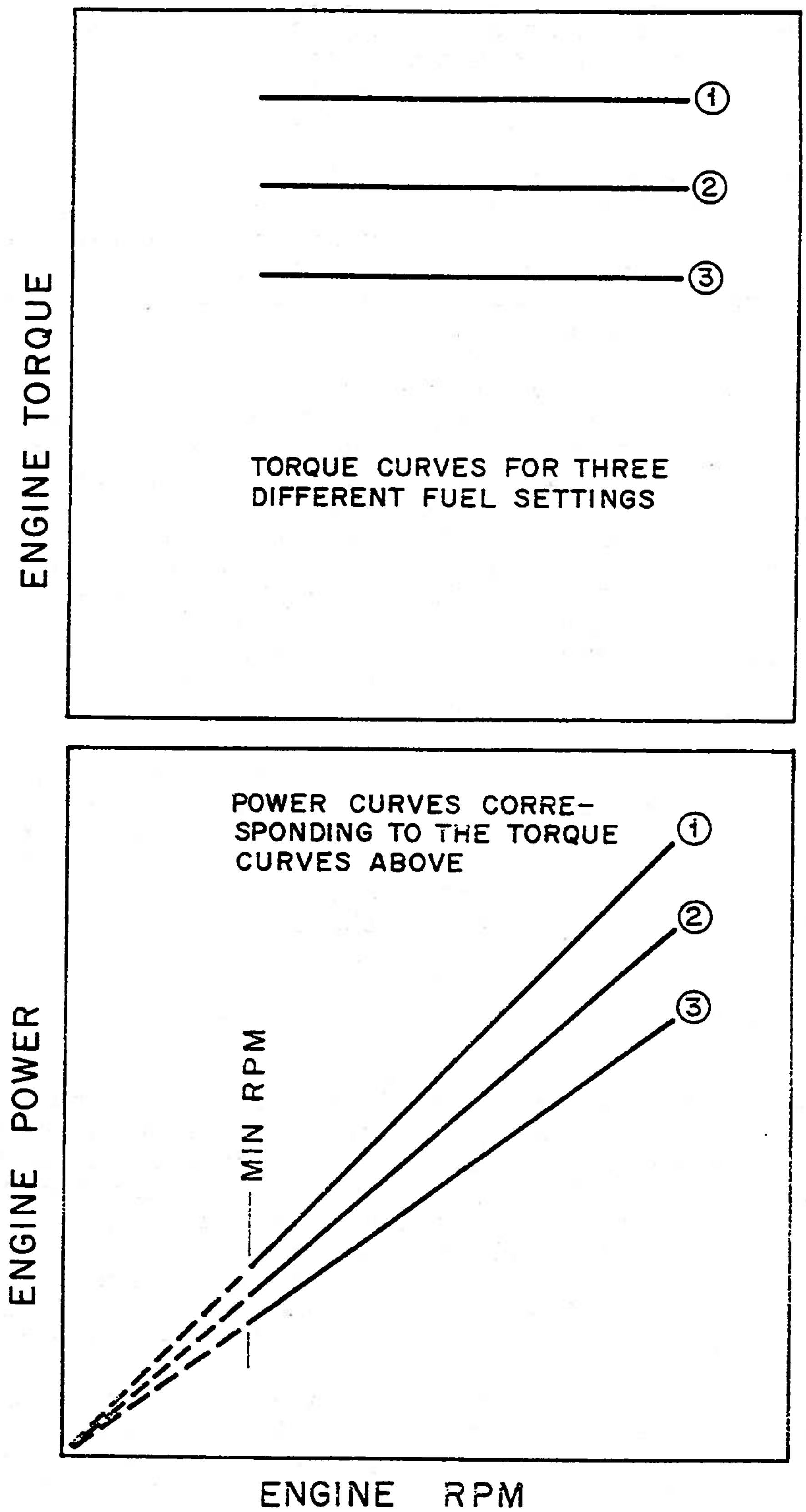


FIGURE 3.6 Ideal Torque-RPM and Power-RPM Curves

The ideal torque-RPM characteristic is thus a horizontal straight line, and the corresponding power-RPM characteristic is a straight line with slope proportional to the torque. The slopes of the latter curves are also proportional to BMEP, since BMEP and torque are proportional. Figure 3.6 illustrates.

The actual torque-RPM characteristic will always differ at least slightly from the flatness of the ideal curve. See Figure 3.7. The falling-off of mechanical efficiency at low speeds, and of the volumetric efficiency of cylinder and fuel pump at high speeds, all modify the case made above for complete speed independence.

In addition, the characteristics of auxiliary machines may influence the engine behavior. Examples are the turbo-charter, and attached auxiliaries such as coolant and lube oil circulating pumps. In the latter instance, the flow rate of these pumps is typically proportional to speed of engine, since they are driven from the engine. If full BMEP were maintained at a low engine RPM, the flows of water and lube oil might be inadequate for the high thermal and mechanical loadings. For this reason, the engine builder may specify that the BMEP (and hence torque) be cut back at low speed, and in some cases load-limiting governors may be installed to cut back the fuel at low RPM.

The turbocharger does not affect the amount of fuel injected, but its performance does determine the amount that can be burned per cycle, since it determines the amount of air that will be present in the cylinder. The match of turbocharger to engine thus influences the shape of the torque characteristic; the torque characteristic tends to peak near the condition of best turbocharger efficiency. This is illustrated in Figure 3.7b, wherein torque curves for identical engines fitted with two different turbochargers are shown.

The two-stroke turbocharged diesel typically suffers from a lack of exhaust energy to drive the turbocharger at light loads. These engines usually have some auxiliary means of carrying out the scavenging and charging function when exhaust energy is insufficient, such as mechanical blower in series or parallel with the turbocharger, or an auxiliary mechanical drive to the turbocharger itself. In any case, a distinct fall-off in torque capability at low engine RPM is typical of the turbocharged two-stroke engine.

To summarize: The torque-RPM characteristic of the diesel engine is ideally a horizontal straight line. In many engines, this is approximately borne out in practice. Lack of cooling and lubrication may make it inadvisable to use full torque at low speeds, or the inadequacy of turbo-charging at low speeds may prevent the engine from developing full torque at those speeds.

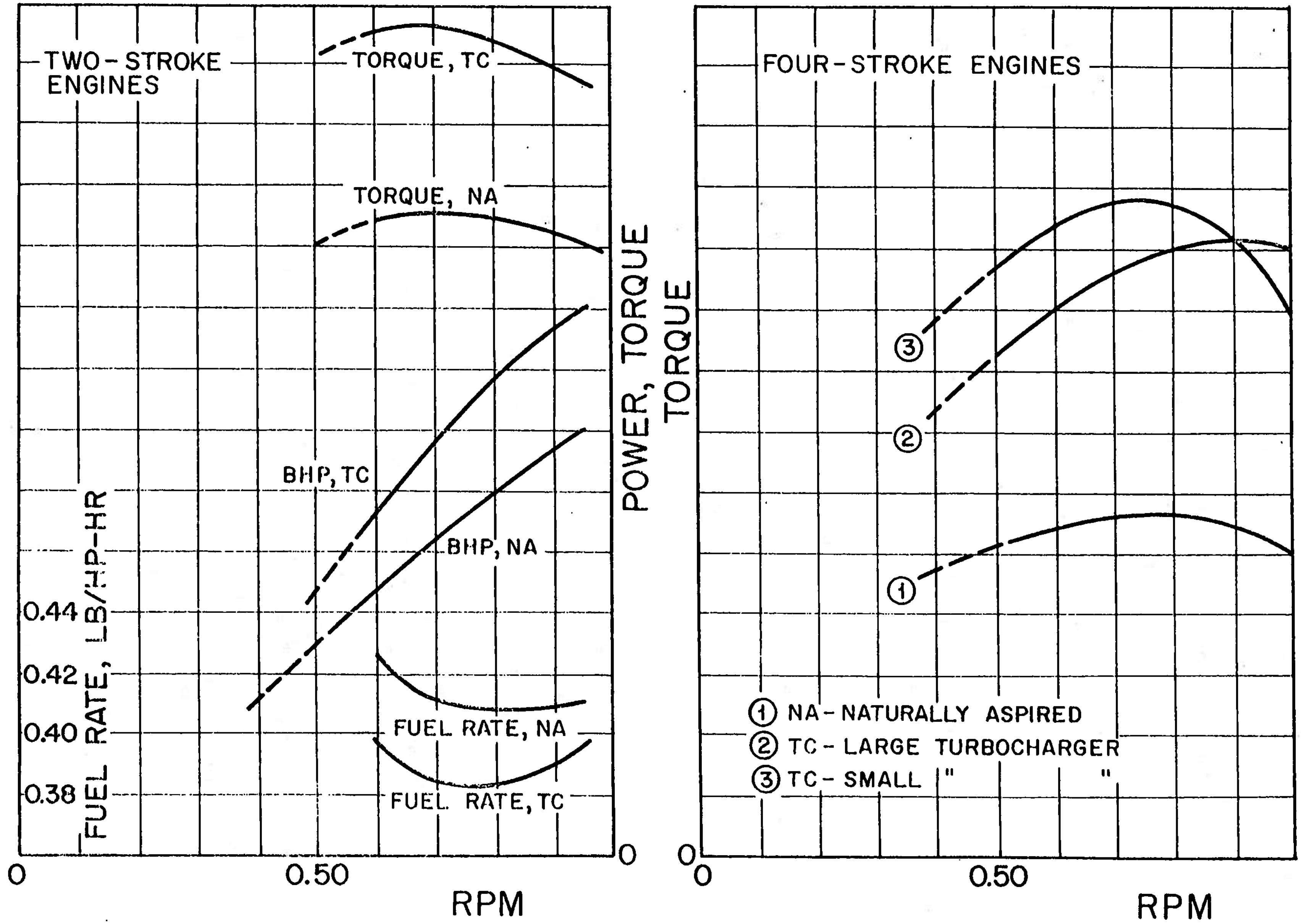


FIGURE 3.7 Examples of Actual Torque-RPM Curves

3.11 FUEL RATE CHARACTERISTICS

Fuel consumption rates vary significantly with varying operating conditions. The information on fuel consumption is presented in a number of different ways by engine builders. Typical ways are fuel rate vs. BMEP with RPM fixed, fuel rate vs. RPM with BMEP fixed, and fuel rate vs. assumed propeller power. See Figure 3.7 for one example.

The most informative way of presenting fuel rate information is by contours of equal fuel rates on a power-RPM or BMEP-RPM plane. Single-line characteristics, such as those listed in the preceding paragraph, are constructed simply by taking a slice in the desired direction across the contours.

Figure 3.8 gives an example of a fuel-rate contour map for a diesel engine.

3.12 REFERENCES

1. R. L. Harrington (editor), Marine Engineering, Society of Naval Architects and Marine Engineers, 1971

2. Rudder Magazine, September 1965

3. C. F. Taylor and E. S. Taylor, The Internal Combustion Engine, 2nd Edition, International Textbook Company, 1961

4. V. L. Maleev, Internal Combustion Engines, McGraw-Hill, 1945

5. J. P. Popson and D. W. Meether, "Turbocharging a Small Medium-Speed, Two-Cycle, Opposed-Piston Diesel Engine", ASME Paper 63-OGP-7, 1963

6. W. E. Wollenweber, "The Turbocharger--A Vital Part of the Engine Intake and Exhaust Systems", Society of Automotive Engineers Paper 700534, 1970

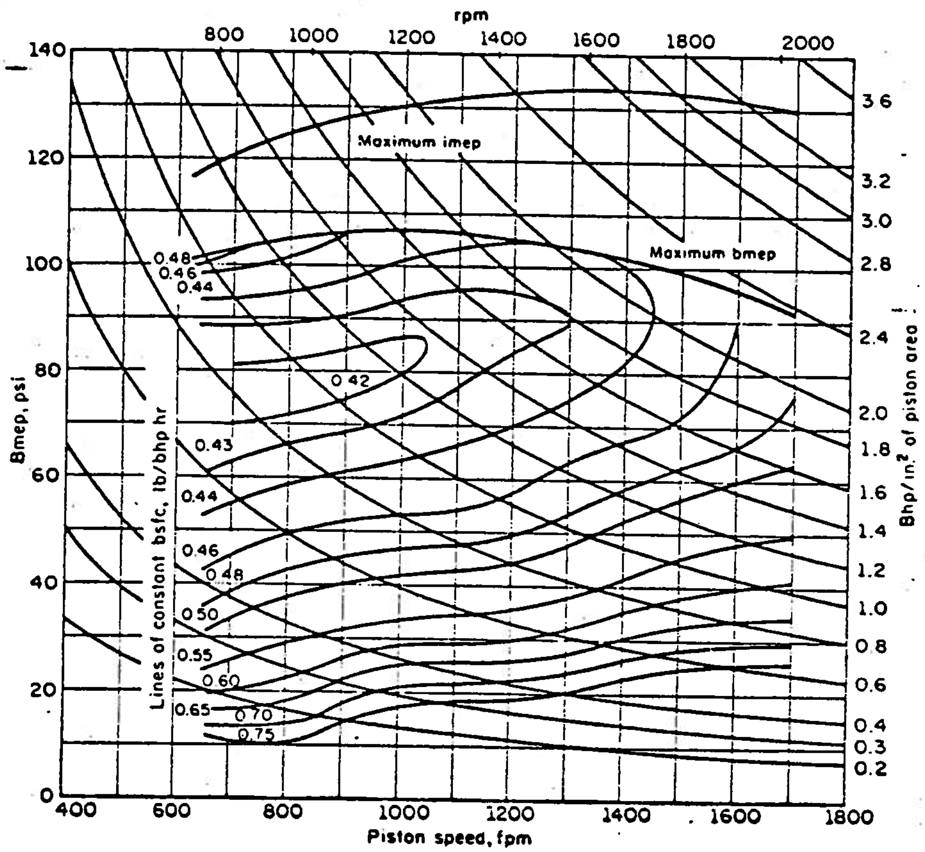
7. G. Gyssler, "Methods of Turbocharging with Special Reference to Large Two-Stroke Diesel Engines", Transactions of the ASME, Journal of Engineering for Power, April 1967

8. G. McConnell, "Influence of Fuel Properties and Effect of Antismoke Additives on Diesel Exhaust Smoke", Society of Automotive Engineers Journal, March 1968

9. H. A. Steiger, "Experimental Approach to Questions of 2-Stroke or 4-Stroke Cycle, Particularly for Diesel Engines of Extremely High Specific Output", ASME Paper 69-DGP-6, 1969

10. ASTM Specification D-975-50T

11. G. A. Carleton, "New Fuel for Naval Gas Turbine Engines," ASME Paper 70-GT-50, 1970.



(from Reference 3 with permission of publisher)

FIGURE 3.8 Typical Diesel Engine Fuel Rate Map

GRADE	CARBON RES % (MAX)	ASH % (MAX)	VISCOSITY AT 100F (MIN) CS	(MAX)	SULFUR % (MAX)	CETANE NO. (MIN)	FLASH POINT (MIN)
1	0.15	0.01	1.4	—	0.5	40	100
2	0.35	0.02	1.8	5.8	1.0	40	100
4	—	0.10	5.8	26.4	2.0	30	130

(from Reference 10)

TABLE 3.2 Properties of Diesel Fuel Oils

CHAPTER 4

THE SPARK-IGNITION MARINE ENGINE

4.1 INTRODUCTION

If outboard motors and the smaller sailing auxiliary engines are excepted, the marine spark-ignition (or "gasoline engine" if you prefer) is basically that most ubiquitous of earthly power plants, the automobile engine. I therefore believe it to be so familiar to most readers that descriptions beyond the strictly marine features would be redundant. Innumerable descriptive articles can be found in the popular boating magazines, engineering textbooks abound, and there are books devoted specifically to description, maintenance instructions, etc., relative to the marine engine; the book that comes first to mind to recommend is that of Conrad Miller [1]. All in all, I have ample excuse to omit the descriptive material.

The SI engine is a reciprocating internal-combustion engine, and therefore shares many features in common with the diesel engine, which is also IC and reciprocating. Much of what has been said relative to principles and terminalogy in Chapter 3 fits the SI engine equally well, for example, the P L A N formula; terms such as BMEP, brake thermal efficiency, etc., apply to both engines, and so require no repeating here.

One section of the diesel chapter that should not be applied to the SI engine is the one on supercharging, for supercharging is used only rarely with the marine SI engine. One reason is that the incentive to supercharge is less because the power per pound is already high compared to the diesel. Another reason is that supercharging decreases efficiency because compression ratio must be reduced to prevent pre-ignition (knocking), a handicap that the diesel avoids by compressing only air rather than an air-fuel mixture.

4.2 THE TORQUE-RPM CHARACTERISTIC

In discussing the ideal torque-RPM characteristic, one factor is significantly different from the diesel situation; since the fuel (in the form of a fuel-air mixture) is not metered directly into the cylinder in imcompressible form, no strong case can be made for a flat characteristic. In practice, however, the result is very much like the diesel. See Figure 4.1 for two examples. The deviation from flatness is caused largely by variation of volumetric efficiency with RPM, with highest torque occurring at the highest value of this efficiency. The location of highest η_v can be shifted by manipulation of valve timing and overlap. For automotive

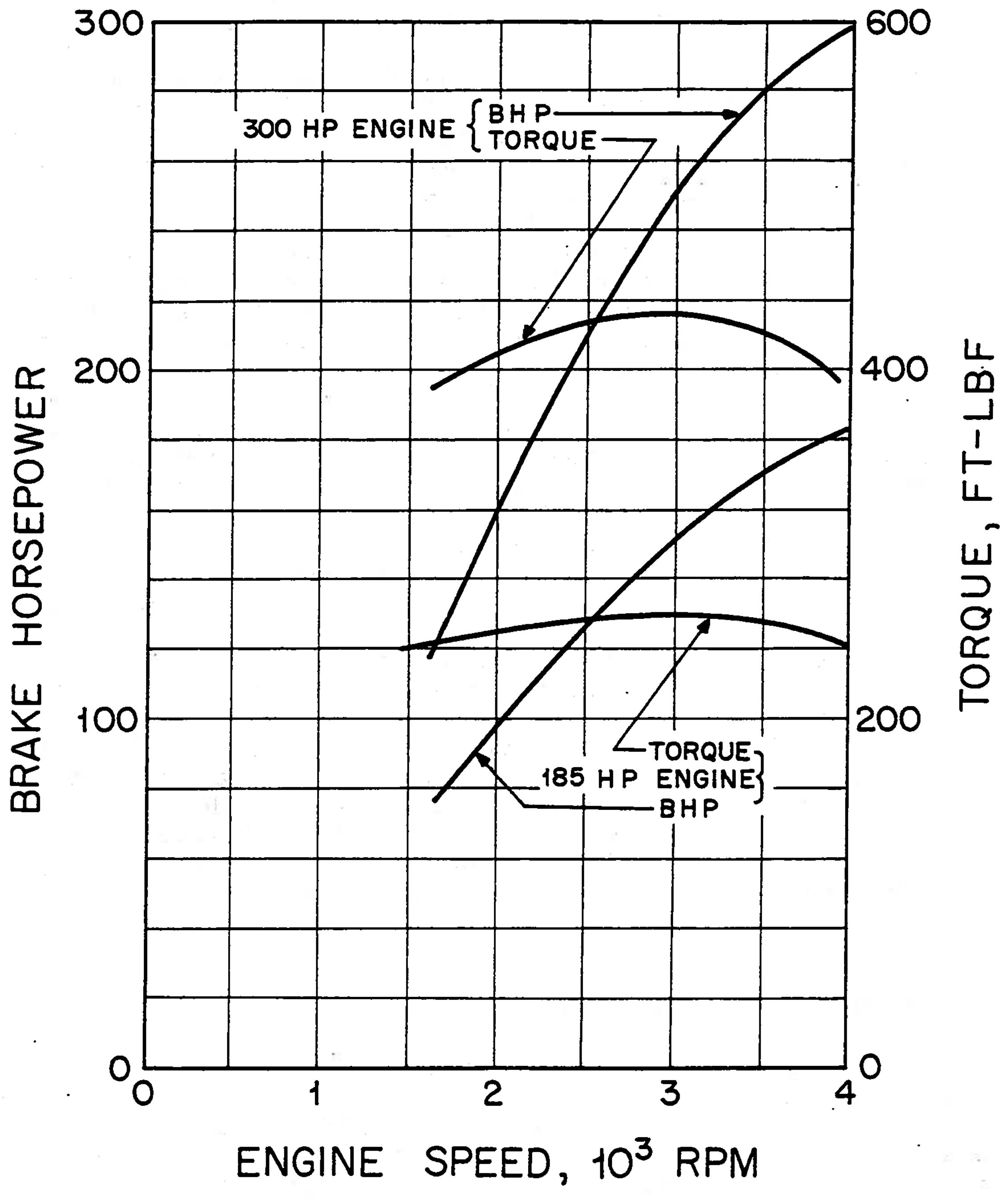


FIGURE 4.1 Torque and Power Characteristics for Two Spark-Ignition Engines

applications, high torque at low RPM is desireable, hence timing and overlap are often chosen for this feature. For most marine applications, peak torque closer to rated RPM might be better, but conversions to marine use are not likely to involve changing such fundamental matters as valve timing.

The actual torque curve of the SI engine tends to drop more sharply than that of the diesel above rated RPM. Because of this, it is sufficiently self-limiting in speed that a safety governor is not usually required.

4.3 FUEL RATE CHARACTERISTICS

The fuel rate characteristics are qualitatively similar to those of the diesel, although magnitudes of the fuel rates are noticeably higher. Figure 4.2 shows a typical fuel rate map for this type engine.

4.4 MARINE SERVICE CONSIDERATIONS

Several things must or should be done to the typical automotive SI engine when it is adapted to marine service. First, the "must" category, the things that appear to be essential:

1. Safety features. Possible fuel leakage points, principally vents and shaft entries in the carburetor must be sealed as part of the essential program to keep gasoline out of the bilges. A USCG-approved flame arrestor must be placed in the air intake. The exhaust manifold must be water-cooled to keep it from being a source of ignition for explosive vapors.

2. Cooling requirements. The automotive radiator cooling scheme is unsuitable, and either a water-cooled heat exchanger, or a direct use of sea water circulated through the engine jackets must be adopted. If the latter, a bigger water pump may be required because the water-cooled exhaust manifold is a marine addition, and because enough water must be circulated to keep the water temperature below the point (approximately 145 F) at which salts begin to solidity in the engine. If indirect cooling is to be used, a second pump to circulate sea water through a heat exchanger must be added.

3. Avoidance of corrosion. An engine using sea-water cooling is not going to last forever, but meanwhile functional parts exposed to the sea water must work. Drain plugs in the coolant system and thermostat elements, for example, must be made of corrosion-

resistant materials. The oil pan is likely to be wetted by bilge water, and should not be steel unless heavily coated.

4. Installation angle. If the engine is to be installed at a steep angle, a special oil pan may be required to allow proper oil-pump suction, and wedges may have to be fitted under the carburetor to level it.

5. Exhaust system. The wet exhaust system is most commonly used with the marine SI engine, and consequently care must be taken to avoid getting exhaust-cooling water into the engine via the exhaust manifold. Decelerations of the boat, and pulsating exhaust flow can both contribute toward the danger. For this reason risers are usually added to the exhaust manifolds to form a water trap.

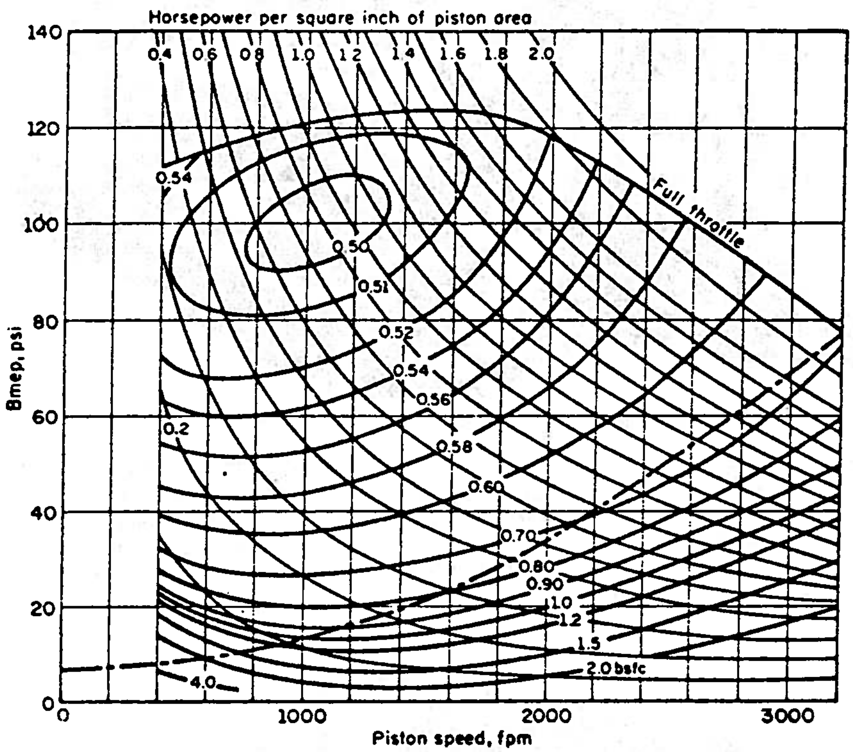
The list continues with other measures that may not be essential, but are often worthy of consideration:

6. The "load profile" of a marine engine is likely to be quite different from that of its automotive progenitor. Even when traveling at maximum highway speed, the typical automotive engine is still lightly loaded (your foot is still a long way off the floor), but a boat will require maximum power to reach maximum speed, and even at cruising speed the engine may be comparatively heavily loaded. It thus may be subjected to service much more severe than automotive service. Valve gear, especially exhaust valves, are sensitive to this condition. Leaded fuel is said to help here because deposits on the valve seat from the lead additive tend to prevent valve and seat from chewing each other up from momentary welding together. See Reference 2 for details of one builder's attention to the exhaust valve problem.

7. Operation on regular gasoline is desireable from the standpoint of availability and cost. For this reason, compression ratio is often reduced in the adaptation process by substitution of lower-compression pistons.

8. Reverse rotation, though not essential, is desired for one engine of the pair in a twin-screw boat. Camshafts, starters, generators, distributor drives, and camshaft oil slingers must be reversed to accomplish reverse rotation.

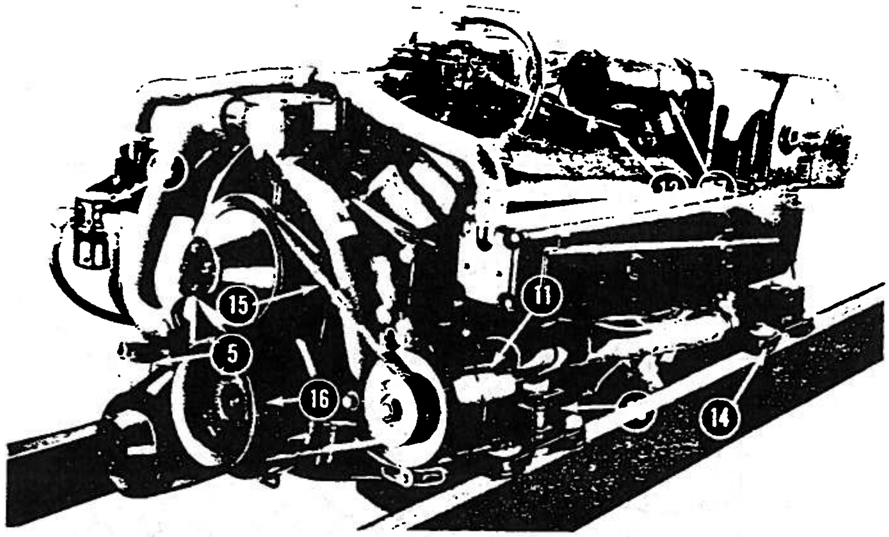
References 2, 3, although treating the products of one company, are informative further reading on the general problem of conversion from the automotive field.



Performance map of United States typical passenger-car engine as installed.
—·—·— Level-road requirement in high gear

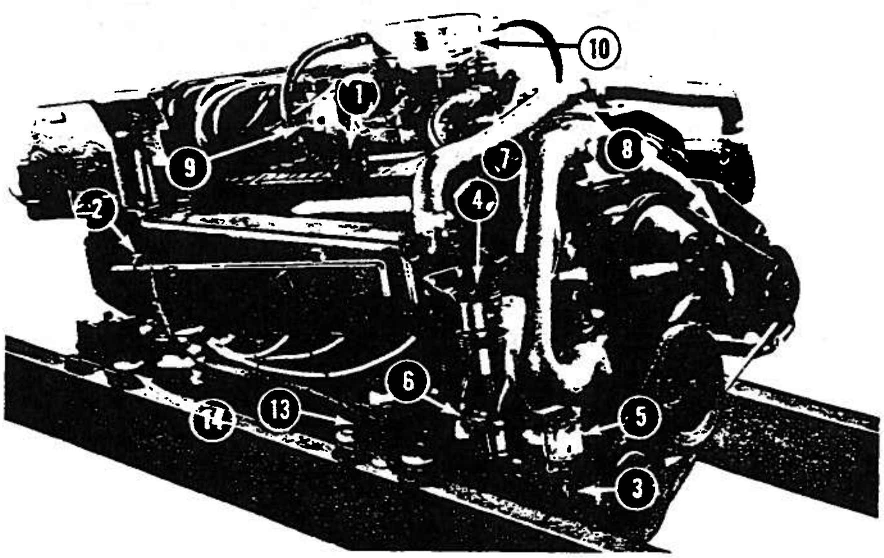
(from Reference 8 with permission of the publisher)

FIGURE 4.2 Fuel Rate Map for a Spark-Ignition Engine



MerCruiser "325" Inboard Marine Engine, Left Side

1 - Oil Filler Cap
2 - Dip Stick *(May Be Located on Rt. Side)*
3 - Water Pickup Pump
4 - Fuel Filter
5 - Sight Glass
6 - Fuel Pump
7 - Carburetor Fuel Filter
8 - Thermostat Container
9 - Distributor
10 - Flame Arrestor
11 - Alternator
12 - Throttle Cable
13 - Front Mounts
14 - Rear Mounts
15 - Alternator Drive Belt
16 - Water Pickup Pump Drive Belt
17 - Oil Filter



MerCruiser "325" Inboard Marine Engine, Right Side

FIGURE 4.3 Kiekhaeffer-Mercury MerCruiser 325 Engine

4.5 SAFETY

As I should hope everyone knows, gasoline forms explosive mixtures with air, and such mixtures do form from gasoline in the bilges of motorboats. Therefrom and consequently, motorboats do explode occasionally with gruesome results. Observance of safety rules in design, building, and operation is therefore essential, and highly worth mentioning here even though this is definitely not a rule book. This paragraph is intended mainly as a vehicle for calling attention to the places where the rules are found, specifically [4], [5], [6], [7]. Although there is considerable overlap among these sources, careful attention to all of them is well worth the effort.

4.6 REFERENCES

1. Conrad Miller, Small Boat Engines, Inboard and Outboard, Sheridan House, 1961

2. Neil Newman, "The Metamorphosis of a Powerplant--from Passenger Car to Pleasure Boat," Society of Automotive Engineers Paper 780B, 1964

3. John W. Hurst, "Genesis of an Inboard Engine," Yachting Magazine, August and September, 1968

4. BIA Engineering Manual, Boating Industry Association, annual

5. Rules and Regulations for Uninspected Vessels, Subchapter C, United States Coast Guard

6. NFPA 302, Motor Craft, 1966, National Fire Protection Association

7. Safety Standards for Small Craft, American Boat & Yacht Council, annual

8. C. F. Taylor and E. S. Taylor, The Internal Combustion Engine, Second Edition, International Textbook Company, 1961

CHAPTER 5

THE MARINE GAS TURBINE ENGINE

5.1 INTRODUCTION

The fundamental components of a gas turbine engine are a compressor that compresses the incoming air to a pressure of several atmospheres, a combustion chamber (burner, combustor) into which fuel is injected continuously and burned, and a turbine that extracts energy from the compressed and heated air to drive the external load as well as the compressor and engine accessories.

Several significant modifications are possible. The free turbine engine has its turbine divided into two mechanically separate units; one drives the compressor, while the second (the free turbine) drives the external load. When there is only one turbine to handle both functions, the engine is said to be single-shaft. Another modification is the addition of a heat exchanger to transfer heat from the hot exhaust to the incoming air before it enters the combustion chamber, this being the regenerative engine or cycle. The simple engine or cycle is one that does not have the regenerative refinement.

Figure 5.1 sketches the types of machine mentioned above. Figures 5.2 and 5.3 picture gas turbines currently being used in small craft.

Two significant design parameters are the pressure ratio and the combustion temperature. The former is the ratio of compressor outlet to inlet pressure, and the latter is the temperature of air as it leaves the combustion chamber and enters the turbine. Their values, along with individual efficiencies of the compressor, combustion chamber, and turbine, determine the efficiency of the engine. Figure 5.4 indicates how they influence efficiency for a simple-cycle engine. The incentive to make their values high is apparent.

Since the turbine blades are immersed continusouly in a steady flow of the heated air, they operate essentially at the temperature of this air. The ability of the blades to endure high temperatures thus sets the allowable combustion temperature limit, and is the principal limit on efficiency. (Internal cooling of the blades is common in the large ship-size gas turbines, enabling some of the advanced designs to come close to the diesel in efficiency.) By contrast, the reciprocating internal-combustion engines expose their working parts (e.g. pistons) only intermittently to the highest temperature. This temperature is therefore higher than in the gas turbine, and efficiency of the diesel,

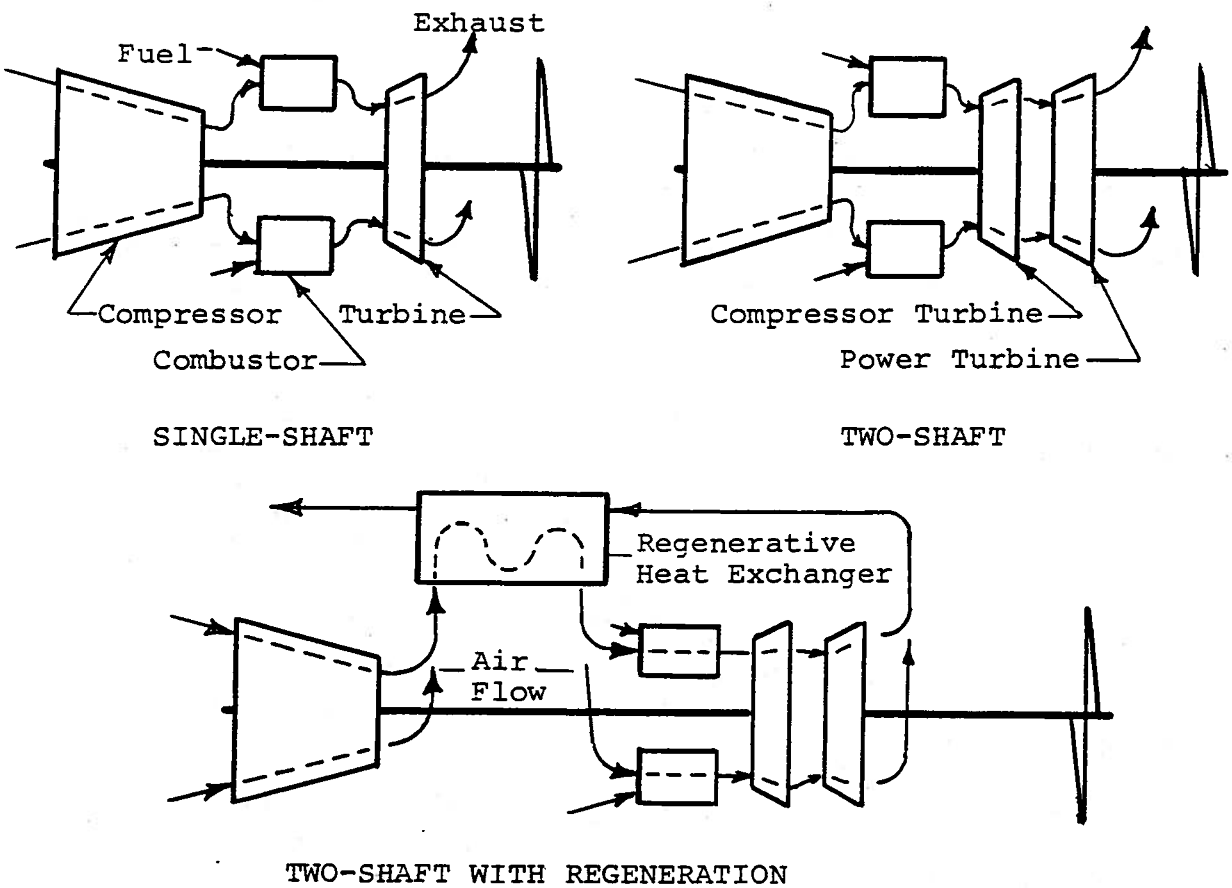


FIGURE 5.1 Principal Types of Gas Turbine Engines

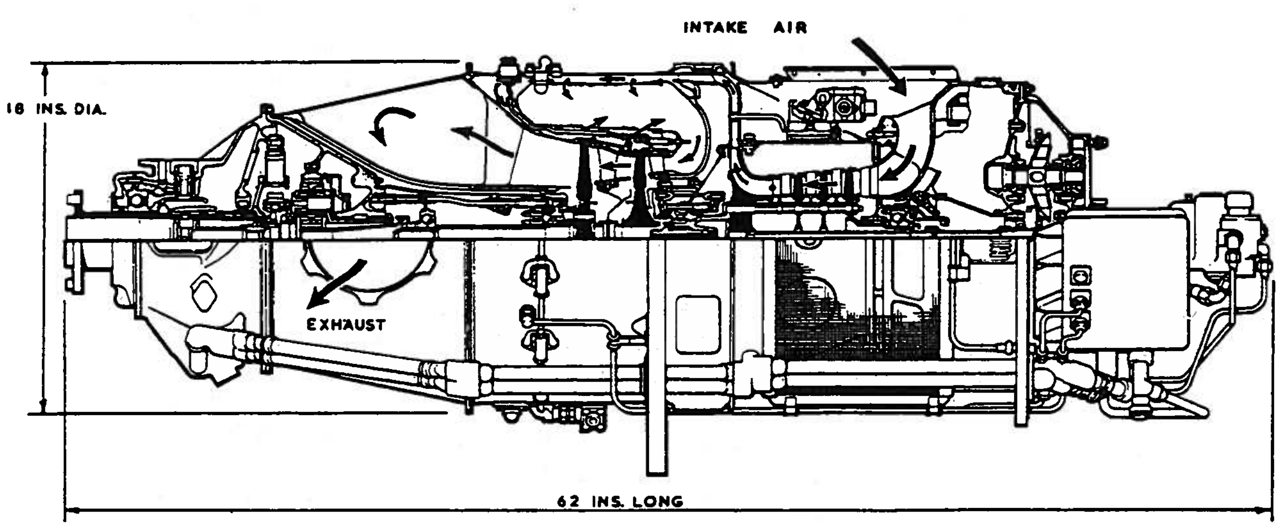
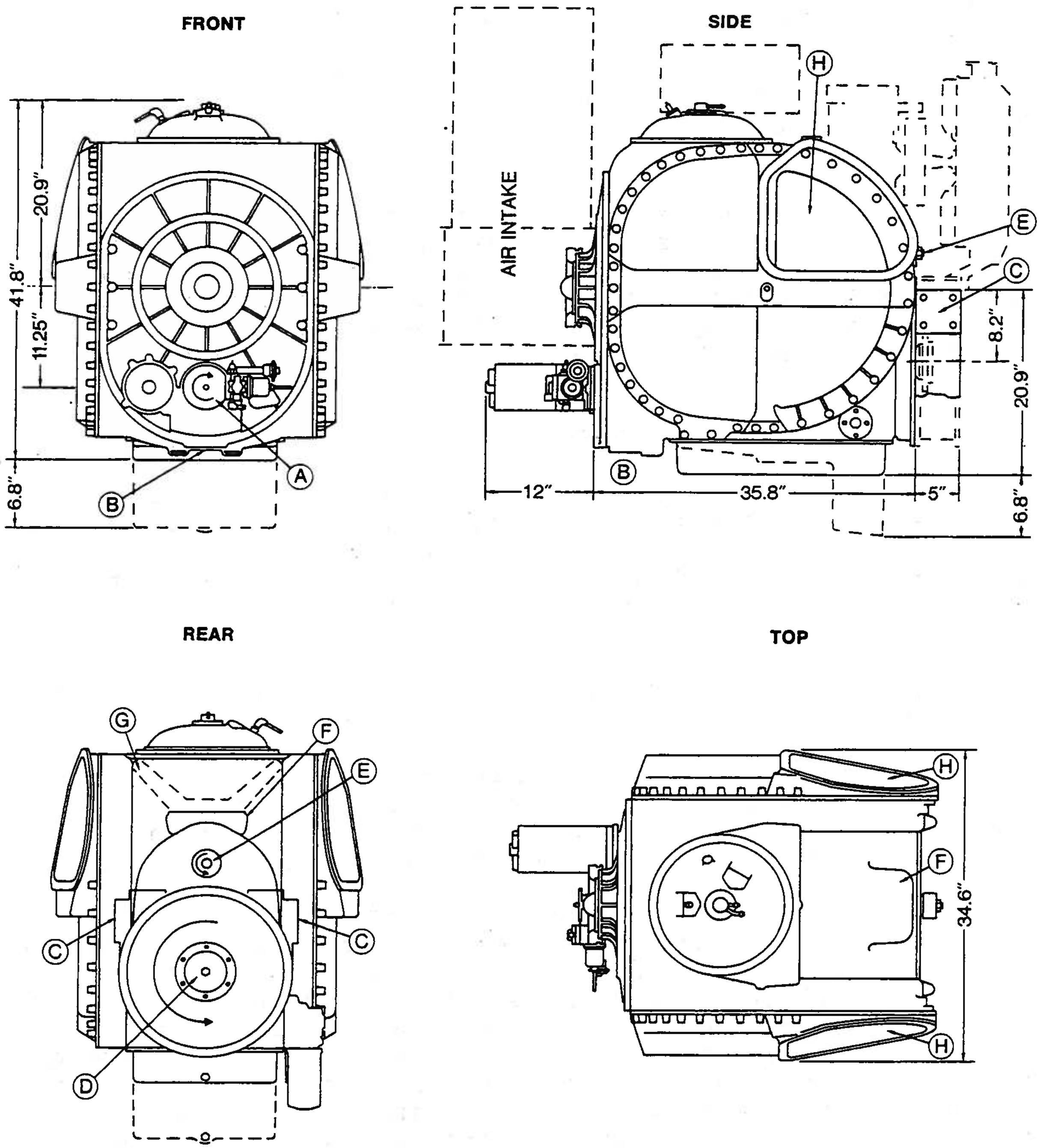


FIGURE 5.2 Small Craft Gas Turbine -- Approximately 500 hp -- United Aircraft of Canada ST6



A. Front accessory drive.
B. Front engine mounting.
C. Rear engine mounting.
D. Main power shaft.
E. Rear accessory drive.
F. Accessory mounting pad.
G. Accessory mounting bracket.
H. Upper or lower rear quadrant exhaust outlet available.

FIGURE 5.3 Small Craft Gas Turbine -- 320 to 525 hp -- Ford Motor Company at least, is correspondingly higher.

The relatively low combustion temperature of the gas turbine is obtained by a high air/fuel ratio, or put another way, by high specific air flow (lb/hp-hr). A significant consequence is the large cross-section of the passages for inlet and exhaust flow, this being one of the outstanding physical characteristics of a gas turbine installation.

The gas turbine shares the advantage of all turbo-machinery over reciprocating machinery: because the working fluid (air) is handled on a continuous-flow basis, rather than on a batch basis, the turbo-machine is much smaller for a given power rating. This is the basis of the gas turbine's principal appeal, its compactness and light weight.

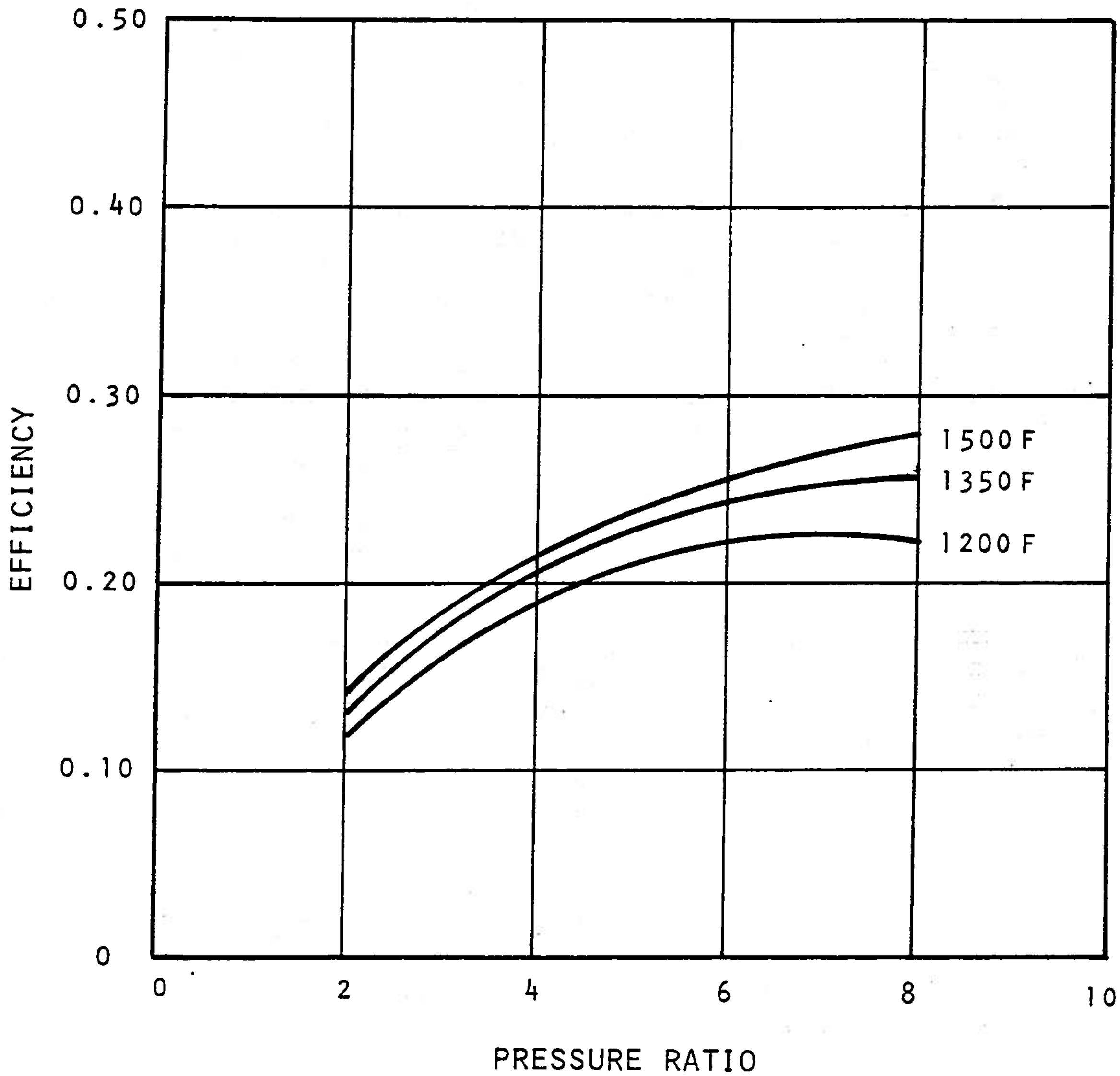
5.2 ADAPTING THE GAS TURBINE TO MARINE USE [1], [2]

Like the diesel and spark-ignition engines, the marine gas turbine is usually an adaptation of an engine originally designed for a non-marine use. Adaptations directly from aircraft (United Aircraft of Canada) and truck (Ford) engines have been prominent among the few small-craft engines now available.

The principal difficulties that the adaptation process must overcome are directly related to the marine environment, and especially to the sea salts that permeate the air that the engine must breathe, and that may also invade the fuel that it must burn. This problem can be divided into <u>cold corrosion</u>, i.e. that affecting the compressor, and <u>hot corrosion</u>, i.e. that affecting turbine nozzles and blades.

Sea salt entrained in the air corrodes the magnesium and aluminum (even if anodized) often used in aircraft compressors. Removal of salt from the air would seem to be the solution, but the practical methods of salt removal cause pressure loss in the inlet ducting, and such loss causes significant degradation of engine performance (see Section 5.5). Although ducting is designed as large as possible to encourage salt to drop out of the slow-moving air, intakes are sheltered from spray as much as possible, and separation (e.g. a filter) is used, some salt always gets through. Coatings, such as cadmium plating, have been used, although subject to failure from errosive wear. Titanium appears to be a completely resistant material for compressor construction, though expensive. Stainless steel can be used, but only if salt is washed from the unit after use.

Salt caking on the compressor blades also affects performance, no matter what the blade material may be. It can be serious in a short time; one published example [1] shows a



COMBUSTION TEMPERATURES ARE GIVEN ON EACH CURVE
COMPRESSOR EFFICIENCY = 0.84
TURBINE EFFICIENCY = 0.85

FIGURE 5.4 Efficiency at Rating of Simple-Cycle Gas Turbine Engines

35% loss in engine output after only 16 hours running time in a small boat. The solution is periodic washing with fresh water, usually accomplished by water from a spray ring installed in the compressor inlet. It is typically used after every trip after the engine is shut down, but while it is slowly turned by the engine starter.

The hot corrosive process is often called "sulfidation" because sulfide corrosive products result. The source of the sulfur is sodium sulfate in sea water and sulfur in the fuel. Sodium chloride from the sea appears to be necessary to make the latter source a threat, because it reacts with the sulfur to form the necessary intermediate corrodant, sodium sulfate [3]. The first solutions are to reduce the salt introduced via air and via contaminated fuel as much as possible, as well as using a low-sulfur fuel. But even with the best of such measures, the marine turbine requires either coatings that will resist the corrosive attack, or hot parts made totally of resistant steel alloys.

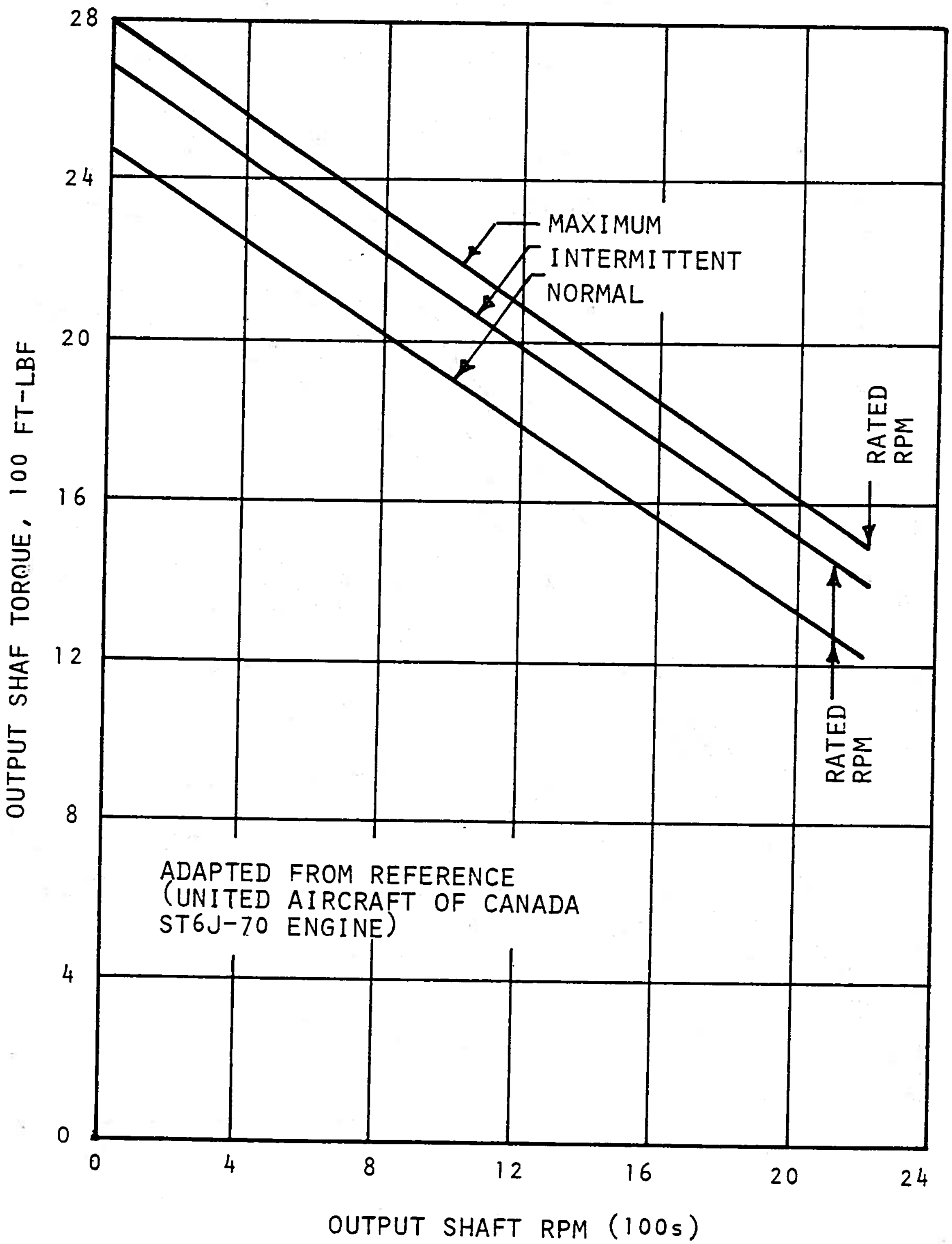
The aircraft gas turbine burns "jet fuel", which is approximately the kerosine of common terminology. This fuel is also used as a marine gas turbine fuel, but the marine engines are more often expected to burn grade 2 diesel oil, or perhaps the Navy's multipurpose fuel. When an aircraft turbine is adapted to marine service, it is consequently often necessary to modify the combustor spray nozzles for the more viscous fuel.

5.3 TORQUE-RPM, POWER-RPM AND FUEL RATE CHARACTERISTICS

The ideal torque-RPM relationship for a single-stage impulse turbine is

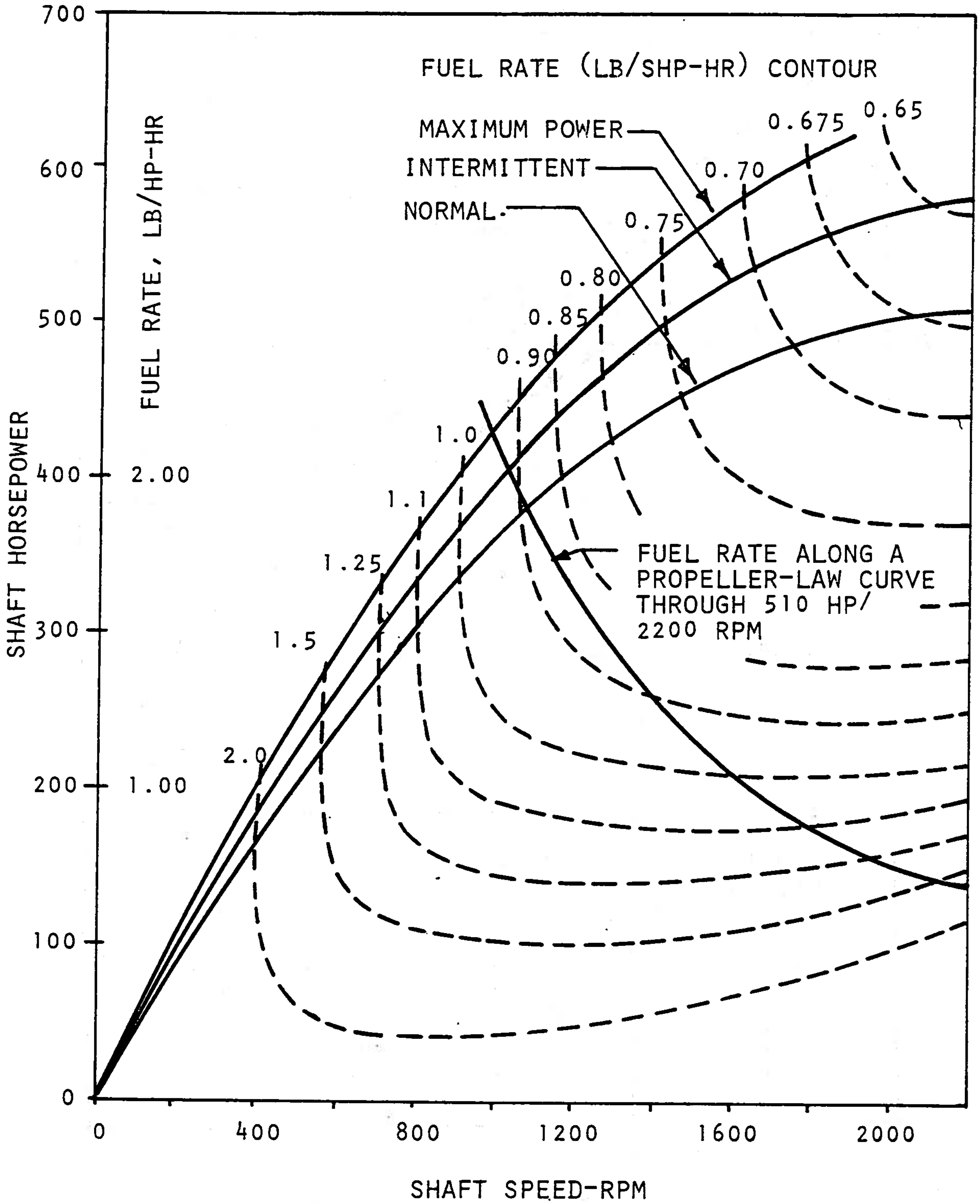
$$\frac{Q}{Q_r} = 2\left[1 - \frac{N}{2\,N_r}\right] \qquad (5.1)$$

where Q is torque, N is RPM, and the subscript r denotes rated conditions. This equation is that of a straight line, with $Q = Q_r$ when $N = N_r$, and $Q = 2Q_r$ when $N = 0$. The actual turbine may be neither single-stage, impulse, nor ideal, yet the characteristic is likely to be quite near the prediction of equation (5.1). Figure 5.5 is a set of actual torque-RPM curves for a small-craft marine gas turbine, as published by the manufacturer. Figure 5.6 contains the corresponding power-RPM curves. It is evident in the latter figure why a turbine is called "a constant-power machine," for the rise in torque tends to componsate for the decline in RPM as the operating point moves to the left. Over a limited range of RPM near the rated point, power does indeed stay approximately constant.



Adapted from United Aircraft of Canada
literature (ST6 engine)

FIGURE 5.5 Typical Torque-RPM Characteristics for a Small-Craft Gas Turbine Engine



Adapted from United Aircraft of Canada
literature (ST6 engine)

FIGURE 5.6 Fuel Rate Map for a Small-Craft Gas Turbine Engine

Three curves appear in Figures 5.5 and 5.6, these representing three different levels of rating, or three different levels of combustion temperature, or three different fuel settings, for the engine. Part load curves, which do not appear, would be correspondingly lower.

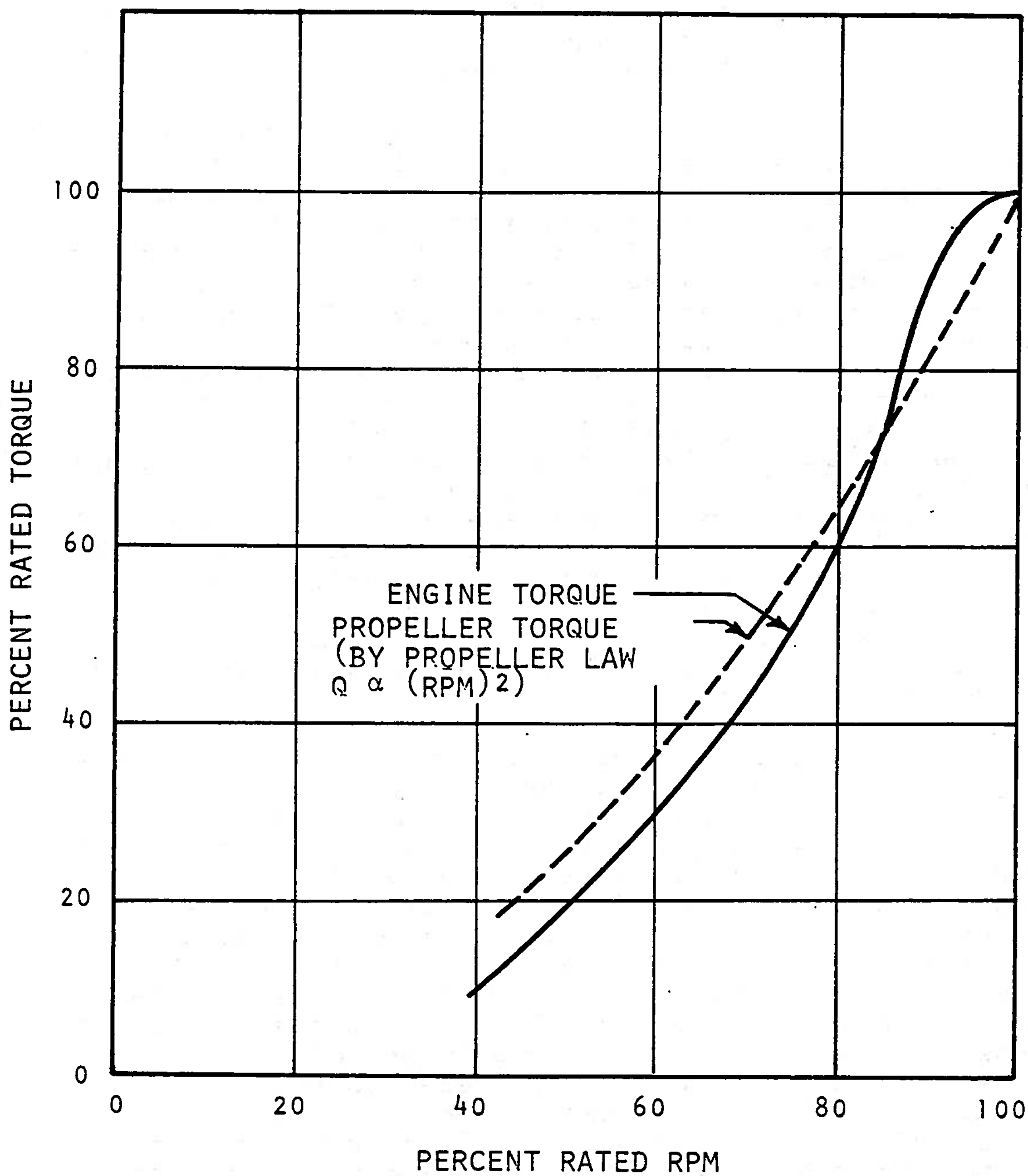
The ideal torque analysis for a turbine which is part of a single-shaft engine is much more complicated than that of the free turbine, since compressor speed, and hence mass flow rate and pressure ratio, must vary with the turbine speed. It is not attempted here, but Figure 5.7 gives the story to an adequate degree. The curve in this figure is the locus of maximum torques available at each RPM for a single-shaft engine, calculated from power-RPM data given in Reference 4. A propeller-law load curve is included in the figure, intersecting the engine curve at an assumed rated condition of 100% torque, 100% RPM. Over much of the RPM range, load torque is <u>above</u> driver torque, meaning that the engine could not accelerate the propeller to the rated point, although a controllable-pitch propeller or higher torque rating for the engine (i.e. whole curve slid up) could take care of the problem. In actuality, the problem might be worse, however, for the load curve might not be the cubic, but might show the "hump" of a typical planing-boat curve. This is a major reason why the single-shaft engine is rarely seen in marine propulsion service.

Figure 5.6 includes fuel rate contours. If you compare these contours to those shown for a typical diesel engine in Chapter 3, you will observe that the gas turbine fuel rates grow worse much more rapidly than the diesel as the RPM and power are reduced. An auxiliary curve is plotted on the figure, this being a plot of the fuel rate vs. RPM as it would occur along a cubic propeller curve (not shown). Note that at half speed, the fuel rate is at least three times what it is at rated speed. With a diesel, the ratio would be no more than one and a half. When a vessel powered by a gas turbine is to run for lengthy periods at low speed, a "cruising turbine" is often considered to forestall this great gulping of fuel. The second engine, to be clutched in to furnish the cruising power, will operate at or near its rating in this condition, and so have a much better fuel rate than the main engine operating at a fraction of its rating.

5.4 FUEL

The most critical properties of marine gas turbine fuels are said [5], [6] to be these:

1. Volatility. This affects ignition (starting), length of flame, clean burning.



Calculated from power-RPM data in Reference[4]

FIGURE 5.7 Torque-RPM Behavior of a Single-Shaft Gas Turbine

2. Metal impurities. Vanadium and sodium, in particular, react with sulfur in the fuel to form corrosive compounds. Other metals form compounds that can clog filters and fuel nozzles.

3. Sulfur. It is the obvious culprit in "sulfidation", mentioned previously, but as noted just above, requires help from other impurities.

4. Stability. The fuel may be subjected to quite high temperatures as it approaches the burners; stability is the ability to resist breaking down under this condition into sludgy stuff that could clog burner nozzles.

5. Combustion characteristics. The aim is to burn without intermediate breakdown into solid carbon particles. These may impinge on turbine blades, and by their incandescence cause local over-heating.

6. Contaminants. This can mean a lot of things, such as water, dirt, even bacteria. Salt water introduces sea salts. Anything that by clogging fuel passages upsets even distribution of fuel to the combustion chambers can cause overtemperature in the localties that get more than the average fuel flow.

The traditional gas turbine fuel has been "jet fuel" which is essentially kerosine, a light, clear, low viscosity, petroleum distillate. Its properties are quite satisfactory with respect to the above list, as long as it is handled carefully to avoid contamination. The same can be said with respect to grade 2 diesel oil, although its impurity level is inherently higher, and its viscosity being higher may require different fuel nozzles when an engine is converted to this fuel (Section 5.2). It is cheaper than the jet fuel, so there is economic incentive to use it, especially when the gas turbine may be in competition with the diesel engine.

The Navy is developing a single fuel for all its power plants, "navy distillate." Its properties are specified by MIL-F-24397 (MIL = military specification). Some of these properties, from the current issue of this specification, are

Viscosity, SSU @ 100F	58.5 max
Gravity, degrees API @ 60F	27 min
Flash point, degrees F	150 min
Carbon residue, 10% bottoms, %	0.4 max

Sulfur, %	1.3 max
Vanadium, ppm	0.5 max

Properties of diesel fuel have been given in Chapter 3.

5.5 CORRECTION FACTORS

All internal combustion engines are sensitive to the density of atmospheric air, and generally their ratings must be reduced when this density becomes less because of changes in atmospheric pressure and temperature, or because of artificial influences such as pressure loss in intake ducting. This aspect was not mentioned in the chapters on diesel and spark-ignition engines, it being reserved for the chapter on rating. It is discussed here for the gas turbine, however, because the gas turbine is much more sensitive than the other two, and its problem is compounded by the necessity for handling large flow rates of air and exhaust gas, a necessity which makes it difficult to avoid significant pressure loss in the ducting between engine and atmosphere.

The sensitivity of the gas turbine to pressure loss can be explained briefly by two factors:

1. The highest pressure in a gas turbine engine is comparatively low compared to other engine types. A pressure ratio of 10 might be used as typical, and this means that the pressure available to drive the turbine with this ratio is, at the very most, 147 psia. Peak pressures in the other internal combustion engines, and indeed in a modern steam plant, are many times higher than this. A 1 psi (say) loss in a gas turbine is therefore a much larger fraction of the pressure available.

2. The net power output of a gas turbine is the difference between its gross output, and the power required internally to drive the compressor. Qualitatively, the same is true for the reciprocating engines, because they too compress their working fluid, but the fraction of output that must go back into compression is a smaller part of the gross output. To illustrate, a gas turbine might have a gross output of 100 hp, but require 60 hp to drive the compressor. If a loss in pressure, in the exhaust stack perhaps, caused a 5% loss in turbine output, the compressor would still demand 60 hp, so that net output would be 35 hp rather than 40 hp. The loss to net output is consequently 5/40, or 12.5% rather than 5%. The opposite extreme occurs in a steam power plant, whose compression work (i.e. power to run the feed pump) is so small that a 5% loss in gross output is tantamount to a 5% loss in net output.

In addition to density, the temperature itself of inlet air is an important influence on gas turbine output. Hotter air in generally means hotter air to the turbine. Since the turbine is typically designed to run at about the hottest temperature feasible in the interest of highest possible efficiency, an above-design inlet air temperature means that fuel flow must be cut back to avoid overheating.

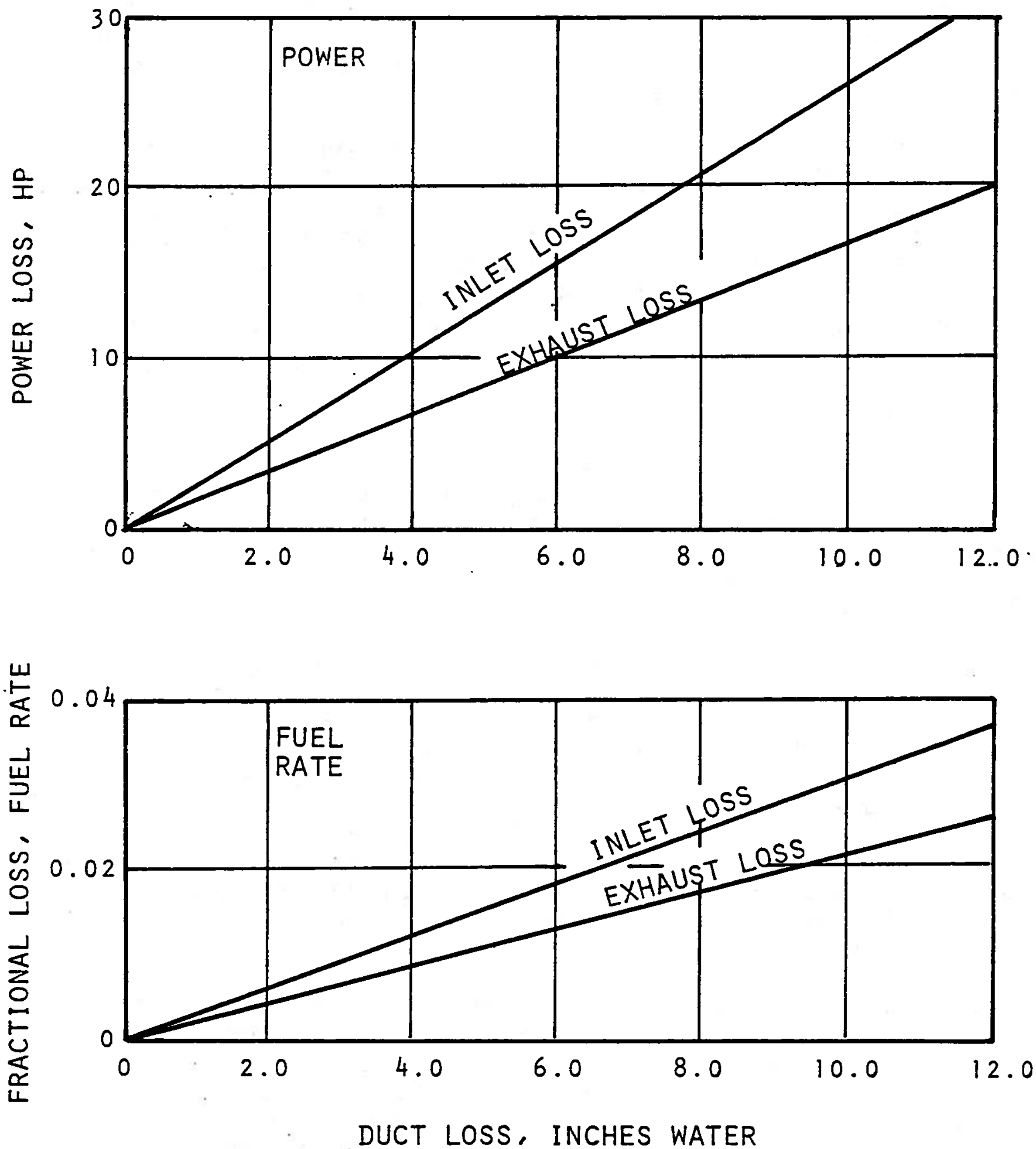
Corrections for the conditions mentioned here are usually published by the engine builders. Figure 5.8 gives correction curves for power output and fuel rate as functions of duct losses for the United Aircraft of Canada ST6 Engine. Figure 5.9 gives corrections to power output for the same engine, due to ambient air temperature, and Figure 5.10 gives the fuel rate corrections that accompany the change in output.

5.6 INTAKE AND EXHAUST

The intake of air for combustion and working fluid, and the exhaust of the air plus burnt fuel, are vital considerations for any type of combustion power plant. It is no more important for the gas turbine than for any other, but this feature of the plant does have more impact on the total plant, and require more effort in design, than in any other type. One reason is that the gas turbine consumes more air than any other; four times as much as a diesel of the same horsepower might be accommodated simply by squeezing it through ducting at higher velocity than used with diesel, but such an idea is immediately defeated by the high pressure drop that would result; the sensitivity of the gas turbine to intake and exhaust pressure drops has been mentioned just above. The inevitable result is that gas turbine ducting is impressively large in cross section. It is likely to be the most space-consuming feature of the entire power plant, so that the designer has incentive to design carefully to keep its size from being any bigger than necessary.

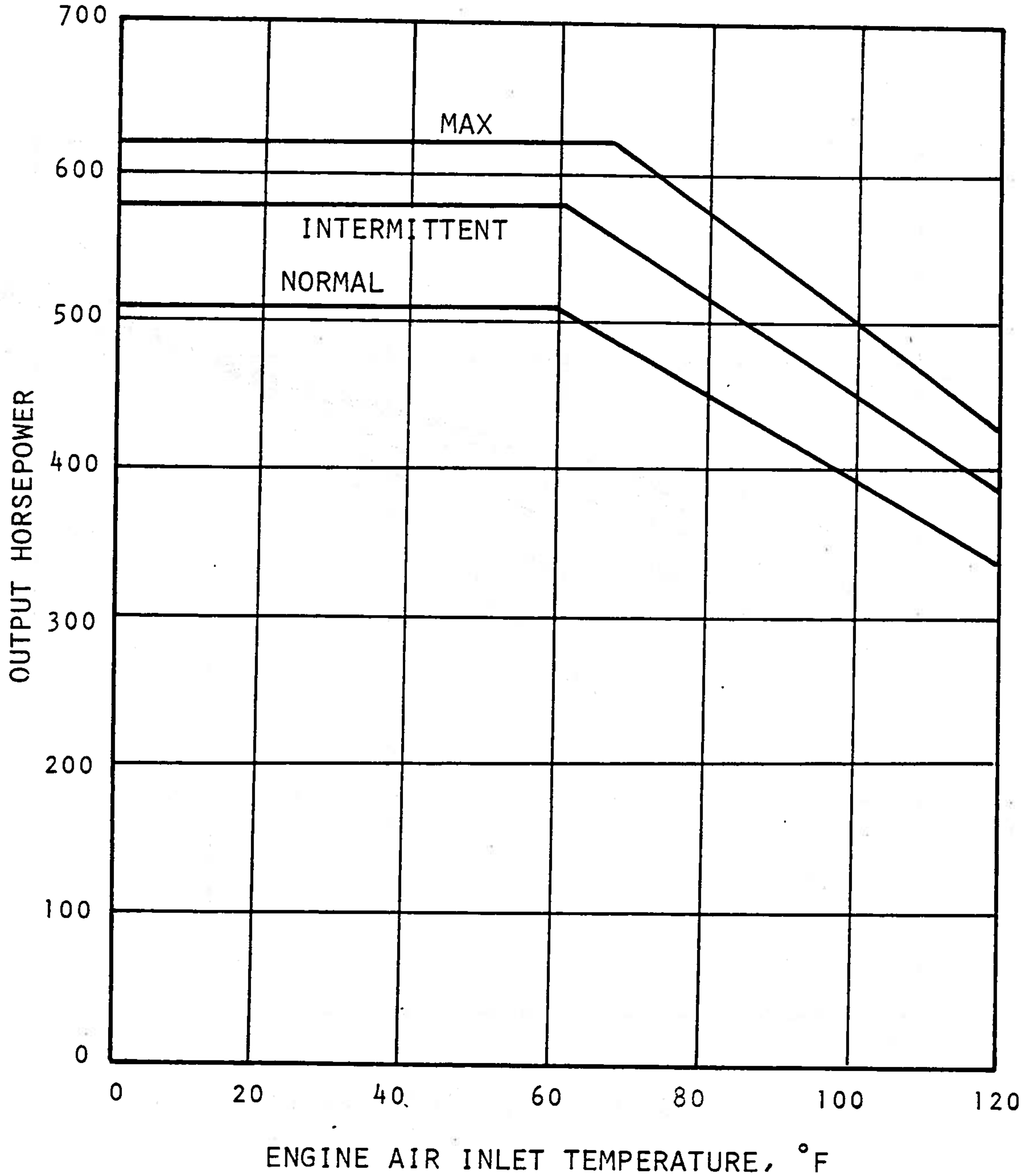
The intake problem is aggravated by the entrained salt. Low velocity helps the salt problem simply by encouraging settling of the particles, and this encourages large cross section. Changes of direction, filters, and inertial separators are often found to promote removal of the salt before it reaches the engine, but all of these add pressure loss, and so must be kept to a minimum, or the duct must be made yet bigger to compensate for the loss. Figure 5.11 shows installation of gas turbines in two small vessels, and illustrates the use of tortuous paths of large cross section for a low-salt intake system.

The designer's problem of analysis is basically one of pressure drop calculation to determine the cross sectional area of the ducts. He may design to some fixed pressure



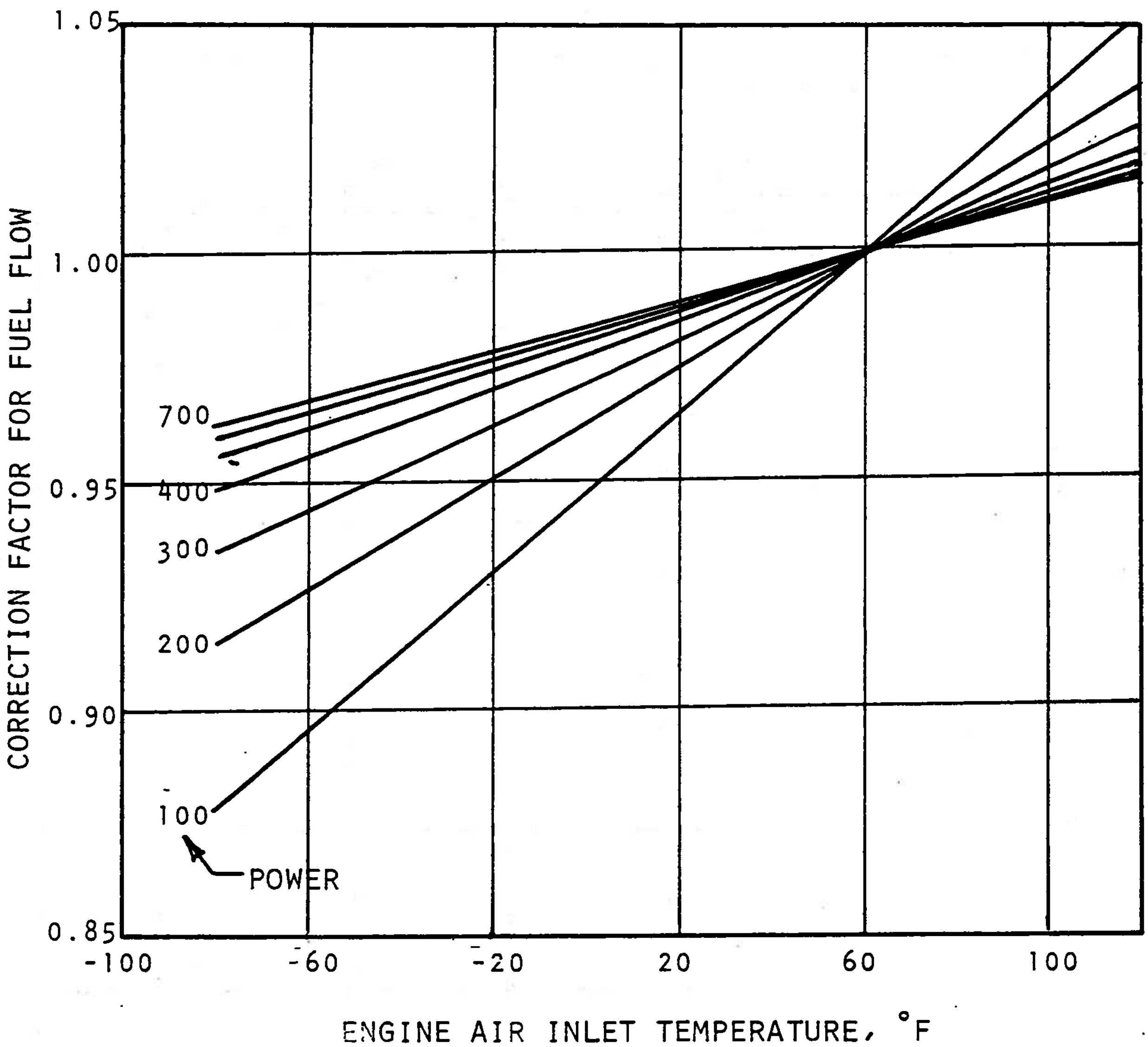
Adapted from United Aircraft of Canada literature (ST6 Engine)

FIGURE 5.8 Power Loss and Fuel Rate Loss Due to Ducting Losses for Small-Craft Gas Turbine Engine



Adapted from United Aircraft of Canada literature (ST6 engine)

FIGURE 5.9 Effect of Ambient Temperature on Output for Small-Craft Gas Turbine Engine



Adapted from United Aircraft of Canada
literature (ST6 engine)

FIGURE 5.10 Ambient Correction Factors for a Small-Craft Gas Turbine Engine drop; for example, the Navy uses 4 inches (of water) inlet pressure drop, and 6 inches exhaust pressure drop, in rating propulsion turbines [7], and this is a good indication of what might be taken as a reasonable set of values to design to. On the other hand, the designer might go into quite a trade-off exercise, trading the benefit of smaller ducts against the cost of loss in engine performance. In either case, his tool is pressure drop calculation, and usually the standard techniques of ventilation duct calculations are used. The techniques and the data needed to implement them are found in engineering handbooks, navy design data sheets, and References 7,8.

Since velocities in the ducting may turn out to be quite high, it has been questioned whether the standard ventilation calculations, which assume the air to be incompressible, ought not to be replaced by analysis that accounts for compressibility of the air. It is claimed in Reference 7 that the incompressible analysis is conservative, i.e., puts the design on the safe side, and the designer might be content with this assurance. But if he is interested in refinements, he might start by consulting Reference 9 , which is a navy-sponsored investigation of duct calculation methods. One helpful product is a chart of correction factors that can be applied to the results of imcompressible calculations. The chart is reproduced in Reference 8 also.

In addition to an acceptable pressure loss in the intake ducting, an "eveness" of flow approaching the compressor inlet is important. If the pressure is much different at one point of the inlet face from elsewhere, the first stage compressor blades may see this anomaly as an impulse on each revolution, and dangerous vibrations may be excited in consequence. It is difficult to predict such conditions by mere calculation, so that construction and testing of a model of the inlet ducting is often resorted to. Smoke traces in a model constructed of transparent plastic can show up any maldistribution of flow at the compressor inlet face.

Thermal expansion of the exhaust ducting is a problem because restrained expansion can put dangerous thrusts and moments on the engine. An expansion joint is required if the duct is anchored to vessel structure. An alternative that seems particularly suited to small craft is the eductor stack. Figure 5.12 gives a visual description of the concept; note that the eductor also provides ventilation exhaust for the engine compartment without the need for an exhaust blower. Design rules for these devices are not widely published, but may be found in a navy guidebook [10].

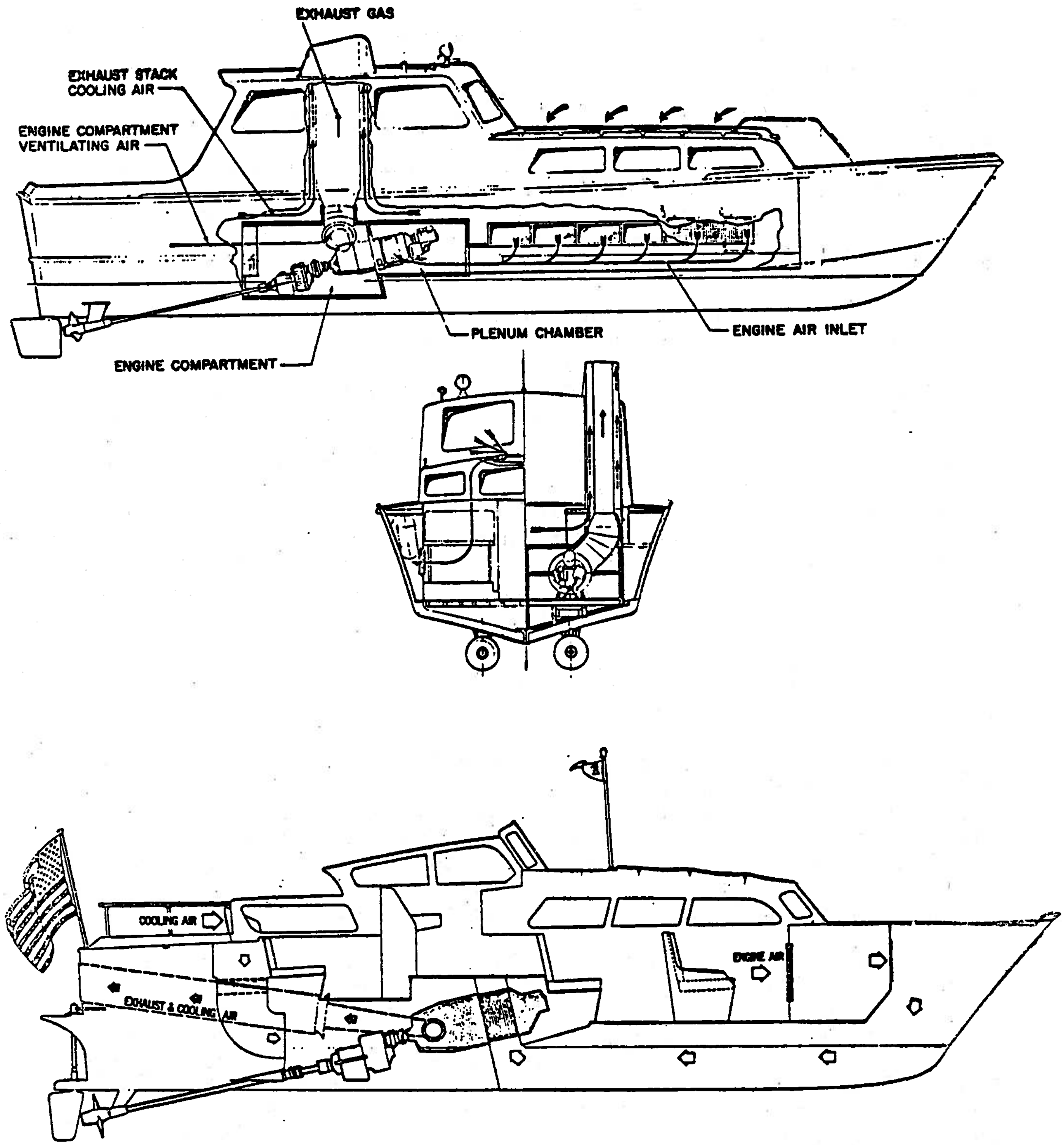


FIGURE 5.11 Examples of Intake and Exhaust Arrangements for Small-Craft Gas Turbine Engines (from Reference[1])

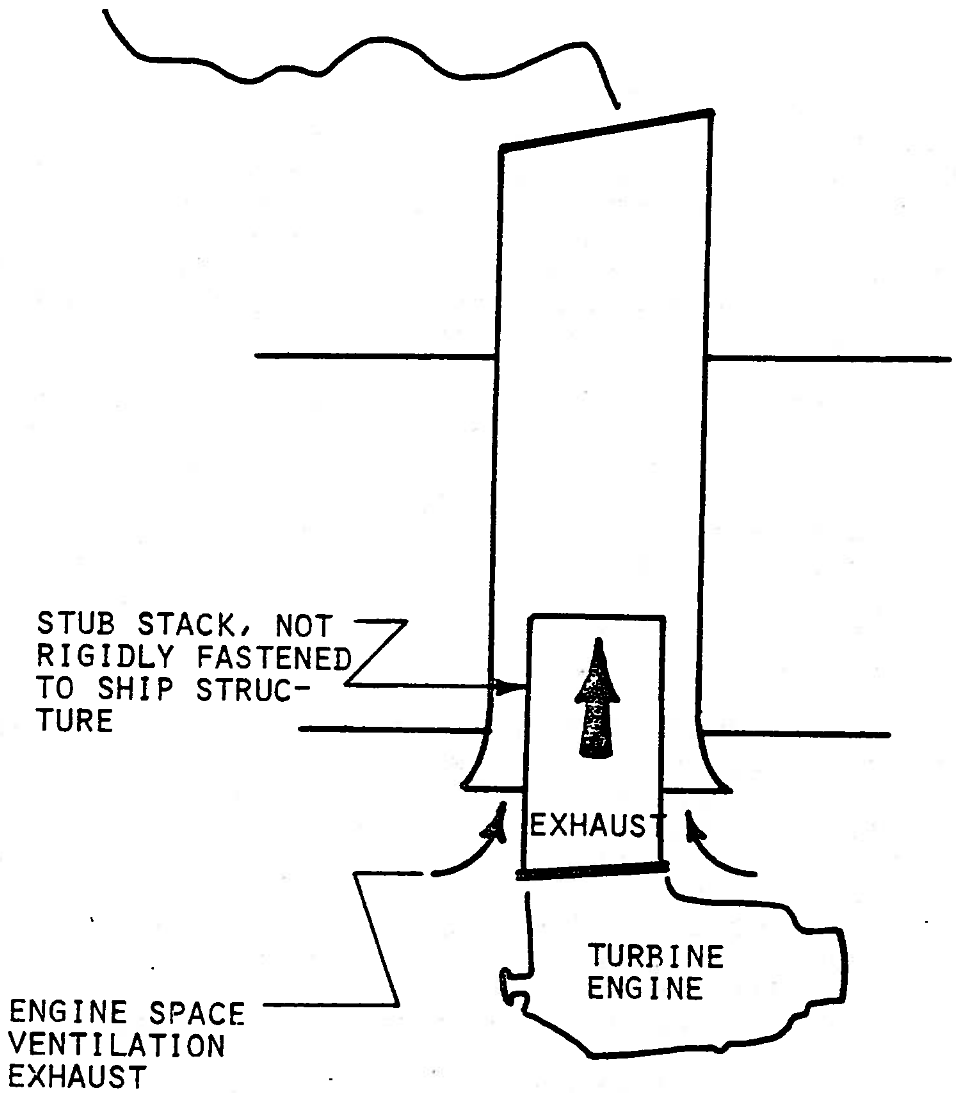


FIGURE 5.12 The Eductor-Stack Concept for Gas Turbine Exhaust

5.7 NOISE

As a general rule, a gas turbine engine must be surrounded by sound-absorbing material, and its intake and exhaust ducts must be treated since they are ready pathways to the outside for noise.

The machinery space of a small craft is not likely to be manned during operation, so that the engine itself will not be shielded, but the surfaces of the compartment almost certainly will need covering by sound-absorbing material to make adjoining compartments habitable. Fiberglass insulation, protected if necessary from mechanical damage by perforated sheet metal, is typically used. It should be noted that the predominant sound from the engine is high-pitched, making it relatively easy to absorb on suitably treated surfaces.

Bends in the intake and exhaust ducting are a considerable help in preventing escape of sound by those paths, since they compel the sound waves to impinge on surfaces where they may be absorbed. Absorption is greatly enhanced by a covering of absorbant material, as mentioned in the preceding paragraph. In the case of the exhaust duct, it must be remembered that the gas is quite hot (perhaps 1200F), so that any absorbing material used must be able to withstand the temperature.

The familiar muffler that appears in the exhaust line of diesel and spark-ignition engines is not used with the gas turbine because the high-pitched nature of its sound makes surface absorption much more appropriate; the entire inner surfaces of intake and exhaust ducts serve as the muffler when covered with absorbent material. In some cases, the surface may not be sufficient in area, particularly in the exhaust duct. It then can be increased by introducing splitters, or indeed added area in any form, within the duct, this area to be covered with the same sound-absorbing material.

The Navy has a guidebook for quieting gas turbines [11], and detailed material on this topic can also be found in Reference 7.

5.8 CONTROL

Control of a gas turbine is more complicated than that of a diesel or spark-ignition engine, and for several reasons. The usual gas turbine will be of the two-shaft type, and controls will have to operate on both shafts since they are not mechanically joined. Both over speed and over temperature are dangers. The starting operation is more complicated, and so may be appropriately handled by an automatic control. Attitude of turbine nozzles may be controlled quite independently of fuel, and there is nothing comparable to this among the other two engines.

In the typical two-shaft engine, a governor is set by the operator to maintain a particular RPM of the compressor turbine, while the power turbine is left free to find its own speed as determined by equilibrium with the load. This is commonly referred to as "power control" since maintenance of constant compressor speed is tantamount to maintaining constant power. However, the power turbine cannot be left completely unsupervised, since there is always the danger of overspeeding, perhaps caused by the propeller pitching out of the water. A topping governor is therefore always fitted to cut back the fuel should the power turbine overspeed. An overspeed trip that quickly shuts off the fuel completely in case of serious overspeed is also common in addition to the governor. Over temperature is also a hazard, and can easily be caused by reduced airflow due to a fouled compressor. Exhaust temperature is therefore monitored, and is an input to the governor to initiate a cutback in fuel flow if necessary. Over torque is possible with a gas turbine, because of the rising shape of the torque characteristic as speed falls, and sometimes the governor provides torque protection; it can determine relative torque by comparing fuel flow (proportional to power) and RPM. (The governor is sometimes called the fuel controller when it handles these extra chores in addition to speed control.)

An alternative to "power control" is "speed control", which implies that the governor senses power turbine RPM rather than compressor turbine RPM. In this case, the compressor turbine must have a separate governor to protect it from overspeeding.

The gas turbine is started by spinning it up to firing speed with some form of starting motor, plus ignition by a spark plug or equivalent. Purging by spinning the compressor for an interval before fuel is admitted admission of fuel at the proper time, shutoff of fuel if ignition fails to take place within a safe time, and repurging before further attempts at starting, are parts of the starting process. To obviate the need for a highly skilled and nevercareless human operator, a starting sequencer is often included as part of the controls, its function being to carry out all of the starting steps automatically upon receipt of the simple signal "start" from its human boss.

It has been mentioned that the gas turbine has a comparatively poor fuel rate at part load. This situation can be ameliorated somewhat by incorporating controllable nozzles for the power turbine. These nozzles are formed by passages between airfoil vanes around the periphery of the turbine inlet; if they are rotated, the area and direction of the jet formed are changed. In this way, the direction and velocity of jet that is better suited to the part-load operating conditions can be formed, and so improved efficiency obtained. The necessary control is always automatic and "built in," i.e. not under direction of the operator (cf. automatic spark advance in an automobile engine).

Control of the engine usually must be integrated with that of the remainder of the propulsion system, e.g. clutches, propeller pitch. This aspect of the control task is discussed in Chapters 6 and 8.

5.9 REFERENCES

1. G. Hardy, "The Development of a Marine Version of the United Aircraft of Canada PT6 Engine," ASME Paper 66-GT/M-31

2. W. H. Lindsey, "The Marine Gas Turbine," Proceedings, Society of Naval Architects and Marine Engineers, Diamond Jubilee Meeting, 1968

3. Charles Miller, notes for University of Michigan short course, "Small Craft Design," 1969

4. Alan W. McCoy, "Gas Turbine Propulsion for High-Speed Small Craft," Marine Technology, July 1968

5. A. V. Cabal, J. H. Warren, "Fuels for Marine Gas Turbines," ASME Paper 67-GT-50

6. G. A. Carleton, "New Fuel for Naval Gas Turbine Engines," ASME Paper 70-GT-29

7. Naval Ship System Command, NAVSHIPS 0941-038-7100, Installation Design Criteria for Gas Turbine Application in Naval Vessels

8. John B. Woodward, Hydraulics for Piping Design, Report 103, University of Michigan Department of Naval Architecture and Marine Engineering

9. W. T. Sawyer, W. J. Kelnhofer, R. A. Smith, "Design Considerations for Marine Gas Turbine Ducting," ASME Paper 66-GT/M-27

10. Bureau of Ships Index NS 622-078, Eductor Design Manual

11. Naval Ship Systems Command, NAVSHIPS 250-645-3, General Criteria of Ensuring Quiet Gas Turbine Shipboard Installations

CHAPTER 6

MATCHING OF ENGINE AND PROPELLER

6.1 INTRODUCTION

In its simplest manifestation, matching means selecting a propeller pitch (or pitch ratio, which is pitch/diameter) that allows the engine to develop its rated power. Let's use the diesel engine as a discussion example, and recall that the power-RPM characteristic is a straight radial line whose slope is proportional to torque or BMEP. There will be a particular slope representing the rated BMEP; rated power lies at the intersection of the corresponding radial line and the line of rated RPM.

Perhaps an aside is in order to define the word "rated" that must be used frequently here. It is the subject of the next chapter, but meanwhile it can be taken to mean "that value of BMEP, RPM, or power that the designer intends should be used in normal service." It is also implied that the rated value is one not to be exceeded because of possible overheating, rapid wear, excessive smoke, and other deleterious results of excess.

Now, for rated power to be developed, both rated BMEP and rated RPM must be reached simultaneously. Figure 6.1 shows what can happen. The intersection described in the first paragraph lies at the "corner" in the upper right. This is where the engine will operate at rated fuel setting, and so develop rated power if the load (i.e. propeller) curve passes through the point. Three propeller power-RPM curves (cubic curves, according to the "propeller law"), represent the behavior of propellers differing in pitch, but otherwise identical. If pitch is too low, rated RPM is reached before rated BMEP, and the engine consequently must be throttled back to avoid overspeeding; if too high, rated BMEP is reached first. In the latter case, the engine may be inadvertently overloaded by the operator in an attempt to reach the expected RPM, unless the limit of throttle setting or fuel rack position is reached near rated BMEP. In both cases, the figure shows how rated power--what the engine was bought to produce--cannot be reached. Only the correct pitch allows the engine to exploit its full capability.

Fundamentally, the process is therefore a simple one of picking the correct propeller pitch. But several complicating factors are to be considered, and their discussions make necessary the bulk of this chapter. For example, the correct pitch as demonstrated just above may not be the one that gives the highest propeller efficiency, necessitating a designer's decision on whether to favor engine, propeller, or some compromise. Also, the differences that can occur between design conditions of hull surface, weather, etc, and those found in service, make it desireable to adjust the point of matching to compensate. An auxiliary load such as a pump or generator must be considered in the matching, unless it is a negligble addition to propeller load. And the controllable pitch propeller gives some additional points to discuss.

6.2 THE GENERAL MATCHING PROCESS

The basic idea has been expressed in the preceding section: pick propeller pitch so that the engine will develop its rated power. But propeller efficiency is a function of pitch, and it may be that the pitch chosen to suit the engine doesn't produce the highest possible propeller efficiency. This potential problem is avoided in many cases because a variety of stock model reduction gear ratios are available (e.g. ratios varying from 1.5:1 to 6:1 are typically available); an RPM at reduction gear output coupling can have a number of different values for the same engine RPM, and consequently it is often possible to pick a propeller RPM that is near that of highest efficiency, while satisfying the engine. But this is not always the case, and it is seldom that the high cost of a non-stock reduction gear is justified. An example illustrates the possible dilemna.

A small research vessel requires 375 shp for its design speed of 10 knots. A propeller as large as 6.0 feet in diameter can be fitted, and since it would be more efficient than any one smaller, the designer tentatively plans to use this size. His propeller selection chart shows that the highest efficiency occurs at about 240 rpm. Figure 6.2 shows the efficiency of the propeller for different choices of RPM at 375 shp, and the same information for two propellers of smaller diameter.

To keep the example simple, let's assume that the choice of engine is limited to those of one builder. Among its offerings two possibilities are to be considered, with the following basic information:

Engine	A	B
Rated SHP	375	460
Rated RPM	1800	1800

At first glance, engine A looks like the obvious choice since its power is just right. But to give 240 rpm at the propeller, a reduction ratio of 7.5:1 is required; the

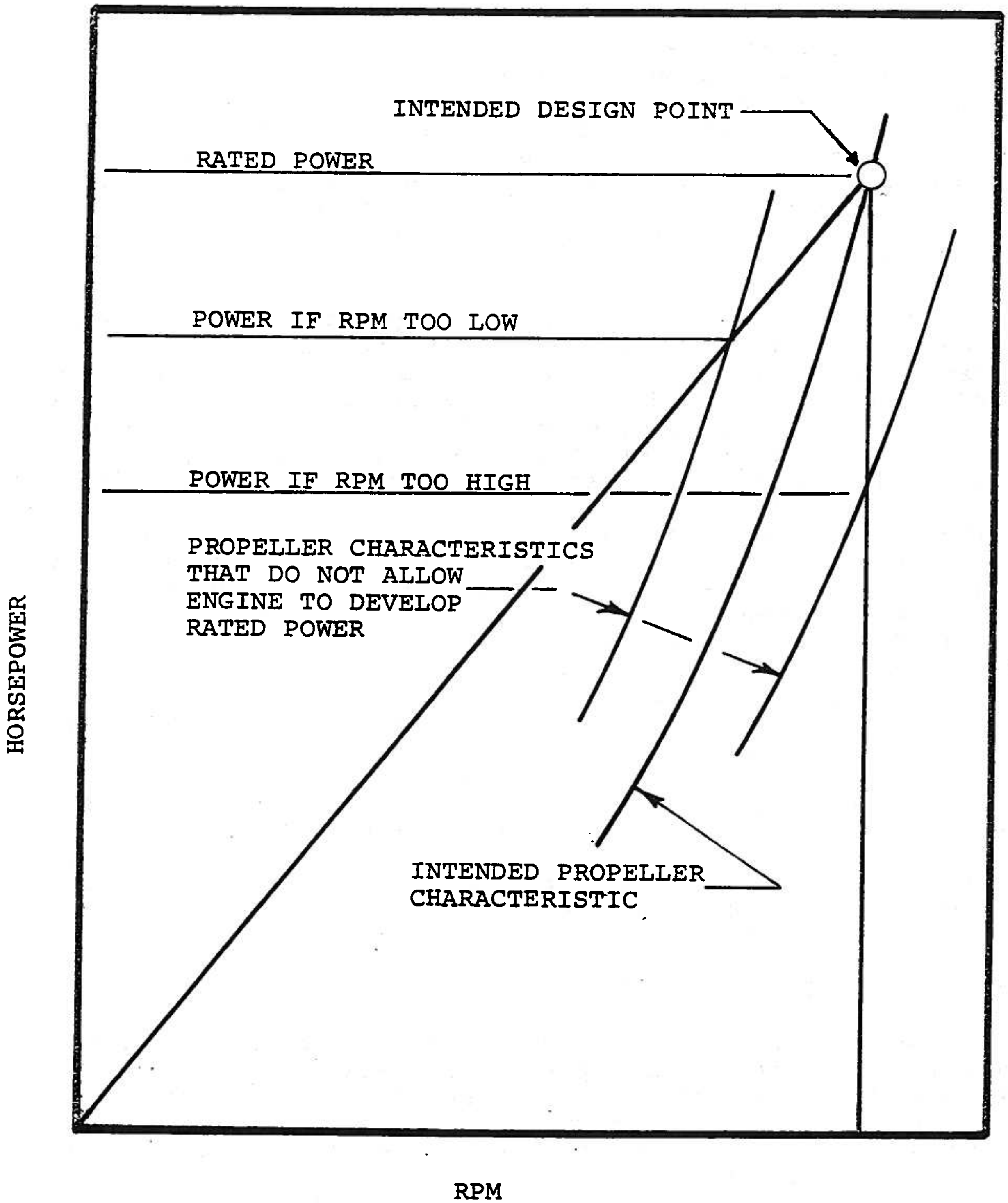


FIGURE 6.1 "Hitting the Corner" with Propeller Characteristic

highest ratio among stock gears for this engine is 5.86:1, and this multiplied by 250 gives 1465 rpm, an RPM which does not allow the engine to develop 375 shp. However, if the designer insists on maximum propeller efficiency, he can use engine B, for (1465/1800) x 460 = 375 shp. Provided that there is no objection raised by the engine builder to operating at the low RPM with rated BMEP, the problem is solved, except that a bigger and more expensive engine than necessary has to be used.

Two alternative solutions are suggested by a look at Figure 6.2. One is to use a smaller propeller. As the diameter decreases, the RPM of maximum efficiency increases, and it is easy to find one that has its maximum at 1800/5.86 = 307 rpm. But the figure shows that the peak efficiency will be less than the peak efficiency of the 6-foot propeller, and indeed may be less than that of the 6-footer operating at 307 rpm. If the solution is to use the biggest propeller with pitch reduced so that it will turn at 307 rpm when absorbing 375 shp, then the figure shows that efficiency is only about 1.5% less than the maximum. This solution does seem to be a reasonable one, since it allows use of the cheaper engine at the cost of such a small efficiency loss. In the absence of other information, I would say it is probably the best choice among the three possibilities offered. However, there <u>is</u> the possibility that the fuel savings resulting from that 1.5% efficiency difference could pay for a bigger engine. The designer should at least take a look at this possibility and also consider the first-cost savings of a smaller propeller. But, to repeat the general message of this chapter, pitch the propeller to let the engine develop its rating, unless there is definite evidence calling for favoring propeller efficiency instead.

6.3 ALLOWING FOR SERVICE CONDITIONS

Sea state, wind, hull roughness, propeller roughness, and draft all affect the speed-power relation of the vessel. In most instances, the changes that occur in service are unfavorable, e.g. the bottom is always rougher than the design condition. In terms of the power-RPM plot, this means that the propeller characteristic shifts to the left; it takes more power to push a fouled hull at a given speed, and a higher RPM of the propeller to develop that power. The consequences to the engine are readily seen in Figure 6.3: either the engine must slow down, thus losing power capability, or if RPM is to be maintained, the engine must be overloaded (still using the diesel as the example engine). There is little data published to indicate what the magnitude of this effect can be, but a 20% increase in resistance due to roughness over the life of an ocean-going merchant ship has been suggested [1]. For a small boat that is kept out of the

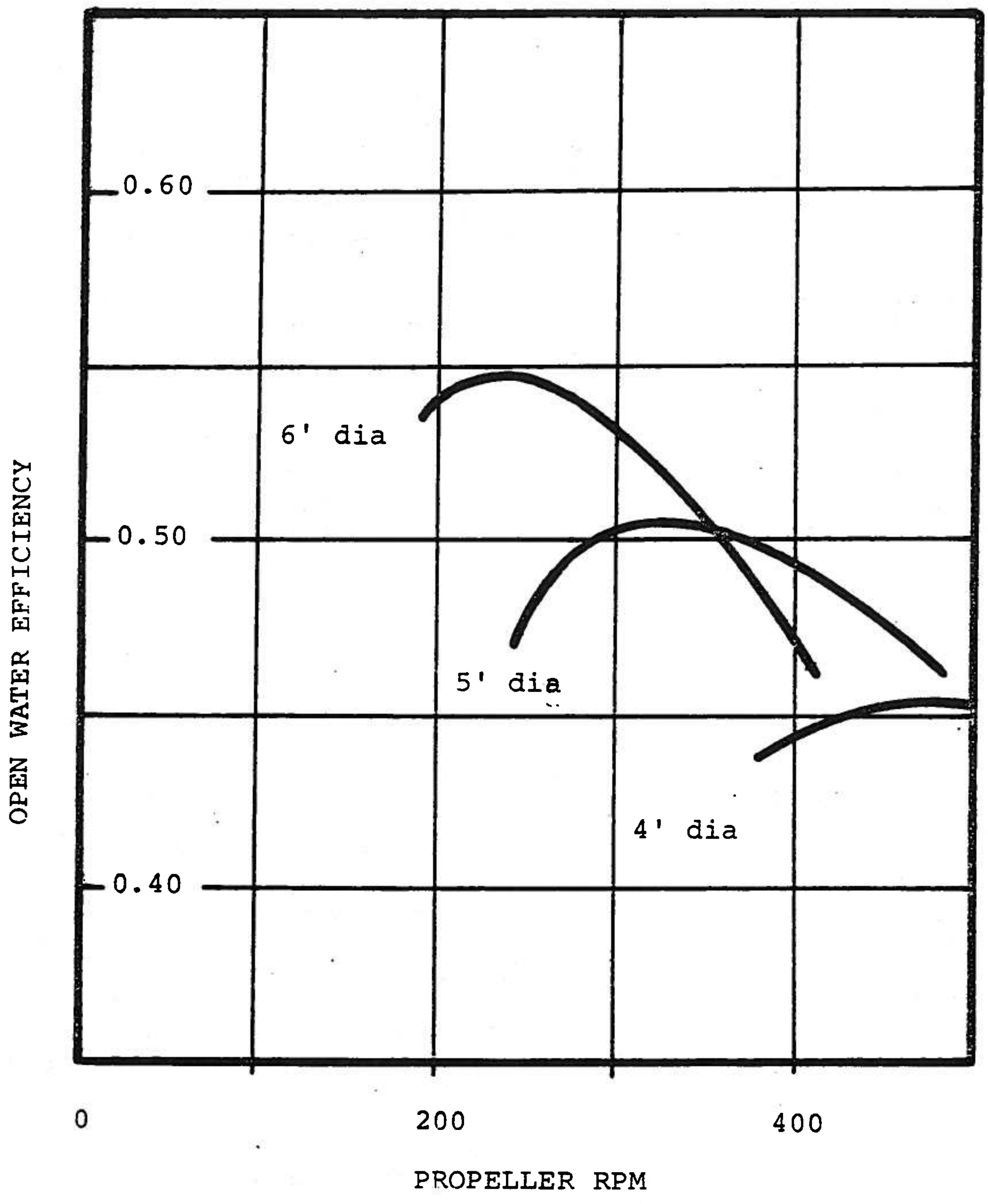


FIGURE 6.2 Propeller Efficiency as a Function of Design RPM - To Illustrate Example

water, and not abused, no increase at all seems reasonable. For illustrative purposes here, the 20% estimate is used in Figure 6.3, and it is seen that this requires about an 18% increase in BMEP to maintain the original RPM.

The remedy is to match the engine to the propeller under service conditions rather than under trial conditions. Figure 6.4 illustrates. Of course, it is difficult to say just what the service conditions will be, since they vary from day to day, for one thing. The equivalent process is to arbitararily select a margin in BMEP, and match the propeller under trial conditions at the resulting power. This is the process that is favored in Figure 6.4. Notice that the design point is set at 85% of rated BMEP, 100% of rated RPM. After this is first done, then the matching process proceeds as suggested in the preceding section. In service, worsening conditions can be met with constant RPM. If the 15% margin was the proper choice, then no need to exceed 100% BMEP in service should arise, save perhaps in extreme conditions. It is, however, difficult to say how much the margin should be, since it depends on speculations about future weather, hull conditions, loading, etc., as well as on the type and service of the boat. Please note that the 15% used here is only an example figure, and is not being recommended for use in any particular situation.

Inspection of Figure 6.4 shows that rated power cannot be attained on trial trip, assuming that this is run under design conditions. The figure does show an RPM margin that can allow the engine to overspeed somewhat on trials, but if this is made excessive, it is merely robbing the engine of RPM capability under service conditions. If the trial speed is the important one, as it may be in some cases, then the idea of a BMEP margin should be ignored. And it might be quite appropriate to do so for vessels, typically pleasure craft, that are not going to be used in bad weather, or that may have to slow down in rough water anyhow, and that do not have hull surfaces subject to noticeable roughening (e.g. fiberglas is not going to roughen due to corrosion as does steel).

If attention is switched to the SI engine, the message here is the same since the torque-RPM or power-RPM behavior is similar to that of the diesel. Things are different with the gas turbine, however. Recall from Chapter 5 that a turbine exhibits nearly constant power behavior near its rated point. As the propeller curve shifts to the left under service conditions, the RPM falls, but unless the deterioration is truly severe, SHP remains essentially constant. In consequence there is usually no incentive for providing the service margin.

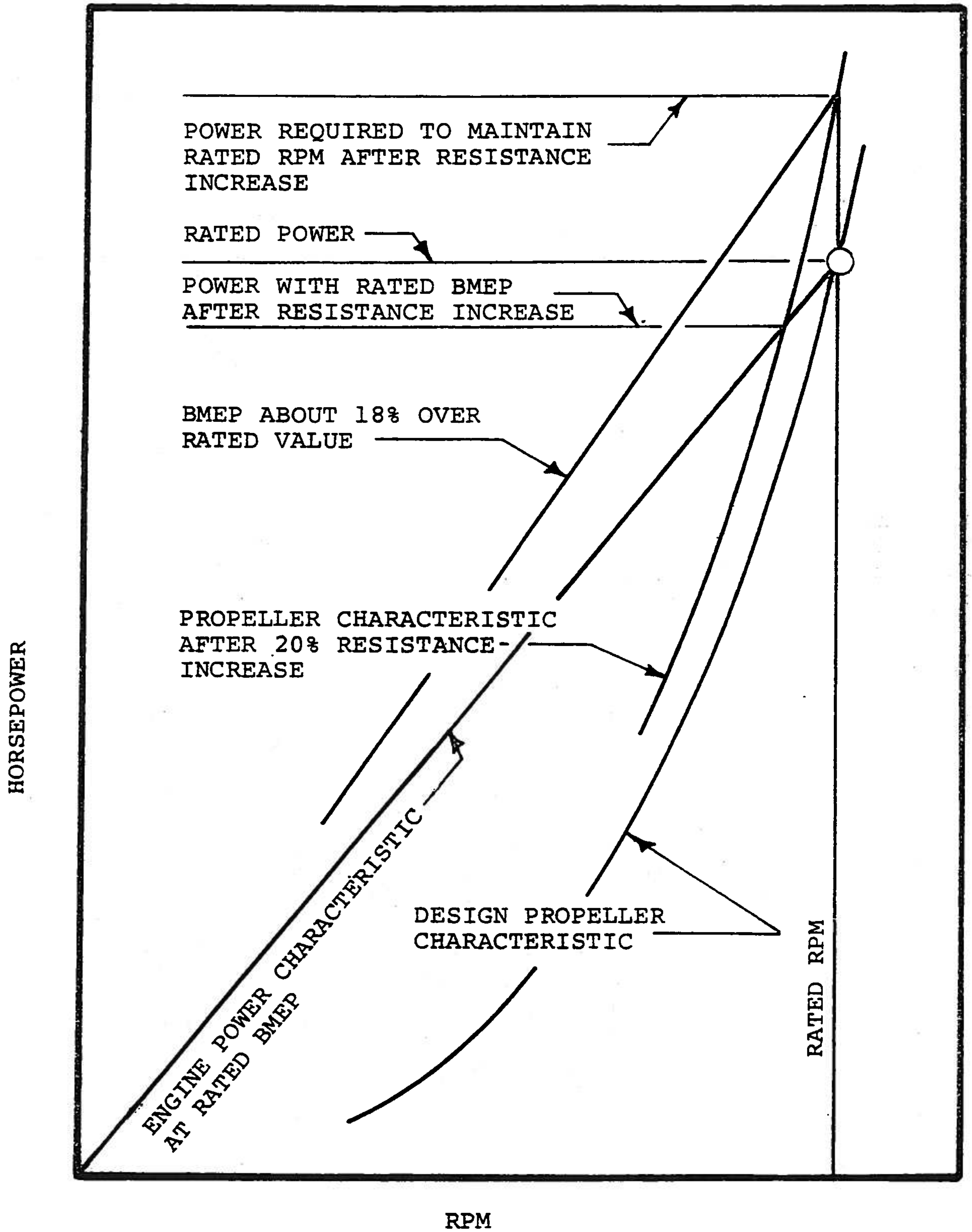


FIGURE 6.3 Effect of Deteriorated Service Conditions on Engine-Propeller Match

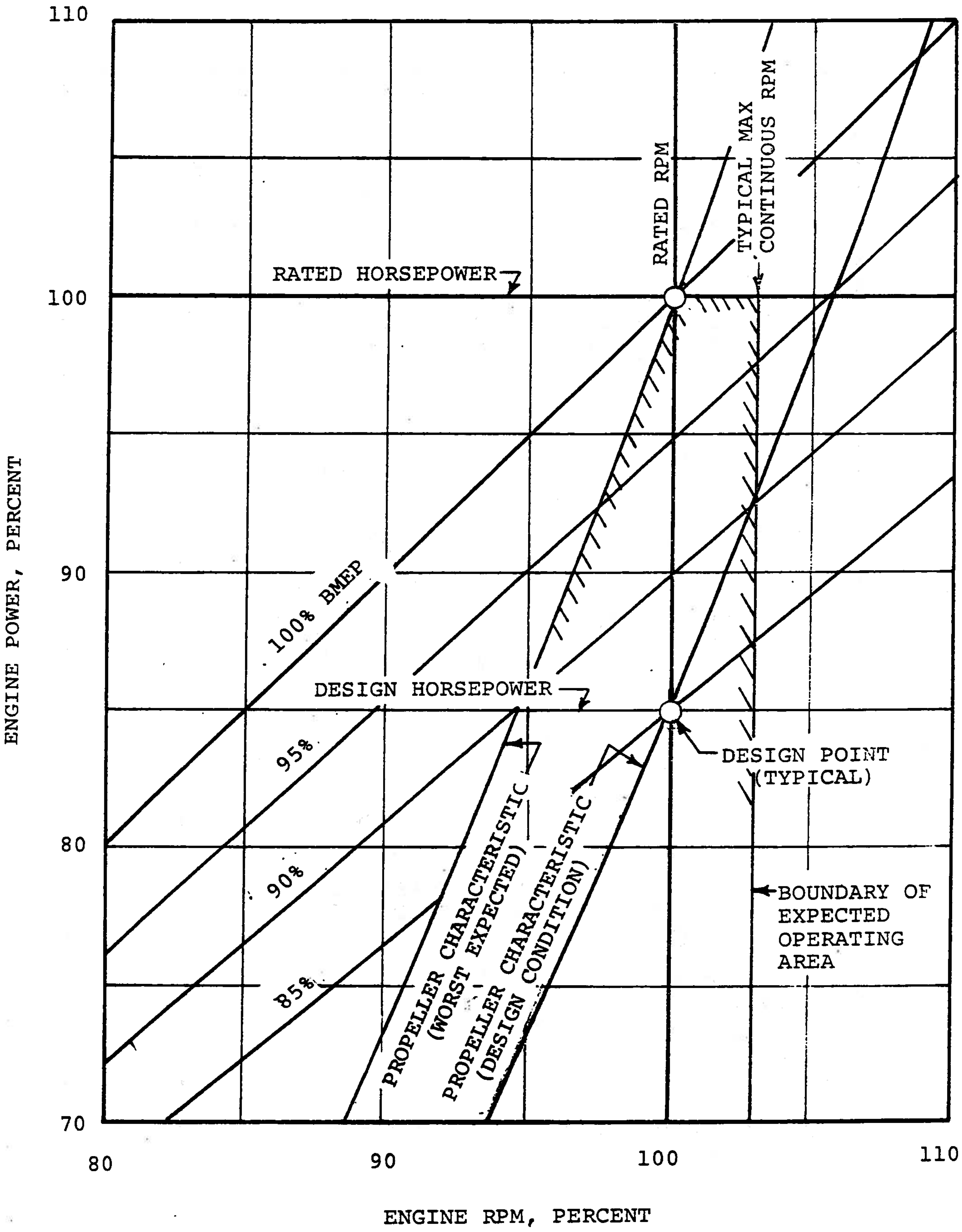


FIGURE 6.4 Inclusion of Service Allowance in Setting Design Point

6.4 NON-STANDARD PROPELLER CURVES

"Standard" here means the "propeller-law" cubic power-RPM curve that has been used for discussion purposes to this point. Recall from Chapter 2 that this is an approximation poor for some cases, especially those involving various types of planing boats. Their resistance curves display a hump in the vicinity of the transition to planing, and this hump also shows up in the power-RPM and torque-RPM curves. In extreme cases, the engine power-at-rated-BMEP curve will intersect the propeller curve at the hump as well as at the rated point. The result is an inability to accelerate past the hump speed, and consequently, a failure to plane. Figure 6.5 illustrates the danger. The propeller curve in that figure is plotted from Figure 2.4, using an assumption that RPM and boat speed are proportional. If this situation occurs, the designer must select an engine that can produce the required power at the hump RPM (he should be looking at the engine's actual torque curve, rather than depending on a straight line idealization seen in most of the figures here). Matching at the design point would then seem to be a secondary consideration, but he must nonetheless check what happens there. For example, the engine may overspeed because its power at rated RPM is more than the propeller can absorb. A governor would be required to protect the engine.

In the design process, about the only practicable way to find the power-RPM curve is by self-propelled model tests, but this is seldom indulged in for small craft. The consequence is that, unfortunately, the hump trouble is found by trial after the vessel is built. Back to the drawing board.

6.5 TOWING VESSELS

If a vessel takes on a tow, resistance increases and propeller RPM falls just as in the discussions of Section 6.3. Other considerations besides design margins are necessary, however, because the magnitude of the resistance increase can be so much greater. The loss in propeller efficiency may be a major factor, and the loss in RPM may be so severe as to justify measures stronger than mere design margin to allow the engine to develop its power capability.

Figure 6.6 shows what happens, on both EHP-speed and SHP-RPM planes. Look first at point A, assumed to be the design point, free-running condition. If the propeller curve shifts to the left as shown, and if the engine torque remains constant, the operating point shifts to the intersection at B. Because the engine RPM has fallen, the engine cannot produce its rated power. Speed of the vessel naturally is less because of the tow resistance, and because of the decrease in propeller efficiency (this loss is evident in the greater drop in EHP than in SHP), and because of this engine power loss.

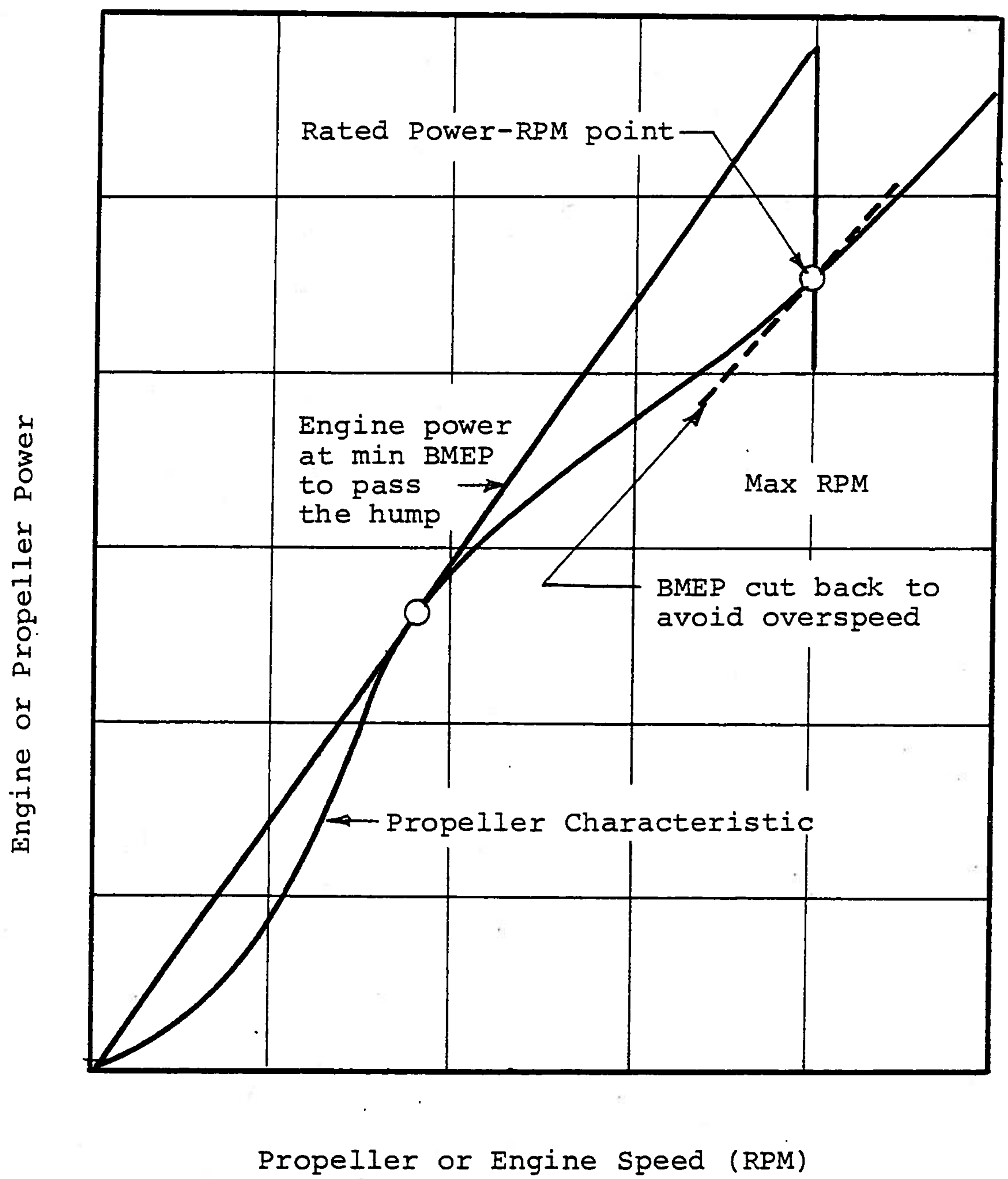


FIGURE 6.5 The Matching Process Complicated by a Planing-Boat Propeller Curve "Hump"

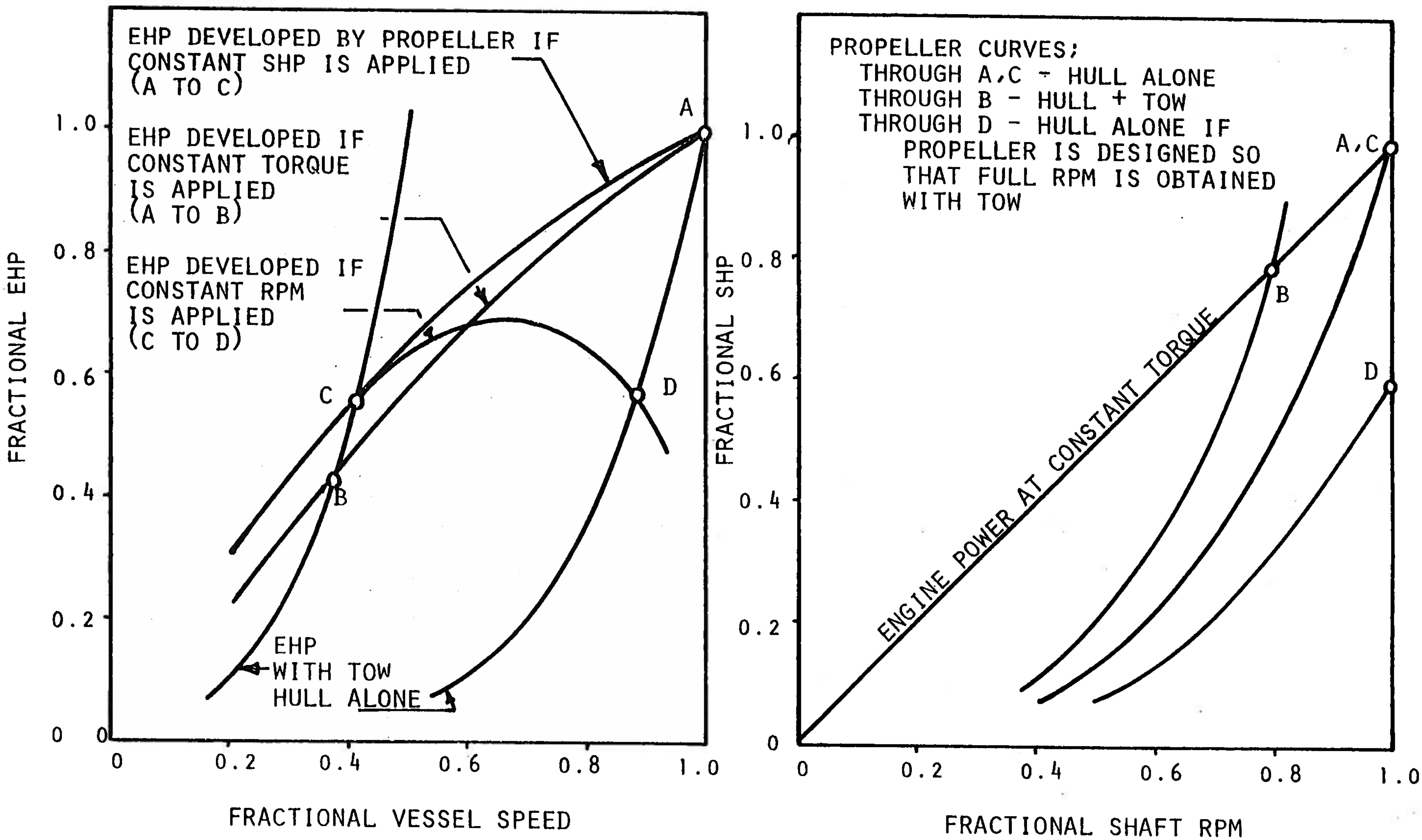


FIGURE 6.6 Vessel with Tow: EHP vs Speed and SHP vs RPM
(Adapted from Vibrans, Reference 2)

However, there are several ways of keeping engine RPM, and hence power, constant. The line A-C on the left sketch (a point in the right sketch) in Figure 6.6 is a constant-SHP contour, and indicates by its intersection with the propeller curve that EHP and vessel speed are significantly higher than otherwise.

The several ways of maintaining constant engine power and RPM in the face of such major changes in resistance are

1. A variable reduction gear ratio. A shift is made to a higher ratio under towing conditions.

2. A controllable-pitch propeller. Pitch is reduced so that RPM stays the same.

3. DC electric drive. In this drive system, speed of the propeller can vary independently of the engine speed.

Another type of remedy, suitable for a vessel specifically designed for towing, is to design the propeller for the towing condition, i.e. select its pitch ratio so that the engine turns at rated RPM under towing conditions. In Figure 6.6 this implies that point C, rather than point A, is the design point on the EHP-speed plane. There is a disadvantage, however, when the vessel is running free. Under this condition, the engine would tend to overspeed because of the light load, and thus the throttle must be cut back to maintain rated RPM. The line C-D on the left sketch is a constant-RPM contour, and by cutting the free-running propeller curve at D shows how much less power and speed must be than with the free-running design (point A).

It is an interesting exercise to construct the contours in the left sketch of Figure 6.6 (lines A-B, A-C, C-D). It illustrates how the propeller characteristics and the engine characteristics are used together to compute the response of the engine-propeller propulsion machine to changes in external conditions. The process is essentially a matter of manipulating the definitions of torque coefficient, thrust coefficient, and advance coefficient for the propeller, aided by definitions of EHP and SHP, and by the assumption that wake fraction and thrust deduction are not affected by the change in resistance. First consider the constant-torque contour; it is found from these relations that

$$\frac{J}{J_O} = \frac{V}{V_O}\,\frac{N_O}{N} \qquad (6.1)$$

$$\frac{K_Q}{K_{Q0}} = \frac{{N_O}^2}{N^2} \qquad (6.2)$$

$$\frac{K_T}{K_{T_0}} = \frac{T}{T_0} \quad \frac{N_0{}^2}{N^2} \qquad (6.3)$$

$$K_Q = K_Q(J) \qquad (6.4)$$

$$K_T = K_T(J) \qquad (6.5)$$

$$\frac{EHP}{EHP_0} = \frac{T\ V}{T_0\ V_0} \qquad (6.6)$$

The first three of these are the relations for advance coefficient (J), torque coefficient (K_Q), and thrust coefficient (K_T), formed into ratios. The zero subscript implies the design point (i.e. point A); its absence implies some general point. Note that torque terms have cancelled out of the K_Q ratio, as a result of constant-torque stipulation, consistent with constant BMEP. The next two denote the graphical relations of K_Q and K_T to J for the particular propeller. (An empirical equation can be used for each of these relations if preferred to graphical work. A straight-line approximation is usually adequate.) The last is a ratio formed from the definition of EHP.

There are here six equations with seven unknowns (J, K_Q, K_T, EHP, V, N, T). Fix a value for any one of the unknowns, and a solution is possible for any or all of the others. By choosing a set of V, you define the corresponding set of EHP, for example, and thereby obtain the contour on the EHP-V plane, as required. My favorite technique is to start with J, i.e. I choose a $J \neq J_0$, then proceed as follows:

1. Read K_T and K_Q from the curves thereof.
2. Solve for N/N_0 by equation (6.2).
3. Solve for V/V_0 by equation (6.1).
4. Solve for T/T_0 by equation (6.3).
5. Solve for EHP/EHP_0 by equation (6.6).

Repetition produces enough points to define EHP/EHP_0 versus V/V_0.

If the contour of constant SHP is wanted, a preliminary step is to multiply numerator and denominator of the K_Q formula by N, giving

$$K_Q = \frac{Q\ N}{\pi D^5 N^3} \propto \frac{SHP}{N^3} \qquad (6.7)$$

Then in place of equation (6.2)

$$\frac{K_Q}{K_{Q_0}} = \frac{SHP\ \ N_0{}^3}{SHP_0\ N^3} = \frac{N_0{}^3}{N^3} \qquad (6.8)$$

Let's use the case of SHP kept constant by constant BMEP and constant RPM, via the agency of a controllable-pitch propeller. If so, $N/N_0 = 1$ also. Including the modification to equation (6.2), the previous equations become

$$\frac{J}{J_0} = \frac{V}{V_0} \qquad (6.9)$$

$$\frac{K_Q}{K_{Q_0}} = 1 \qquad (6.10)$$

$$\frac{K_T}{K_{T_0}} = \frac{T}{T_0} \qquad (6.11)$$

$$K_Q = K_Q(J,P)$$

(P symbolizes pitch) (6.12)

$$K_T = K_T(J,P) \qquad (6.13)$$

$$\frac{EHP}{EHP_0} = \frac{T\ V}{T_0 V_0} \qquad (6.14)$$

To plot a point on the constant-SHP contour, choose a $J \neq J_0$. This plus equation (6.10) fixes a point on K_Q - J plane, and from this you can read the pitch ratio by interpolating among the K_Q curves drawn for various pitch ratios. The same pitch ratio must be used on the K_T - J plane, so that this pitch and the J chosen fix a point there. Once a value of K_T is in hand, it should be easy to deduce from a glance at equations

(6.9 - 6.14) how to compute EHP and V for this point.

6.6 AUXILIARY LOADS

An engine may drive more than one load. Or you could say that it always drives more than one load, because there are always attached pumps (lube oil, cooling water, fuel, etc.) that are essential to engine operation. But their demands should always be subtracted from engine output before the output data is published, so that the vessel designer has no occasion to concern himself with these things. On the other hand, the designer may himself add something; most engines can be had with power take-offs for just this purpose. An attached generator or bilge pump is common, and hydraulic pumps to drive deck gear are frequently found on fishing vessels. Sometimes the power needed for these devices is trivial in comparison to propulsion power, and can be neglected. But it also may be significant, and for example, it may be necessary to predict how propeller RPM changes when the auxiliary load is cut in and out. Recall the load-driver principles discussed in Chapter 2, and the advice there that all load-driver analyses be reduced to a single load-driver pair. Here is a situation in which this process is followed. An example illustrates.

A 100-hp engine drives the propeller of a small fishing vessel. A hydraulic system powers certain fishing gear; its pump is belted to the forward end of the engine. The problem is to find the RPM and power available to propulsion when the pump clutch is engaged.

Figure 6.7, bottom sketch, shows pump head-flow characteristics at several pump RPM (assumed to be same as propeller RPM) and the head-flow characteristic of the piping. Pump power is calculated at each intersection (Power = GPM xPSI/ 1714 x efficiency) and is plotted in the top sketch as a power-RPM characteristic. This characteristic is added to the propeller characteristic to produce a total load characteristic, which then shows the operating point by its intersection with the engine curve. The engine produces 90 HP at 450 propeller RPM, instead of the 100 HP at 500 RPM with the pump declutched. The power is divided 72 HP to propulsion, 18 HP to the pump.

Note the seeming paradox here: 18 HP is subtracted from the original 100 HP used for propulsion, leaving 72 HP available.

6.7 THE ENGINE AND THE CONTROLLABLE PITCH PROPELLER

If the propeller pitch can be changed in operation, it gives the vessel operator the freedom to pick the RPM inde-

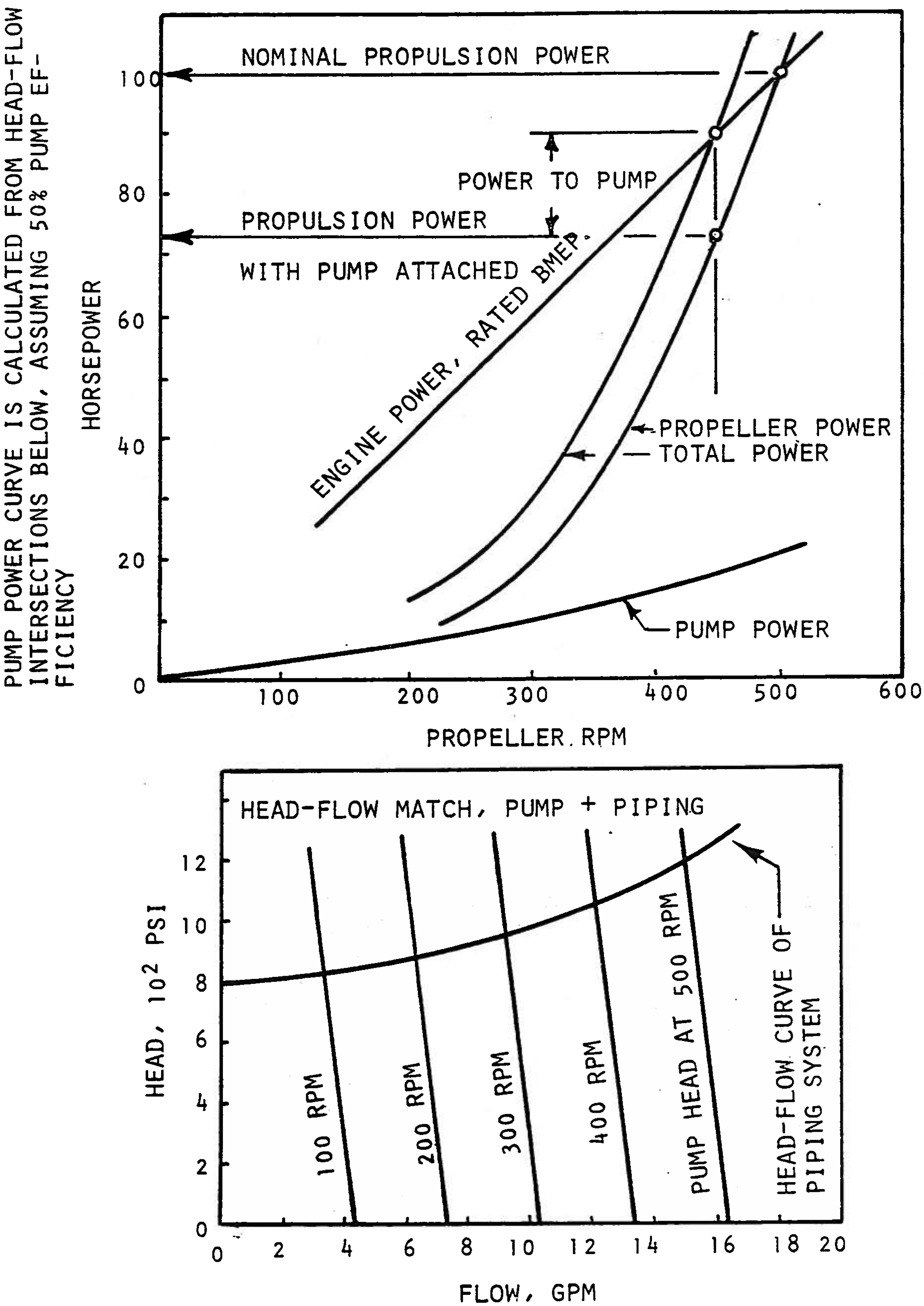


FIGURE 6.7 An Example of a Combination of Loads on the Propulsion Engine

pendently of power and speed. This ability can have several advantages. It obviates the problem of making allowances for deteriorations in service, as discussed earlier in this chapter. If major changes in resistance coefficient occur, as when a large tow is taken on, then a pitch change allows the engine to continue developing its rated power, also as discussed earlier. By reversal of pitch, it allows thrust to be reversed without reversing the engine. It permits fine adjustment of vessel speed in the low-speed range without a reduction of RPM below the idle speed of the engine. For vessels that operate for significant periods of time at low speeds, it allows the choice of pitch-BMEP combinations that produce the best possible efficiency at these low speeds. And if a constant-speed auxiliary load is to be driven (e.g. an AC generator), it allows the constant RPM of this device to be maintained as vessel speed is varied via pitch change. These, then, are the common advantages to be had from use of the controllable-pitch propeller, as related to the characteristics of the propulsion engine.

The first consideration is the difference in the matching process when the CPP is used. It has been noted that when the engine is diesel or SI, an allowance should usually be made to compensate for the loss in RPM that may occur as a result of the various service deteriorations. But compare Figure 6.4 with Figure 2.9 or 2.10. With a CPP, any of the curves in the latter two can be had by pitch change; by reducing pitch, the tendency of the original propeller curve to move to the left due to deteriorations can be cancelled by the move-to-the-right of the pitch reduction. Thus the margin is not required. If a "hump" problem is present at the beginning of the planing region, this complication to the matching problem may also be overcome by a reduction in pitch to allow the engine to develop full power in passing the hump, followed by an increase in pitch to prevent overspeeding as rated boat speed is reached.

The second important consideration is the relation of pitch control to fuel control. At first glance it would seem reasonable to give the operator freedom to set both as he pleases. This is indeed sometimes done, but it is usually better to restrict his freedom, for there are good chances for the operator to use it to make operating conditions worse than with the fixed pitch propeller, rather than better. For example, it has been mentioned that an advantage of the CPP is that it allows the best possible fuel rate to be chosen at part-load points of operation. Figure 6.8 illustrates this possibility. But it also allows the worst possible to be chosen. The operator could easily do so by always obtaining his part-load speeds via pitch reduction, keeping the engine at rated RPM. The operating points shown in Figure 6.8 would then lie along a vertical line (contour of constant RPM) through the rated point, and a glance at the figure shows that the fuel rates would be worse than with a

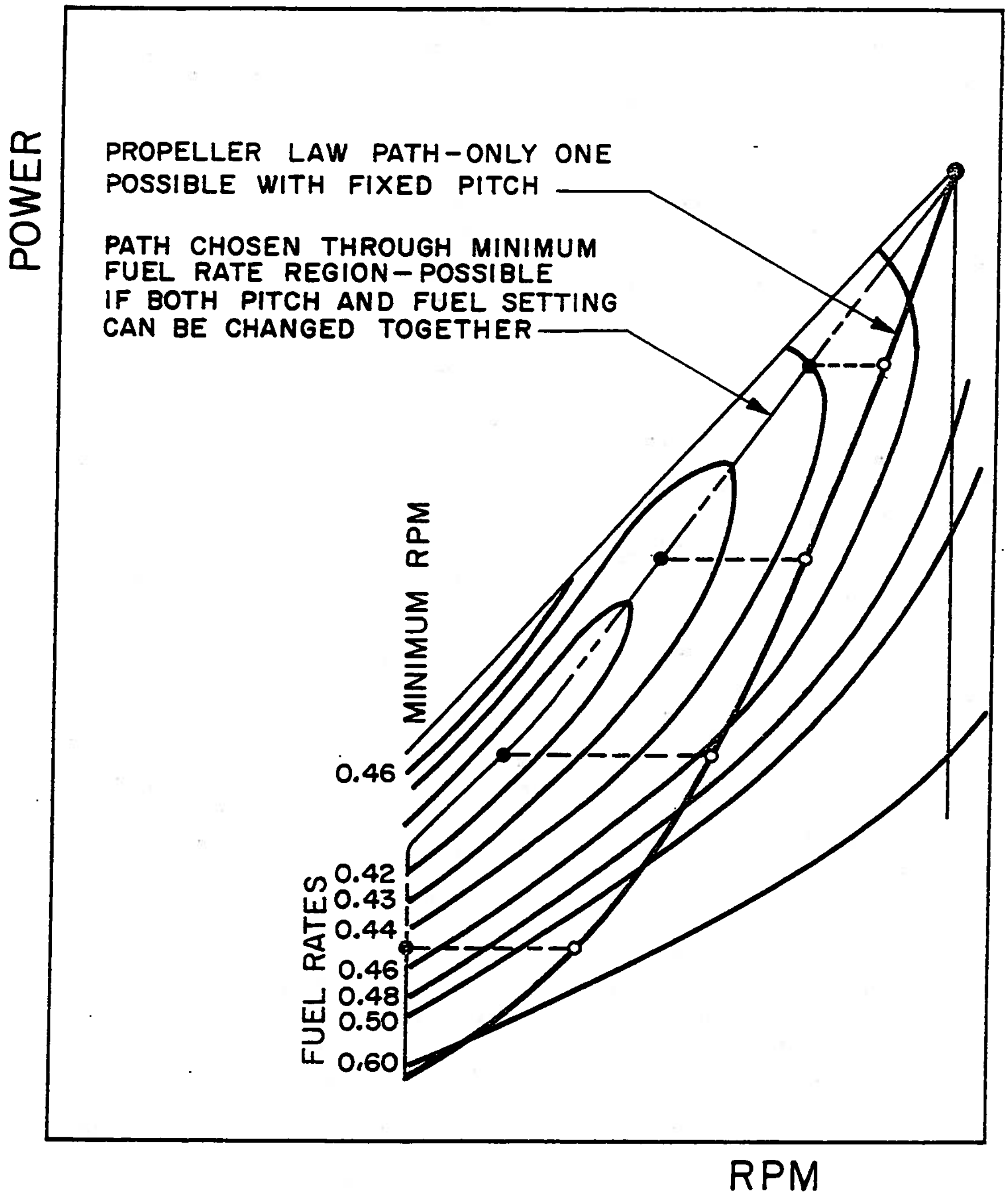


FIGURE 6.8 Part-Load Paths on the Engine Power-RPM Plane: Fixed-Pitch Path and Minimum Fuel Path with CPP

fixed-pitch propeller, rather than better. Another possible operator fault would be to set pitch too high, then attempt to reach rated RPM; unless a overload-protection feature were built into the engine governor, he could easily overload the engine.

It is therefore usually preferred to link fuel control and pitch control, giving the vessel operator a single control handle to manipulate. This handle will be marked off in increments of power, speed, thrust, or whatever may be appropriate for the particular vessel, and will be passed through a complete range, from full ahead to full astern. It typically sets control pressures in a pneumatic system that in turn establishes speed settings on a pitch governor and a fuel governor. The result is the pitch-fuel combination determined by the designer to be appropriate.

The designer must therefore establish a program--a specified relationship between fuel and pitch settings--which he or a specialist in control technology will then design into the control hardware. Two other control features must be considered also. One is the behavior of the system when conditions are not those upon which the design was based, and the other is the control at low speeds, speeds that with a fixed-pitch propeller would occur below the idling RPM of the engine.

Before exploring the three features of a CPP combined fuel-pitch control system, let's seek familiarity with how the features might be physically accomplished. A number of schemes have been used, some of them patented. The Escher Wyss/Greiner system is used as the example here (named after the two Swiss companies that developed it), and I have chosen it for discussion because it is described in an excellent paper [3] which can serve as further reading on engine-CPP interactions, if you seek such. The system is sketched in Figure 6.9, and is seen to be conceptually simple. Note that pitch setting and fuel setting are each controlled by a governor, this being a device that nominally maintains constant the speed of the machine it controls. Position of the control handle commands each to maintain a certain RPM. Now look at the top sketch in figure. The pitch povernor is <u>isochronous</u>, meaning that it manipulates pitch to maintain RPM precisely (well, almost) constant, no matter what the load (i.e. power being transmitted) is. In contrast, the fuel governor has <u>droop</u>, meaning that under increasing load it allows the RPM to decrease slightly from its setting. As evident in the figure, any given command from the control console contains a slightly higher RPM order for this governor, and this then tends to overspeed the system beyond the desires of the pitch governor. But as speed tends to increase, so does the power required to maintain it, and so occurs the droop until the two governor characteristics cross. There the system is in equilibrium--RPM is consistent with the

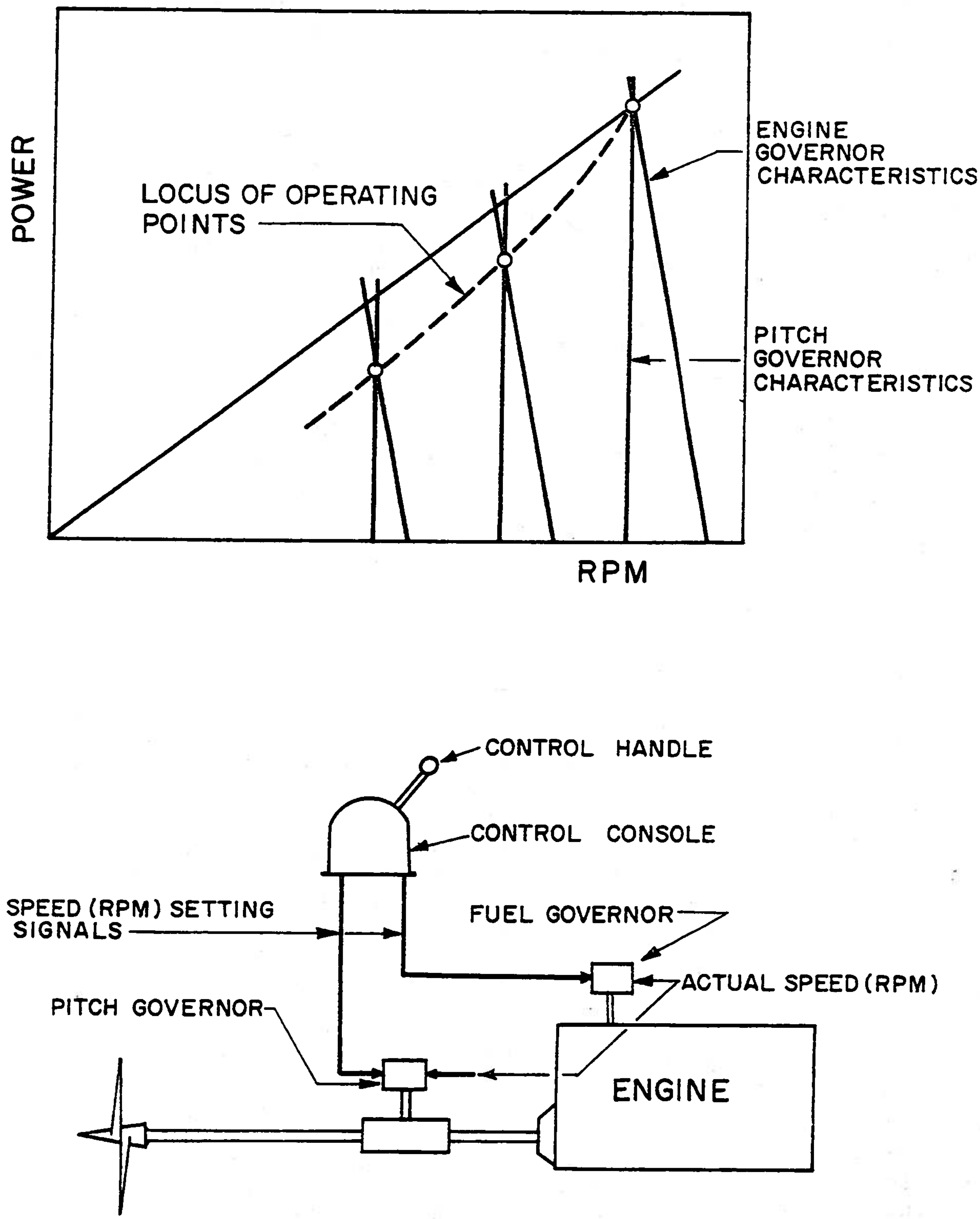


FIGURE 6.9 CPP - Engine Control Program: Some Features of the Escher-Wyss/Greiner System

desires of both governors. If something occurs to upset this equilibrium, such as worsening weather, taking a tow, or any of the resistance-change factors discussed earlier, the droop process continues, tending to drop RPM, but the isochronous pitch governor responds by reducing pitch, allowing RPM to increase back to the intended RPM equilibrium. The result is a constant SHP response, i.e. contour A-C in Figure 6.6. Propeller efficiency is not likely to be as high as at the design condition, but at least the engine is allowed to continue developing rated power.

Three features have now been mentioned for a CPP-engine control system: (1) a "program" defining a path through the part-load refion, (2) a response to off-design conditions, and (3) a low-speed scheme. These are elucidated below.

The program path should be based on some criterion defining a best result. Figure 6.8 suggests one criterion; note that points are drawn to define a path that follows the "ridgeline" of lowest fuel rates. Corresponding points (i.e. at same powers) are drawn along a cubic propeller curve to indicate how much fuel rate is improved at the several arbitrarily chosen points (the amount obviously depends on the shape of the propeller curve). Figure 6.10 is a similar plot for a propeller, again with the propeller-law cubic curves being used. There the path of highest possible propeller efficiencies is followed. If the two figures were drawn to the same scale on the same sheet of paper, you would see that the two paths don't quite coincide, and probably wouldn't, no matter what the shape of the propeller curves, save by coincidence. But a little compromise, effected perhaps by trial and error juggling, could produce a choice of points that would be best by some overall criterion, say, least fuel consumption per mile. Once a choice of points sufficient to define the path is made, the subsequent laying out of the program is a simple matter. Figure 6.11 shows the construction of the program for an example set of points placed at increments of 0.2 fractional power. The figure has the engine characteristics and propeller characteristics superimposed on the same axes, although the fuel rate and efficiency contours of the preceding figures are omitted to avoid too gruesome a confusion of wandering lines. The program appears in tabular form in the lower part of the figure, and is a summary of the power, RPM, pitch, BMEP, and speed coordinates of the points, read directly from the graph. It now serves as a specification for the design of the control hardware.

The response to off-design conditions is usually as described in the Escher Wyss/Greiner example discussed above, i.e. pitch automatically changes to keep RPM constant, so that engine power is not changed. If, in the design condition, the program point has been selected by some "best"

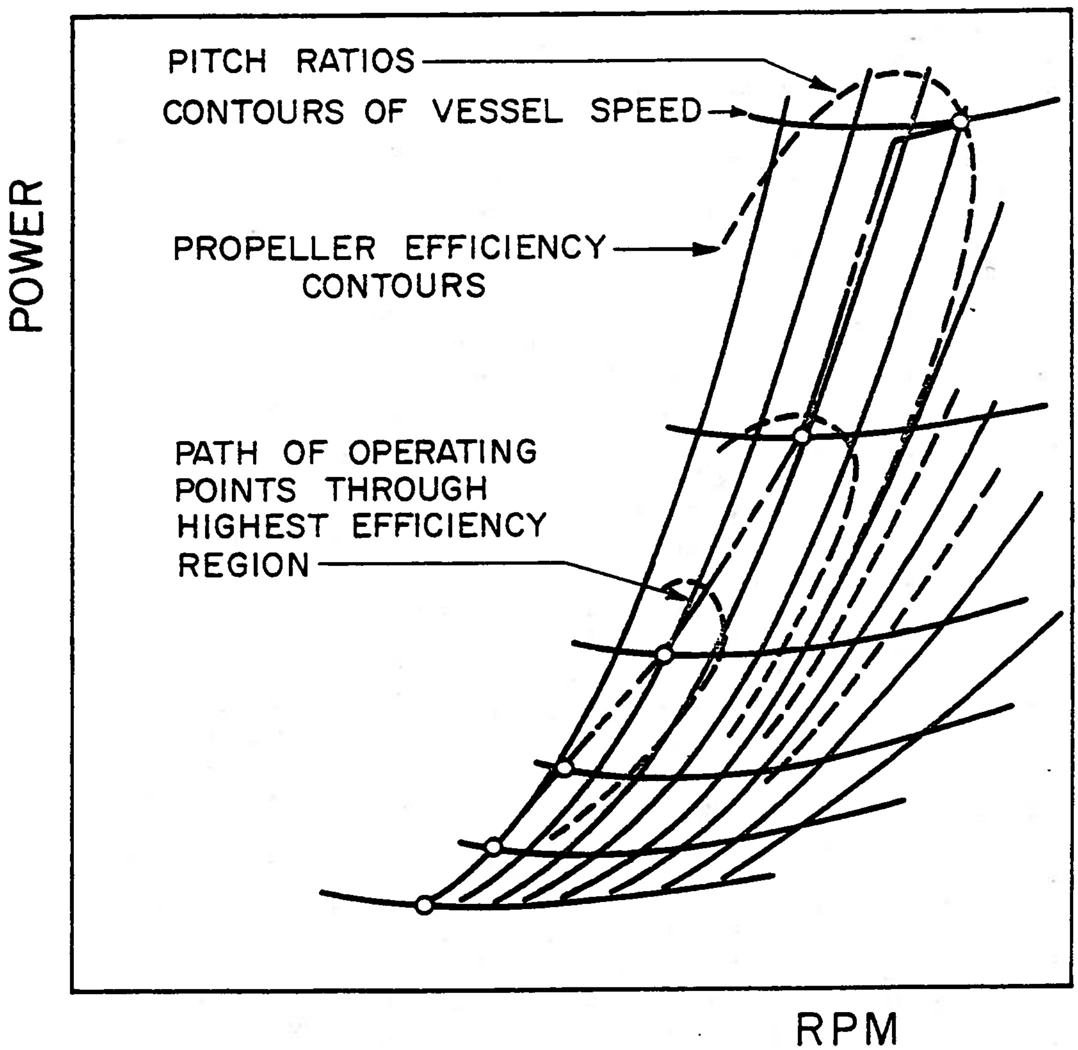
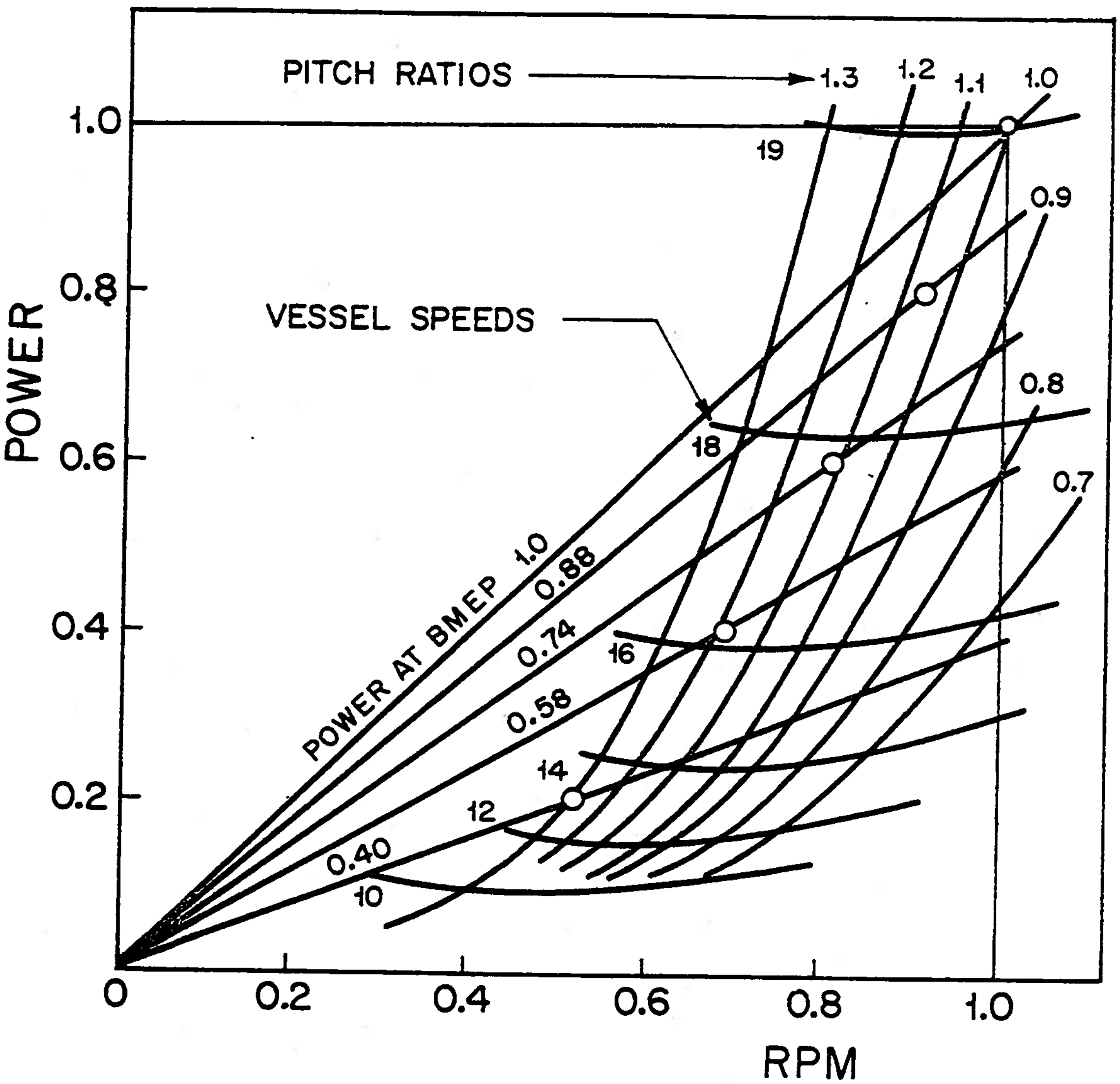


FIGURE 6.10 Best Efficiency Path for Part Loads with CPP



POWER	RPM	PITCH	BMEP	SPEED
1.00	1.00	1.00	1.00	19.0
0.80	0.91	1.05	0.88	18.5
0.60	0.81	1.11	0.74	17.7
0.40	0.68	1.19	0.58	16.2
0.20	0.52	1.30	0.40	13.0

FIGURE 6.11 Example of CPP - Engine Control Program, with Points Shown at Equal Increments of Power

criterion, doubtless it is no longer best after the change, but at least engine power is maintained at the desired level. The benefit of this has been discussed earlier, and can be reviewed by looking back at Figure 6.6. If the engine is a gas turbine, however, the load response feature might be omitted because of the constant-power characteristic of the turbine; pitch could be ordered directly without intervention of a governor, thus simplifying the control system. On the other hand, these remarks might not apply in the case of a vessel designed to take a heavy tow, since "constant-power" is a good approximation to actual gas turbine behavior only for a limited range near rated power.

Special conditions prevail at what might be called "creeping" speeds of the vessel. For one thing, there is the practical difficulty of accomplishing a desired power-speed-pitch-RPM-BMEP combination that is an essential feature of the program defining operating conditions over most of the power range. This difficulty is analogous to the one that faces the draftsman trying to draw the propeller characteristics in the lower left corner of a figure such as Figure 6.10 or 6.11. At the scale necessary to fit the whole view on the page, he finds it tedious and exacting to fit the separate curves in accurately, even if he has accurate data to work from. As you see, he gave up. Likewise, it is not practicable for the control components to act with the precision that would be required to reproduce specified operating conditions in this low-power corner. The other thing is that it scarcely matters. So little fuel is usually consumed during the brief maneuvering periods which this low range represents, that there is no incentive to strive for efficiency. The most reasonable solution is to uncouple the normal fuel-pitch link (i.e. the "program") and let the operator set the pitch directly. Thus, in a typical system, movement of the control handle into a range near the neutral position between ahead and astern does the following: (1) it sets a minimum engine RPM into the engine governor, which is thereafter maintained as long as the control handle is in this low range, and (2) it bypasses the pitch governor (if used) with a signal directly to the actuator for a pitch appropriate to handly position. Shaft RPM is constant, vessel speed and direction is controlled directly by varying pitch, and power is maintained at whatever level necessary by the engine governor.

6.8 REFERENCES

1. J. A. Smit, "Considerations of Propeller Layout from the Enginebuilder's Point of View," Shipbuilding and Shipping Record, May 10, 1968

2. Frank Vibrans, "Variations in Marine Engine Performance with Departures from Design," ASME Oil and Gas Power Proceedings, 1956

3. F. Schanz, "The Controllable Pitch Propeller as an Integral Part of the Ship's Propulsion System," Transactions SNAME, 1967

CHAPTER 7

MARINE ENGINE RATING

7.1 THE PROBLEM

The word rating can be applied to any engine performance parameter, such as power, torque, RPM, BMEP, exhaust temperature, etc, but most often is applied to the power; "it's a 200 horsepower engine" would be a common way of expressing a rating. Such a statement is fraught with ambiguity, however simple it may seem at a glance, for a rating may be a maximum rating, or an advertised rating, or a gross rating, or a net rating, or a service rating, or a commercial rating, or an intermittent rating, a pleasure boat rating, and doubtless some more. The confusion is somewhat due to non-standard terminology (different people use different terms to mean the same thing), but is also caused by two fundamentally different types of rating; one is the power (or torque, RPM, etc) which the engine is capable of producing under some specified ideal condition, and the other is the power which it should be allowed to produce under service conditions. These might be the same, but usually they are not. A further complication is that internal combustion engines are sensitive to ambient atmospheric conditions, so that rated values have to be adjusted if operating conditions are to be much different from design conditions.

7.2 TERMINOLOGY

Bare or gross horse power is the maximum brake horsepower produced on a test stand under standard atmospheric conditions [1], without such accessories as raw water pump, intake silencer or filter, alternator, etc, that are likely to be found on an engine in service. This term is most often used for spark-ignition engines, and unfortunately is also sometimes the advertised horsepower. The advertised power is the power boastfully proclaimed in magazine or brochure handed out at a boat show, and you should agree that if it is the gross horsepower, the whole truth is not being told. The net horsepower is likewise the brake horsepower obtained under test stand conditions, but with all accessories that would be used in service in place.

Maximum horse power appears to be self-explanatory, but may mean different things for different engine types. For the spark-ignition engine, it can be taken to be the same as the net horsepower. For a diesel, let me borrow a definition from a prominent builder, "maximum rating is the horsepower capability of the engine that can be demonstrated within five percent at the factory under standard conditions."

This will probably not be the maximum in the sense of being the very most that the engine can produce, but is more likely the most power that it will be allowed to produce. For example, the diesel exhaust smoke will become objectionable at some output, and this may be selected as the maximum level. Different manufacturers may have different ideas about this, however. Standards of the Diesel Engine Manufacturers' Association might be expected to be more firm in definition than I am here, but their modern edition of marine engine standards had not been published as this was written. For a gas turbine, maximum power is that brake horsepower developed at a maximum allowable combustion temperature. For both the diesel and gas turbine, maximum and advertised horsepowers are often the same.

Service power is that power which the vessel designer intends should be developed in service. It may be the same as the maximum power, but often it is not. The reason for not is that the life of an engine is dependent on level (BMEP for reciprocating engines, combustion temperature for gas turbines) at which it is operated. If operated at the maximum level, its life, or time between major overhauls, will be too short for many users. Since this turns into a fairly complex topic, it is discussed in a separate section to follow. The other terms that are mentioned in the first paragraph (continuous, intermittent, etc), and not defined to this point, are part of the service rating concept and are also discussed later as part of that concept.

7.3 CORRECTIONS FOR AMBIENT CONDITIONS

Assume that the maximum rating of the engine is established, most likely by accepting the manufacturer's stated maximum power. This rating should be accompanied by a notation of the ambient atmospheric conditions at which it applies. Usually these will be the standard conditions of the Society of Automotive Engineers [1], which are 29.38 inches mercury, 85F (standard atmosphere at 500-ft elevation). Another common standard corresponds to sea level conditions, and is 29.92 inches mercury, 60F. If the engine is of European manufacture, its standard conditions may be those of CIMAC (Congres International des Machines a Combusion), namely 736 mm mercury, 20C, and 60% relative humidity (in English units, pressure and temperature are 29.0 inches and 68F). CIMAC also has had another set of standards for ship engines, namely 760 mm (29.92 inches), 30C (86F), and 60% relative humidity. In short, there are several different sets of standard atmospheric conditions in use.

Two things are of concern here. The first is to put all engines on a common basis when you are comparing outputs or ratings of competing engines. There can be a significant difference. For example, there is about a 4% difference between the two American standards, i.e. an engine rated at 100 hp at 29.92 inches, 60F, should be rated about 96 hp at 29.38 inches, 85F. The second is to correct the ratings of the engine to conditions actually expected in service. If an engine is to operate in the tropics, or even if it is to take its air from within a hot machinery space, its rating should be reduced correspondingly or it will, in effect, be running overloaded according to what the rating should be for the conditions at hand.

So how to adjust the rating? The simplest answer is to advise using the engine builder's rating correction curves. Such curves have been shown in Chapter 5 for the gas turbine, and it is safe to assume that they will always be furnished with a gas turbine because of the high sensitivity of this engine to atmospheric temperature. They are also often furnished for diesel or spark-ignition engines, but may not always be available. There are standard formulas that can be used in such a case. The simplest one in common use is

$$BHP_o = BHP_s \frac{P_o}{P_s}\sqrt{\frac{T_s}{T_o}} \tag{7.1}$$

where P is pressure, T is temperature, s subscripts implies standard conditions, and o subscript implies observed or actual conditions. Although this formula is widely used, it is really more of a rule-of-thumb than an accurate formula. (The 4% difference between standard conditions, as quoted in the preceding paragraph, was found by this formula.)

CIMAC is attempting to promulgate more accurate correction formulas, and of course, they are more complicated. The standard form is

$$\frac{P_x}{P_r} = k - 0.7\,(1-k)\left(\frac{1}{\eta} - 1\right)$$

$$k = \frac{P_{ix}}{P_{ir}} = \left(\frac{p_x - a\,\phi_x\,p_{sx}}{p_r - a\,\phi_r\,p_{sr}}\right)^m \left(\frac{T_r}{T_x}\right)^n \tag{7.2}$$

where:

- P = Brake power
- P_i = Indicated power
- k = Ratio of indicated powers
- p = Atmospheric pressure
- p_s = Saturation vapuur pressure
- ϕ = Relative humidity
- T = Absolute air temperature
- η = Mechanical efficiency

The coefficient and exponents are given by the following table:

	a	m	n
non-turbocharged spark ignition and Diesel engines whose output is limited by the excess air (formula CIMAC A)	1	1	0.75
non-turbocharged Diesel engines whose output is limited for thermal reasons (Formula CIMAC T)	0	1	1
non-intercooled turbocharged diesel engines	0	0.7	1.5

The values for turbocharged engines with intercoolers had not been determined as this was written.

A complication with intercooled diesel engines is their sensitivity to temperature of the water used in the intercooler (or aftercooler, as it is also called). The colder this water, the denser will be the air leaving it and entering the engine cylinders, and hence the higher can be the rating of the engine. If seawater is used in the cooler, the engine will thus be sensitive to seawater temperature. At present, there being no general formula available for corrections, reliance must be placed on engine builder's correction data. However, an alternative scheme of cooling is often used, this being to use engine jacket water as the cooling medium. This water is, of course, always much hotter than sea water, so that engine rating must be less with this method, but since jacket water temperature is kept nearly constant by a thermostat, the sensitivity to sea water temperature is essentially absent.

7.4 SERVICE RATING

Now assume that two steps have been carried out: the maximum rating has been determined, and then corrected (if necessary) to the ambient conditions under which the engine will operate. The third, last, and toughest step is to decide the service rating of the engine will be. In terms of Figure 6.4, this means deciding what the rated horsepower labelled in the figure will be, i.e. at what power the engine will run during the intended service condition of the vessel. It might be the corrected maximum power in some cases, but often not, for reasons discussed in Section 7.2. The situation is further illustrated by Figure 7.1, which is a plot of expected life between overhauls vs. a parameter proportional to power level at which the engine is generally run, all for a prominent small-craft marine diesel [2]. The lighter the loading, the longer should be the life, and vice versa. The spark-ignition and gas turbine engines experience similar life-load relationships, though I have not seen curves such as Figure 7.1 published.

There is in this situation an appeal to the designer to plunge into an economic trade-off study. If he uses the maximum rating as the service rating, the boat will have the highest possible speed with a particular engine, and its earning power or other benefit will be a maximum. If he reduces the rating, some of this will be lost, but economic benefit will follow from longer engine life. In actuality, a designer is not likely to indulge in a fancy analysis to pick the rating because the data he would need is largely speculative in most instances. Even if a curve such as that of Figure 7.1 could be accepted as quantitatively accurate for the particular engine being analyzed, the hours of running at different loads, the cost of overhaul or replacement, and the maintenance care that the engine is to receive from its operators all lie in the future, and so are not subject to accurate determination. It is only in the rare situation where careful records have been kept on a similar vessel in similar service that the designer can feel justified and able to set down some dollar-vs-dollar comparisons to choose the engine rating. Nonetheless, he should always keep in mind the concept: the lower the rating, the longer the engine life (the engine might rust away while doing nothing, but that is another problem).

What, then, does the designer do in the absence of data upon which to base a truly rational choice? In the case of diesel engines, at least, the builders publish ratings as a guide to application. Below, for example, are two such sets of ratings plucked at random from application literature that was lying close at hand as I wrote this:

Cummins NT-335-M

Maximum BHP	335 @ 2100 RPM
Maximum SHP	315 @ 2100 RPM
Pleasure boat rating, SHP	315 @ 2100 RPM
Work boat rating, SHP	221 @ 1800 RPM

(The difference between BHP and SHP is reduction gear loss.)

Caterpillar D330C

Maximum BHP	200 @ 2200 RPM
Intermittent BHP	165 @ 2200 RPM
Continuous BHP	125 @ 2000 RPM

Caterpillar's definition of maximum is the one quoted in Section 7.1. To quote further from the same company: "Intermittent is the horsepower and speed capability of the

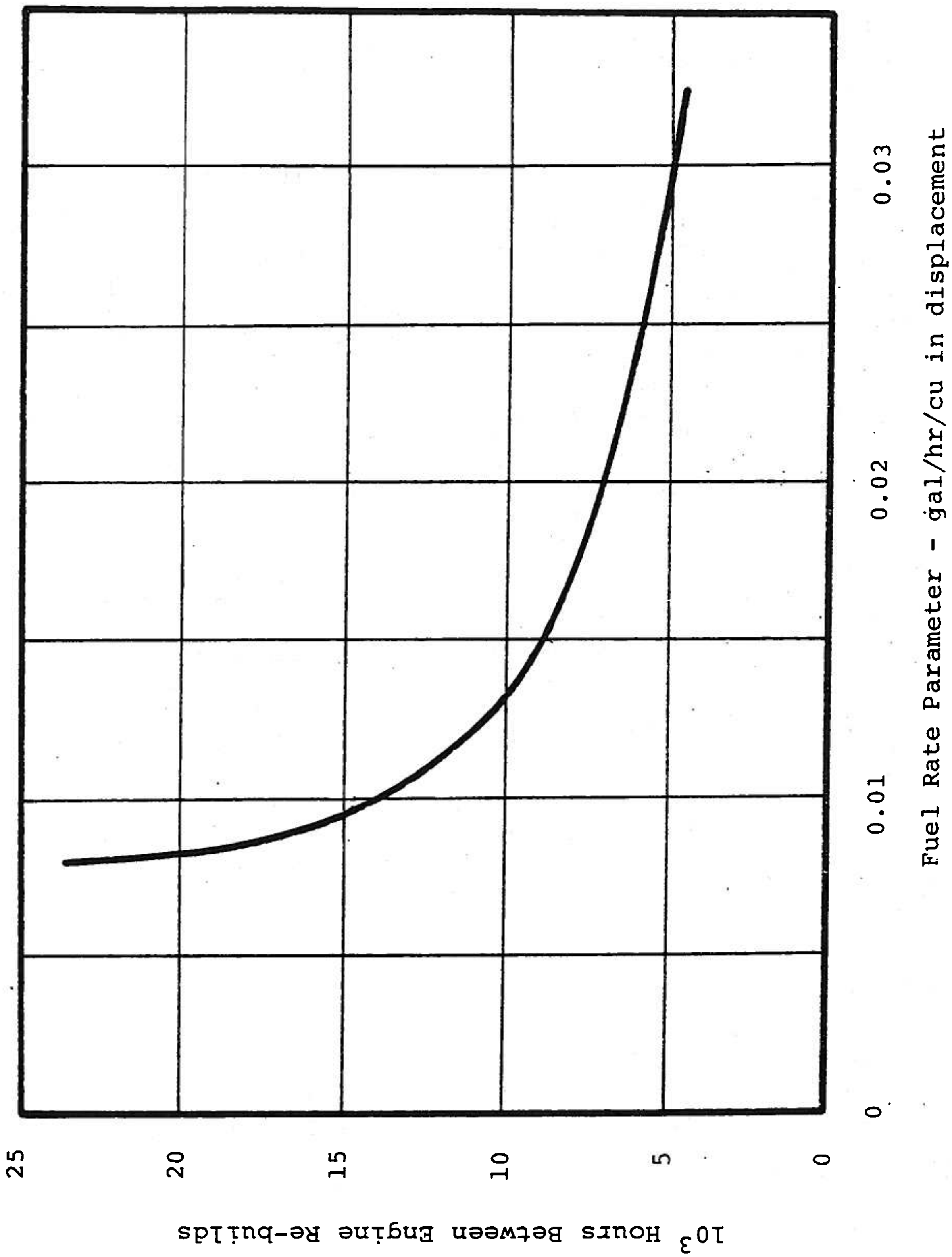


FIGURE 7.1 Life of Marine Diesel Engines as a Function of Rated Load

engine which can be utilized for about one hour, followed by about an hour of operation at or below the continuous rating. Continuous is the horsepower and speed capability of the engine which can be utilized without interruption or load cycling." Although this is one company's definition it is typical. Yours is the choice as to which description fits your design, and of course you may use the definitions as starting points from which to interpolate if none of the three fit closely. Cummins is more explicit in the case cited in describing the type of service in mind for the ratings quoted, but the idea is the same: a pleasure boat is likely to be used only intermittently at its top speed, whereas a work boat (which can mean many different things, to be sure) typically runs continuously at the design speed. But such labels cannot be accepted blindly; some pleasure boats are indeed used for intermittent helling around, but others may be low-speed long distance cruisers (and diesel is more likely in this type) that will operate continuously at rating, quite in the fashion of a "work boat." So there is no simple answer. The basic and important requirement in any case is, however, that the designer understand the meaning of rating, and how it must relate to the intended service of the vessel. Beyond that, his judgement, experience, and the hopefully expert advice of the engine builder should put him close to a good choice.

Much the same message can be repeated for the gas turbine. Glancing back at Figure 5.3, you will note that the gas turbine builder also gives several ratings which are to be fitted to the type of service expected.

And the message might be repeated substantially the same for the spark-ignition engine also. However, a cloud of misunderstanding befogs the issue with this engine. One element of misunderstanding arises from the different traditions of advertising between diesel and spark-ignition people. Although not universally the case, one often finds that the diesel ratings advertised are those suitable for the service, while the spark-ignition blurbs give the gross ratings. As seen from the definition of gross horsepower that has been given, it just cannot be the service rating, whatever the vessel designer intends, and a lesser power is actually developed in service. This contributes to the notion that a diesel horsepower is somehow "worth more" than a gasoline horsepower.

Further difficulty comes from the automotive origin of the engine. In a passenger automobile, the engine is rarely fully loaded (your foot is not all the way to the floor even at highest speed). It is somewhat in the situation of a marine engine driving a propeller pitched too low to load the engine fully before the maximum BMEP is reached. Thus, although an automobile engine may otherwise suffer from frequent cold running in city traffic, it is seldom, perhaps almost never actually loaded continuously to its maximum rating. Now, if the engine in its marine version is matched to its propeller at that maximum rating (or the net rating as defined earlier), it may in fact be used for lengthy running periods at that power. The owner, accustomed to having an automobile engine serve for many years without needing overhaul, may be amazed to see how quickly his boat engine shows signs of distress (probably the exhaust valves will be the first part to suffer [3]). Since the advertised power of the engine cannot be used for long continuous running, here is more evidence that a diesel horsepower is "worth more."

In truth, a rated horsepower is indeed worth more than a gross horsepower, or a maximum horsepower, when engines are run continuously. With the spark-ignition engine, just as with the diesel, one must set a service rating, probably less than the maximum rating, appropriate to the service that the engine will be subjected to. Unfortunately, the SI engine builders are typically unhelpful in advancing this concept. As noted earlier, some state only the gross horsepower. Others give (to quote one example) "useful power," which is net SHP, and hence probably the maximum SHP. How do you tell what is being stated? Sometimes reading the fine print tells it. Sometimes you may have to inquire diligently, and if a crisp and confident answer is not forthcoming, assume the worst, i.e. gross BHP. Whatever the situation, the designer should find out the maximum net power at the reduction gear output coupling, corrected if necessary for non-standard atmospheric conditions. From here his judgement and imagination must guide. If it's a roaring-around boat whose major appeal is a top speed that no one is likely to try to sustain for lengthy periods, then he might well set the service rating equal to the corrected maximum rating. If, however, it is a boat that is to be run steadily at more than a few hundred hours a year, then he might set the service rating at something like 70% of the maximum. Please note, though, that this is a speculative figure, unsupported by data.

One last item on the "worth more" question between diesel and spark-ignition horsepower. It is sometimes said that diesels are inherently more rugged, and consequently can have a service rating set closer to the maximum. This may be so, but I believe it to be more a case of individual engine design. In both types, engines are designed with different degrees of ruggedness (e.g. different bearing surface areas). The belief arises from the circumstance that diesels are more commonly used in heavy-duty services, while spark-ignition engines are more commonly light-duty automobile engines. Therefore, if one compares the typical diesel with the typical spark-ignition engine, he may indeed find support for diesels as a class being built for heavier service.

7.5 FURTHER POINTS ON RATING

Rating for naval service should be mentioned. The Navy has usually required that a propulsion engine be capable of continuous operation at 10% above rated power, and required that this ability be demonstrated by test run [4]. In terms of Figure 6.4, I interpret this to mean that there should be at least 10% power between the <u>design</u> power and the <u>rated</u> power as labelled in the figure, i.e. the engine would be rated at what is called the <u>design point</u> there. Except for this slight switch in terminology, all of the previous discussion should apply to naval work.

Something should also be mentioned concerning RPM in the process of selecting a service rating. An engine builder will always specify an RPM at which the advertised power is developed; you may recall from the chapter on engine-propeller matching that an engine must reach rated RPM in order to reach rated power. There is less opportunity for ambiguity with this parameter, since it is not necessarily affected by the things that make the difference between gross and net power, nor need corrections be made for atmospheric conditions. Generally then, the advertised RPM can be accepted for use even though the accompanying power figure is a gross power. Usually, a reduction in RPM is taken if the service power rating is selected for continuous service. This is evident in the sample rating figures from Cummins and Caterpillar quoted in the preceding section, and they can be taken as guides to what is typical. Looking back once again to Figure 6.4, note that an RPM margin is shown. It is not intended as an additional margin to the one just mentioned, but as a small margin to allow for uncertainties in design, manufacture, and instrumentation when the maximum rating is used as the service rating.

7.6 REFERENCES

1. SAE Standard J816a "Engine Test Code for Non-Turbocharged Spark-Ignition and Diesel Engines," <u>SAE Handbook</u>, Society of Automotive Engineers.

2. Michael Wagner, "Rational Methods for Engine Rating," Great Lakes and Great Rivers Section, Society of Naval Architects and Marine Engineers, October meeting, 1970.

3. John W. Hurst, "Genesis of an Inboard Engine," <u>Yachting</u> Magazine, August and September, 1968.

4. C. F. Perry, "Engine Selection and Rating for Military Ship Application," Society of Automotive Engineers Paper 670953, 1967.

CHAPTER 8

PROPULSION PLANT CONTROL

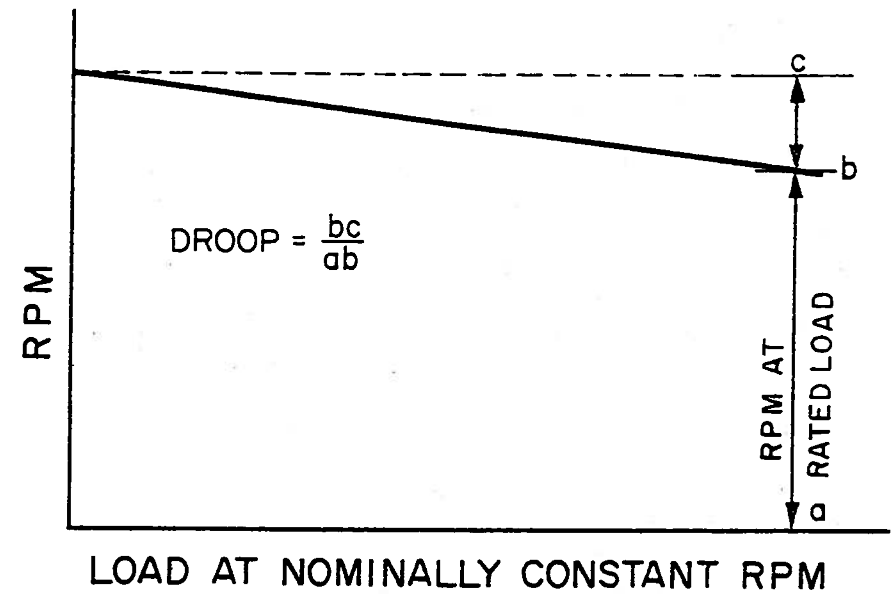
8.1 INTRODUCTION

The propulsion engine must be started and stopped, its speed must be changed at the will of the operator, and its speed must be maintained constant between ordered changes. Some of this is quite elementary; the speed of the common spark-ignition engine, for example, is usually controlled by a simple throttle valve actuated by a mechanical cable from the hand of the operator, and the steady speed of all the engines needs supervision only for safety purposes because of the stable load-driver relationship (recall Chapter 6). Complexities there be, however. Governors are used on marine engines, especially diesel and gas turbine, and they warrant considerable discussion. Means of reversing are universally required; a reverse train in the reduction gear is widely used for this purpose in small craft, and the appropriate clutches must be controlled in conjunction with the engine. Shaft brakes are sometimes found, their usual purpose being to aid rapid reversal, and their control must also be integrated with that of other components. Much the same can be said when the controllable pitch propeller is used; control of its pitch is usually integrated with engine fuel control, as pointed out in Chapter 6. In fact, it is this integration of engine control with that of other parts of the propulsion system that is the chief concern of the marine engineer or vessel designer. Although this, too, can be simple, it sometimes involves fairly complex hardware, and in any case, there are several principles to be observed. Integration of component controls, in fact, is the major topic of this chapter.

But first, some words about governing.

8.2 GOVERNING

A <u>governor</u> is a device that automatically maintains speed (usually a rotational speed) constant in the face of load changes. If it is an <u>isochronous</u> governor, it does this job with negligible steady-state deviation from the set speed, no matter what the load on the machine it is controlling, i.e. it has no <u>droop</u>. Droop is the property of allowing a small decrease in speed as the load increases; the definition is illustrated by Figure 8.1. The same figure also illustrates one of the principal virtues of droop, which at first thought might seem like a defect in a device intended to maintain speed constant. The lower sketch shows how it establishes load sharing when two engines share the same load. Most cases



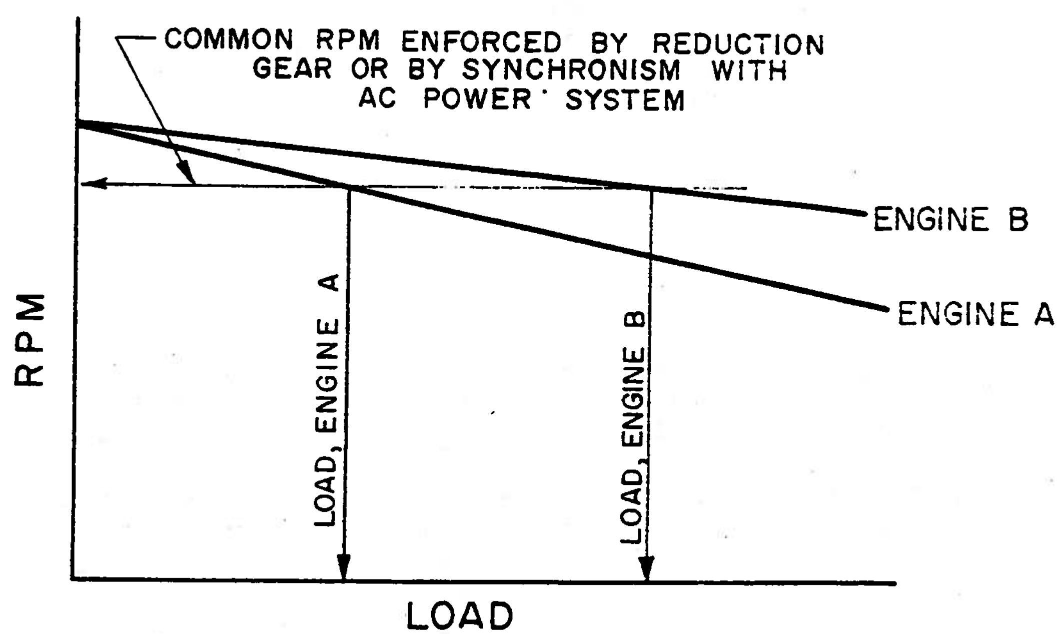


FIGURE 8.1 Governor Droop and Its Effect on Load Sharing between Two Engines

of load sharing require by their very nature that the RPM be the same for all drivers, and of course, the total load must be the sum of contributions by all individual drivers. For a given total load, then, there is only one RPM and only one combination of loads where equilibrium is possible, as the sketch should make clear. The case shown is actually one of very poor load sharing, since the individual loads are so radically different; it is usually desireable to have equal loads assigned to each driver. The figure is drawn as it is merely to emphasize how droop does fix the share of each.

The term "governor" is also applied to a device which only limits a speed, i.e. prevents it from going below or above a desired value. This is to be contrasted to an overspeed trip, a device that will stop an engine by cutting off its fuel or air in the event of a serious overspeed.

None of the small-craft marine engines requires a governor to maintain shaft speed in steady-running conditions, for the propeller-engine combination is strongly stable, but governors are nonetheless nearly always found on the diesel and gas turbine engines.

The diesel requires at least a speed-limiting governor for protection. Often a small-craft marine diesel will be equipped with a governor that limits both the upper and lower (to prevent stalling) speeds, but has no control over speed in between; speed in the operating range is controlled by manual adjustment of fuel rack position. These engines also often have a quick-closing valve in the air intake to accomplish positive shutdown, either manually or by overspeed trip.

The marine diesel also is found with a governor that maintains set speed throughout the operating range, as well as performing the limiting function. It must, of course, have an adjustable speed setting so that any RPM within the operating range can be selected. The use of such a governor, in conjunction with a governor controlling propeller pitch, has been introduced in the discussion of controllable-pitch control programs in Chapter 6. It can also perform other valuable functions. For example, it may have a load-limiting feature, which by cutting back on the allowed amount of fuel at low speeds, prevents the BMEP from being excessive under low-speed conditions. When two or more engines are geared to the same shaft, their governors establish equal load sharing, as explained in the discussion of governor droop. With twin-screw propulsion plants, a synchronizing feature is sometimes used to prevent audible beats between engines.

The gas turbine also requires governing to prevent over speeding, and indeed requires at least this protection for each of its two turbine shafts. The use of a speed-controlling governor on the compressor turbine shaft with several protective features incorporated, plus the use of a topping governor on the power turbine shaft, has been discussed in Chapter 5.

A governor is rarely found on a marine spark-ignition engine, since overspeed protection is not thought to be necessary with this type. With this need absent, other possible incentives are usually not strong enough to justify the expense of adding a governor. The reason for doing without overspeed protection is that the torque curve falls so steeply above rated RPM that the engine is essentially self-regulating against dangerous over speeds.

Some explanation of how the typical governor works will help in understanding their use. It consists of three basic elements, (1) a speed-sensing device, (2) an amplifier to increase the small forces generated by the speed sensor to usable magnitude, and (3) a servomotor to exert the force or torque on the engine fuel-control device, pitch actuator, etc.

There are several ways in which engine speed can be sensed, classifiable as electrical, mechanical, or hydraulic. The traditional method (James Watt used it) is the mechanical flyball, and it is yet the most commonly used method. See Figure 8.2. The whole assembly rotates at a speed proportional to engine speed; inspection of the figure shows that centrifugal force in the flyballs produces couples about the pivot points that are balanced by couples produced by the spring force. Position of the base of the spring is a function of initial spring force, and of speed. For a given initial spring force, which is the operator's way of setting a desired speed, there is hence a unique position for every speed. The output force somewhat upsets this relation, especially if the force is used to overcome mechanical friction. Nonetheless, this force is sometimes used in simple governors to set the fuel directly. In this case, sensor, amplifier, and servomotor functions are combined into a single device. They are commonly called <u>mechanical</u> governors. Alternatively, the output force moves a hydraulic pilot valve (the amplifier) which in turn directs oil under pressure to an actuating cylinder (the servomotor), thus providing what is usually called a <u>hydraulic</u> governor. The sketches in Figure 8.3 show these elements clearly. The flyball output force is quite small compared to that needed by the mechanical governor, hence the balance between centrifugal couples and spring couples is much less distorted.

Three qualities should be expected of a governor, (1) force or torque output adequate to position the fuel (or other) setting, (2) adequate speed of response, and (3) stability (i.e. the engine should not "hunt" about the set RPM).

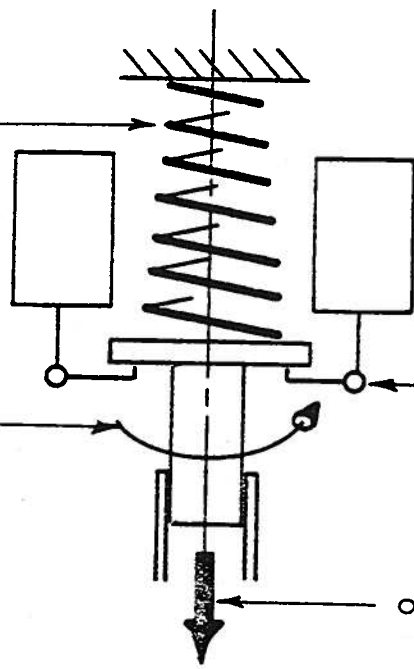


FIGURE 8.2 Elements of the Flyball Governor

These, especially the last two, involve engine characteristics as well as governor characteristics, and especially the dynamic characteristics of both. And understanding of the problems must be based on knowledge of at least the basics of automatic control theory. All of this is beyond the scope of this text; therefore I must beg off from giving a complete discussion of governing. Readers further interested might well start with a book by D. B. Welbourn [1], which is one of the few modern control theory textbooks that specifically treats governing of engines. The balance of this section discusses some of the governor characteristics in only a descriptive manner.

Look at Figure 8.3, which consists of three sketches showing the essentials of a hydraulic governor. The first shows the simplest device, but unfortunately one that is unstable. The instability is caused by the lack of any feature to prevent the engine speed from zooming right past the setting before the flyballs stop correcting for the speed error. The second sketch shows a more complicated arrangement in which the instability is eliminated by a lever connecting throttle and speeder spring. As the throttle opens (say) to increase speed, the lever moves to relieve the speeder spring, and to allow the pilot valve to move toward closure. The speed thus creeps in on the set value, stably. The consequence, however, is droop, since the set point (top of the speeder spring) is moved in the process. The resultant speed is less than the set value by some small amount that increases as load on the engine increases. There is therefore a load-speed characteristic that droops, as noted earlier. The third sketch shows a yet more complicated mechanism that is both stable and isochronous. Stability is obtained via transient droop. It can be seen that if the needle valve is closed, the assembly of transmitting piston, receiving piston, etc., must act as a solid lever, and the action is the same as in the second sketch. However, the needle valve does allow oil to leak out so that the centering spring can slowly restore the receiving piston to its original position, thus uncoupling the set point from the throttle position.

Figure 8.4 is a semi-pictorial exploded view of the actual parts of a governor of the type used on marine diesel engines.

8.3 INTEGRATED CONTROL OF ENGINE AND OTHER DRIVE-LINE COMPONENTS

If a reverse-reduction gear or a controllable-pitch propeller is used, the control of these devices must be coordinated with control of the engine. For example, the ahead and astern gear trains in the gear will be engaged and dis-

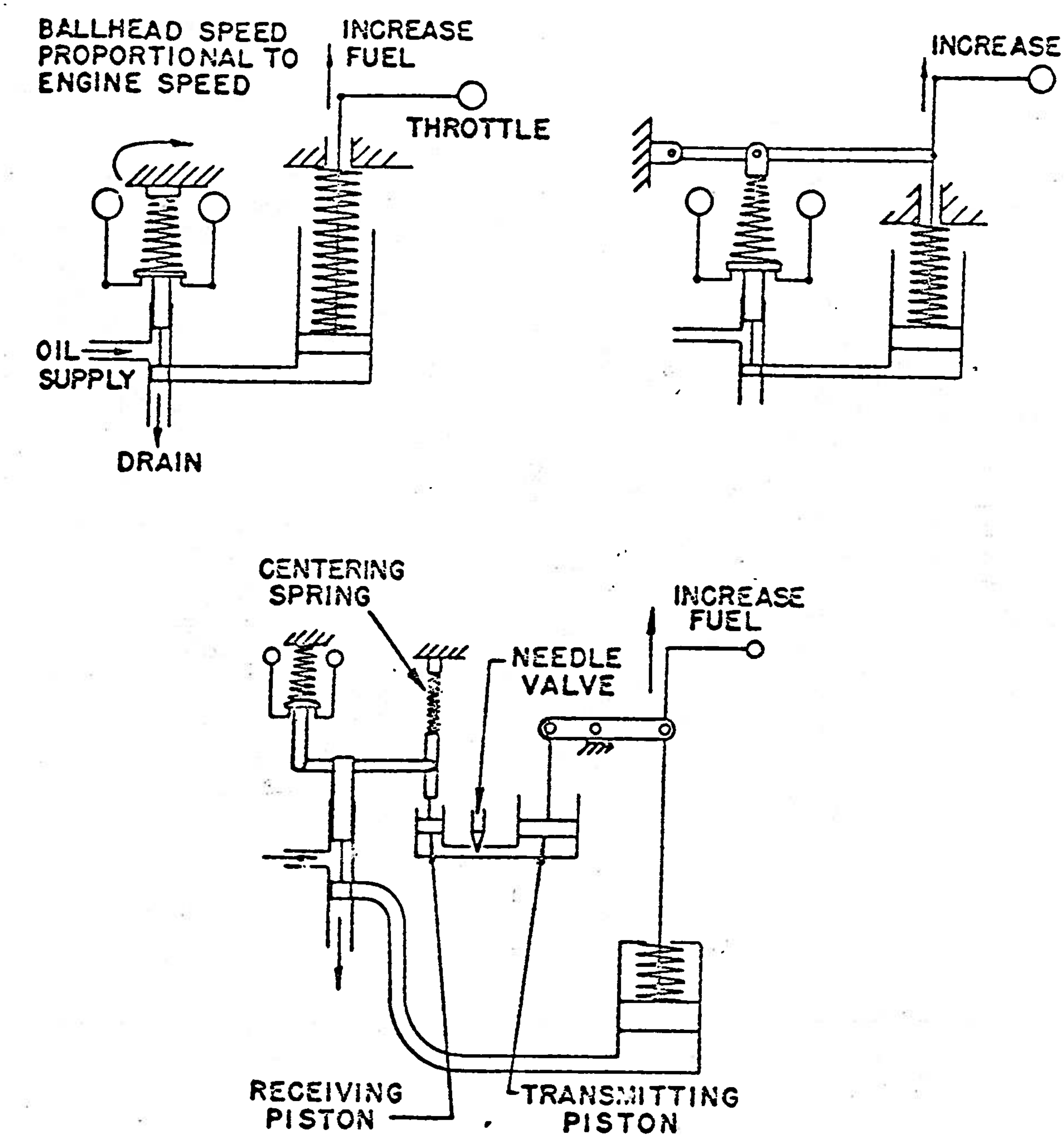
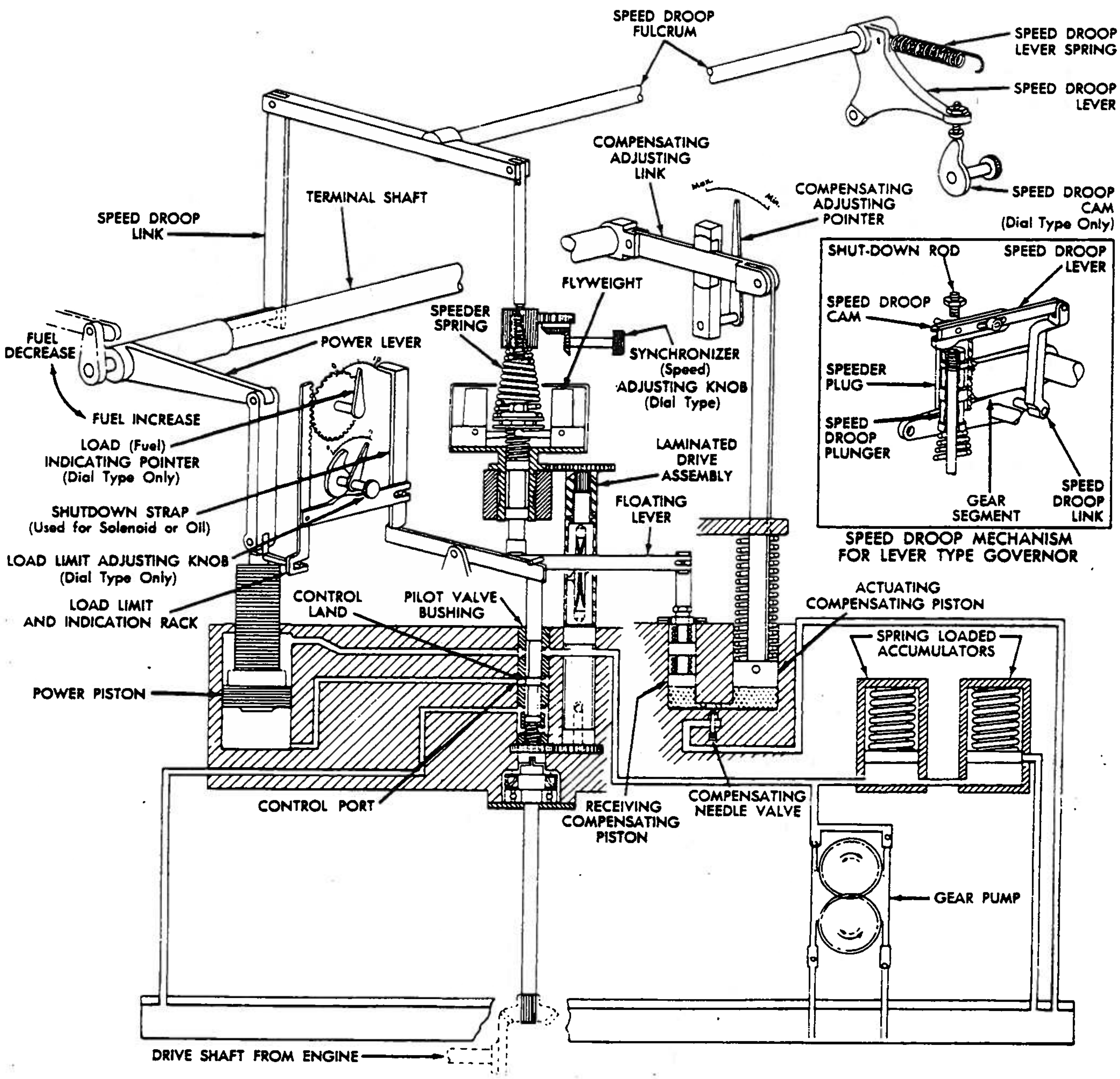


FIGURE 8.3 Elementary Governor Mechanisms (from Forrest Drake "The Governing of Internal Combustion Engines," ASME Oil and Gas Power Proceedings, 1951)



from Reference 1

FIGURE 8.4 Guts of a Hydraulic Governor for a Diesel Engine engaged by a clutch or pair of clutches. For protection of the engine, an engaged clutch must not be disengaged unless the fuel is cut back; for protection of the gears, ahead and astern clutches must not be engaged simultaneously; for protection of the clutches, engagement or disengagement must not occur under conditions that would produce high-speed slipping or slipping under heavy load. Similar precautions are necessary in maneuvering with the CPP: pitch must not be reduced without simultaneous cutback in fuel to the engine, and perhaps rate of pitch reversal must be limited to avoid stalling the engine during a crash reversal.

All of this can be left to the knowledge and skill of the operator, and quite successfully too if he indeed has knowledge and skill, is never careless, and never panics. That is, separate and independent controls for fuel and clutches (or pitch of CPP) can be provided, and the brain of the operator used as the coordinating element. So it is with that outboard motor that sometimes maneuvers my sailboat up to its mooring: a forward-neutral-reverse lever for one hand, and the throttle for the other; one hand pushes the lever from one direction to the other while the other simultaneously twists the throttle down to "shift" position and back up again. (And I'm all the while managing the tiller with my toes). Such elementary arrangements are common in the smallest craft because inertias are small, meaning that shafts and engines decelerate almost as rapidly as the operator can move the controls, and that there is little rotating kinetic energy to be absorbed by the clutches. As the size of vessel and power plant become larger, inertias of hull and machinery increase. In a rapid reversal, for example, it may easily be possible to have the shaft still spinning at high RPM ahead when the operator is ready to engage the astern clutch. Since the designer must assume the machinery will not always be handled with high skill, he usually provides an integrated control system for all components involved in a maneuver. The operator is provided with a single control handle with which he gives his commands; the control system carries these out as rapidly as is feasible while protecting the several components from damaging interactions. Thus the integrated control system.

For the CPP, it is not only maneuvering, but all operating conditions that are better served by integrated control of pitch and fuel. Recall that pitch-fuel programs are discussed in Chapter 6.

Figure 8.5 shows the most elementary system for an engine plus reverse-reduction gear. Engine throttle control and gear clutch control are each individually manipulated by push-pull cables from the operator's position. Figure 8.6 shows the two functions combined into an integrated, yet relatively simple, mechanical system. By the mechanical linkage between the single control handle and the individual cables, throttle movement is interlocked with clutch position so that idle RPM

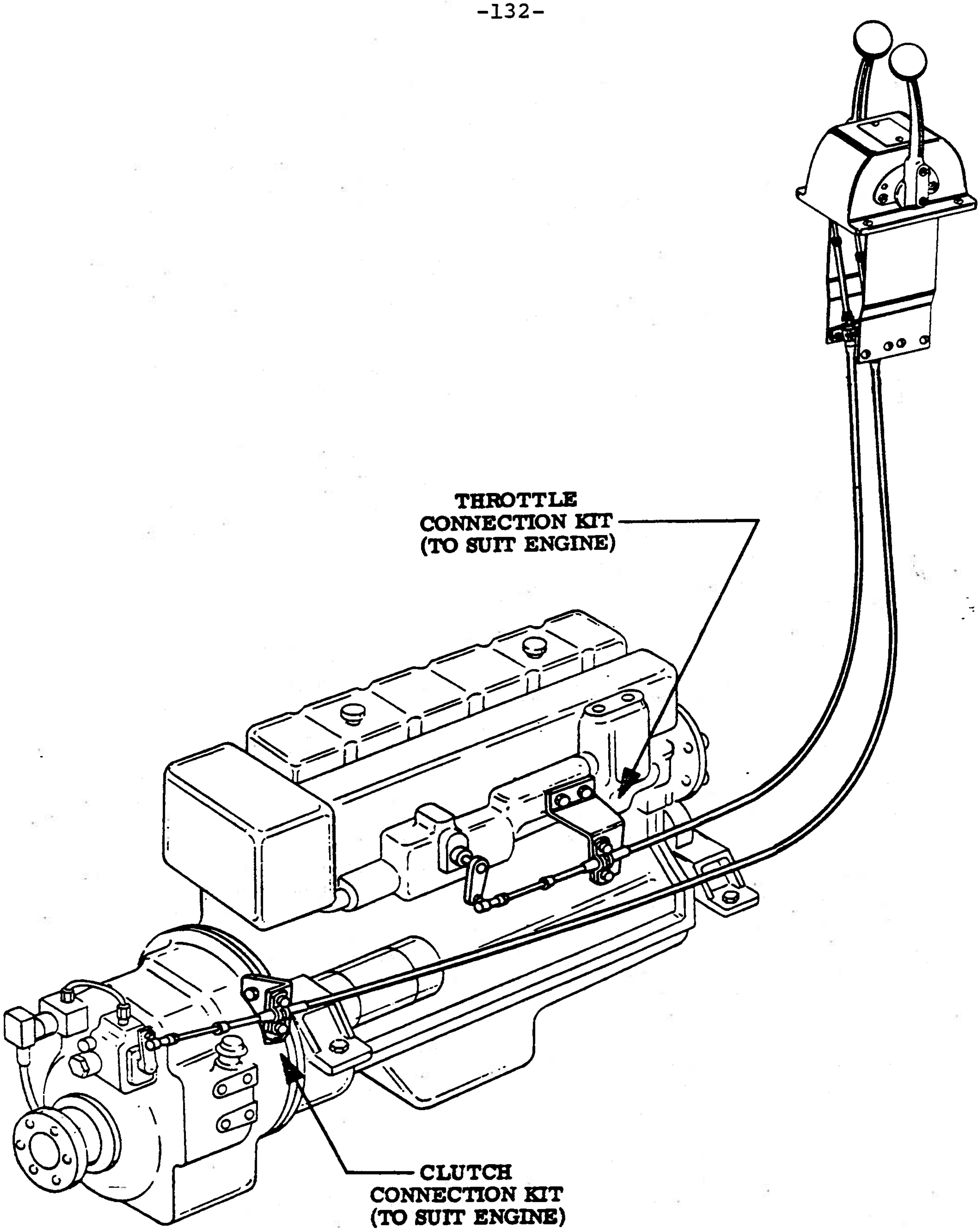


FIGURE 8.5 Independent Manual Control of Fuel and Clutch
(courtesy Caterpillar Tractor Co.)

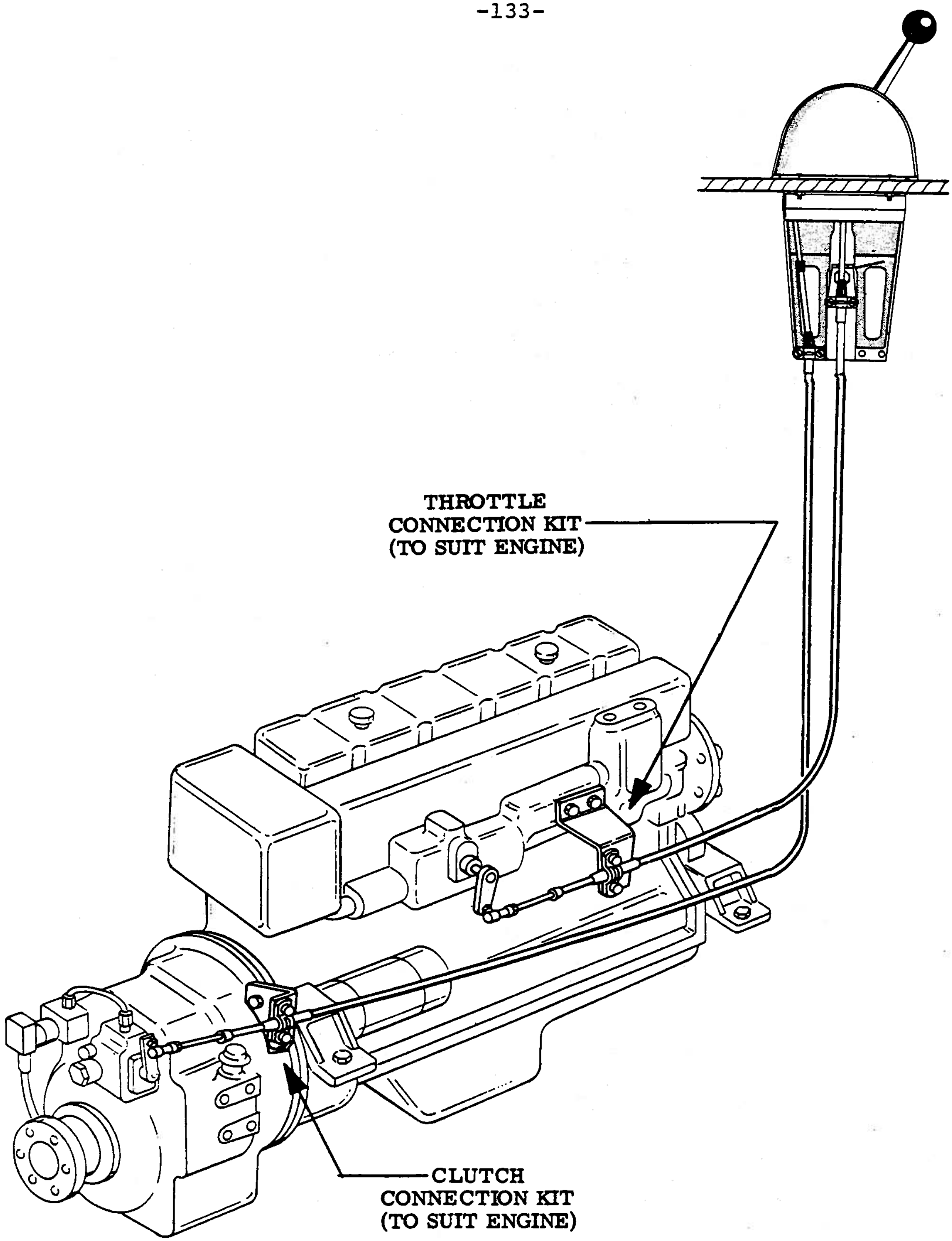


FIGURE 8.6 Integrated Manual Control of Fuel and Clutch
(courtesy Caterpillar Tractor Co.)

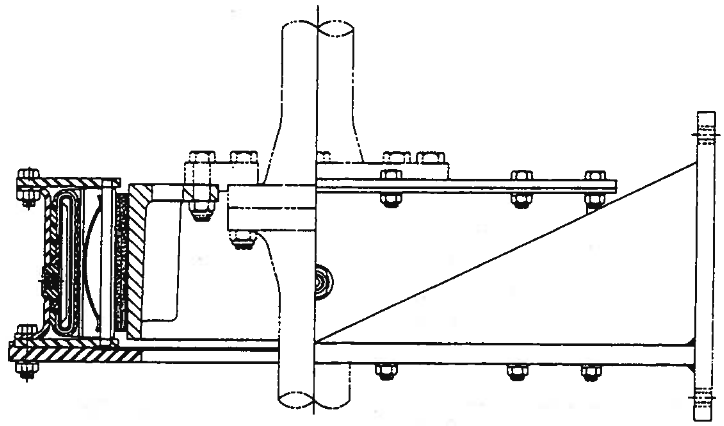
is not exceeded until either ahead or astern clutch is engaged.

The addition of a shaft brake adds a third element to control. While direct mechanical control is doubtless feasible with this addition, it is rarely seen. The reason is that brakes are most often found when the vessel and its machinery are large enough to make inertia effects significant, and thus require some degree of automatic timing of functions, as well as braking. Pneumatic control components are employed most typically to provide this feature, which is difficult to obtain with the simple mechanical system. Brakes are therefore usually associated with pneumatic control, and the brake itself is likely to be a pneumatic one. Figure 8.7 gives a diagramatic view of a pneumatic brake. It is actuated by inflation of a tire that fits between the rotating and stationary members. A better view perhaps is given by Figure 9.3, which shows the same device used as a clutch; the ahead and astern clutch tires are clearly visible and labelled (though the word "tire" does not appear).

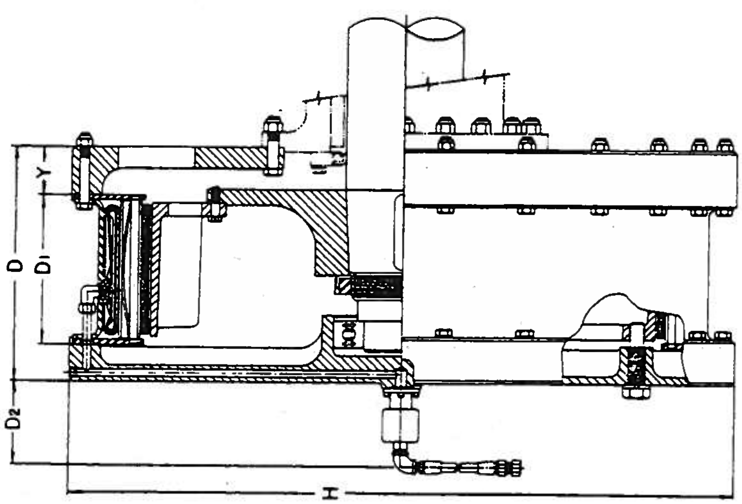
Figure 8.8 shows a proprietary (WABCO and Caterpillar) system with a combination of pneumatic and hydraulic components actuating engine fuel control, ahead and astern clutches, and shaft brake, all from a single control handle. Compressed air is supplied from an external system; the hydraulic line is supplied by a pump which is an integral part of the reduction gear. Note the interlocks among clutch, fuel, and brake actuators. The pneumatic speed signal to the fuel actuator is blocked by low clutch oil pressure, meaning that the engine must remain at idle RPM while the clutches are disengaged. The brake air supply is applied or vented by the clutch pressure, so that the brake is engaged only when the clutches are both disengaged.

The system of Figure 8.8 does not necessarily relieve the human operator of the need for intelligent action. Take, for example, the case of a rapid reversal from ahead to astern. As the astern clutch engages, and the brake drum releases, the propeller may still tend to windmill in the forward direction sufficiently to stall the engine. Or, in spite of the operator's attention, the brake may not empty fast enough to prevent its stalling the engine. The key to coordination of events so that these things do not happen is the proper selection of time constants within the system, the time constant being the time in which an event characteristically takes place*. It is generally a function of volumes (especially in a pneumatic system) and resistance to flow. Thus the diameter of pneumatic tubing, and its length, determine the time to respond of components such as pneumatic actuators. The same can be said for the hydraulic part of the system, but its time constants are usually much shorter because of the incompressibility of the fluid, so that the

*not a precise definition



The clutch here is shown applied as a shaft brake. Note that the shaft is attached to the inner member, and that the outer member is attached to a foundation for mounting on ship structure.

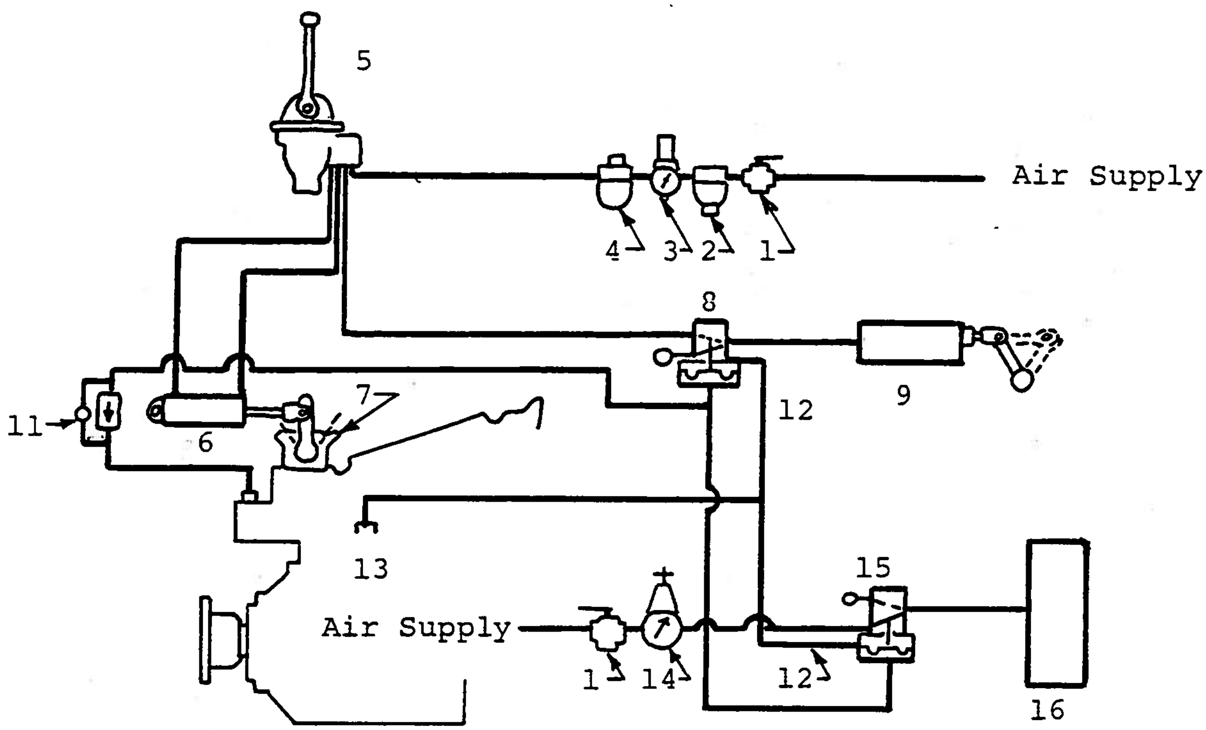


SIZES 38VC1200 THROUGH 66VC1600
(Capacities to 14,000 hp)

SIZE	D	D1	D2	H	Y	Rotorseal Size
38VC1200	19¾	13¾	7¹³⁄₃₂	49⅞	2	1" RH
42VC1200	19¾	13¾	7¹³⁄₃₂	54⅛	2	1" RH
46VC1200	20¼	13¾	7¹³⁄₃₂	60¾	2½	1" RH
52VC1200	20¾	14¼	8⅜	67½	2½	1¼" RH
51VC1600	24¼	18¼	8⅜	67½	3	1¼" RH
60VC1600	24¼	18¼	9⁵⁄₃₂	76½	3	1½" RH
66VC1600	24¼	18¼	9⁵⁄₃₂	83½	3	1½" RH

from Fawick application literature

FIGURE 8.7 Details of Pneumatic Clutches



1. Shut-off valve, supply air.
2. Air filter.
3. Pressure regulator—regulated pressure, 100 PSI.
4. Line lubricator.
5. Pilot house control, single lever combined speed and directional function. Regulated delivery pressure 0-65 PSI.
6. 3-Position reverse gear control cylinder.
7. Reverse gear clutch control valve.
8. Throttle cut-out valve. Controls air supply to speed actuator. Engages upon rise in clutch oil pressure.
9. Speed actautor.
10. Oil pressure line from clutch control.
11. Oil flow control, choke-check type valve.
12. Safety oil drain.
13. Oil drain return gear sump.
14. Pressure regulator. Pressure set to shaft brake manufacturer's recommendation.
15. Shaft brake cut-in, cut-out valve. Engages brake on drop in clutch oil pressure. Releases brake upon rise of clutch oil pressure. Details will vary with system requirements.
16. Shaft brake.

Adapted from Figure III 12.8 of <u>Cat Marine Application and Installation Guide</u>, Caterpillar Tractor Company

FIGURE 8.8 Pneumatic Control System for Engine-Clutch-Brake pneumatic time constants are likely to be the significant ones. Therefore the sizes of components and connecting lines are chosen according to the inertias involved in vessel and propulsion system. The choice is usually made empirically, based largely on the experience of the engine builder.

Figure 8.9 shows another proprietary control system featuring a shaft brake (Mathers Control, Inc., Seattle). The brake is the pneumatic clutch type discussed above. The key component is obviously the "air drive unit," which translates mechanical commands from the control stations into pneumatic signals to engine governor, clutch actuator, and shaft brake. The guts of this device consist of several valves actuated by the shaft to direct the air to the three components, and pneumatic timing devices (volumes and orifices) that establish the sequence of operation. For example, when the control handle is thrown from forward to reverse, the brake is engaged, the clutch put in neutral, and the engine cut to idle. The brake air leaks off so that the brake is released about the time the shaft comes to rest. As the brake is released, the clutch proceeds to catch up with the position ordered by the control handle. A speed boost is given the engine just before clutch engagement to prevent stalling. All timing is accomplished by the time constants of the pneumatic components, i.e., by the time taken to fill or empty, and they are built in to the system, rather than being set by automatic sensing of operating conditions. The designer must therefore have a good idea of the stopping time, and of the time constants of his components. These are usually established by experience, rather than by analysis.

Figure 8.10 gives an idea of response times to be expected with a pneumatic control system on an engine-clutch-brake combination, although it should be realized that the time magnitudes will be different for vessels and propulsion systems of different size. The figure is taken from Reference 2, and is for a harbor tug propulsion plant.

Pneumatic control systems can be more elaborate than those just discussed, although the basic features are likely to be no more than those mentioned to this point. For an example of elaboration, the vessel may need more than one control station, and it may be desireable to disconnect all but one station at a time, thus requiring additional interlocking. If two or more engines are geared to the same shaft, further interlocking is also required to prevent an off-line engine from being put on line when shaft RPM is above idling speed. Details of a system incorporating these complications are set forth in Reference 3. (This paper is also reproduced in Chapter 7 of Reference 4.)

Slipping clutches are sometimes used to obtain low propeller RPM for maneuvering. The clutches are made oversized

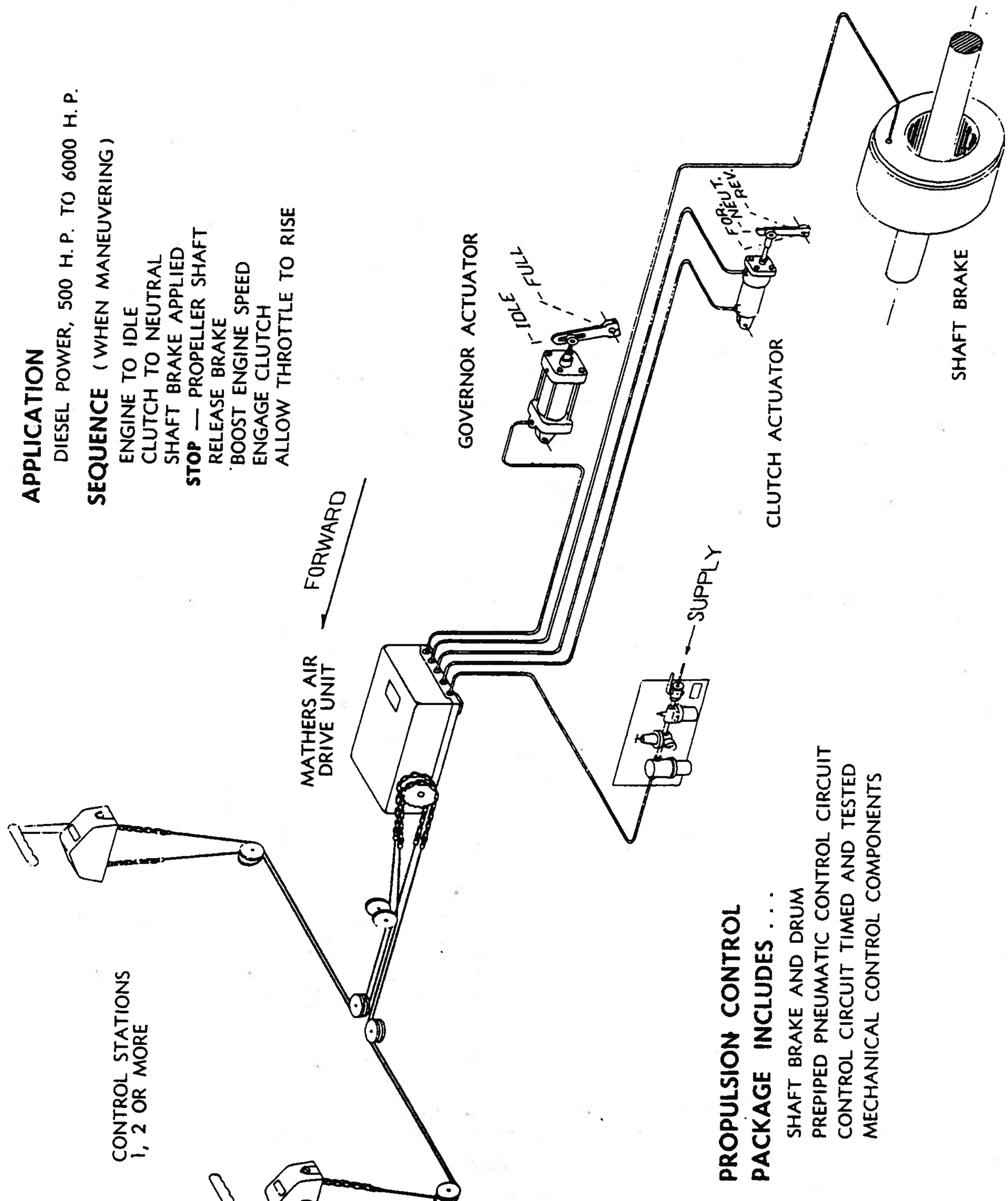


FIGURE 8.9 Pneumatic Propulsion Control System as Furnished by Mathers Controls, Inc, Seattle, Washington

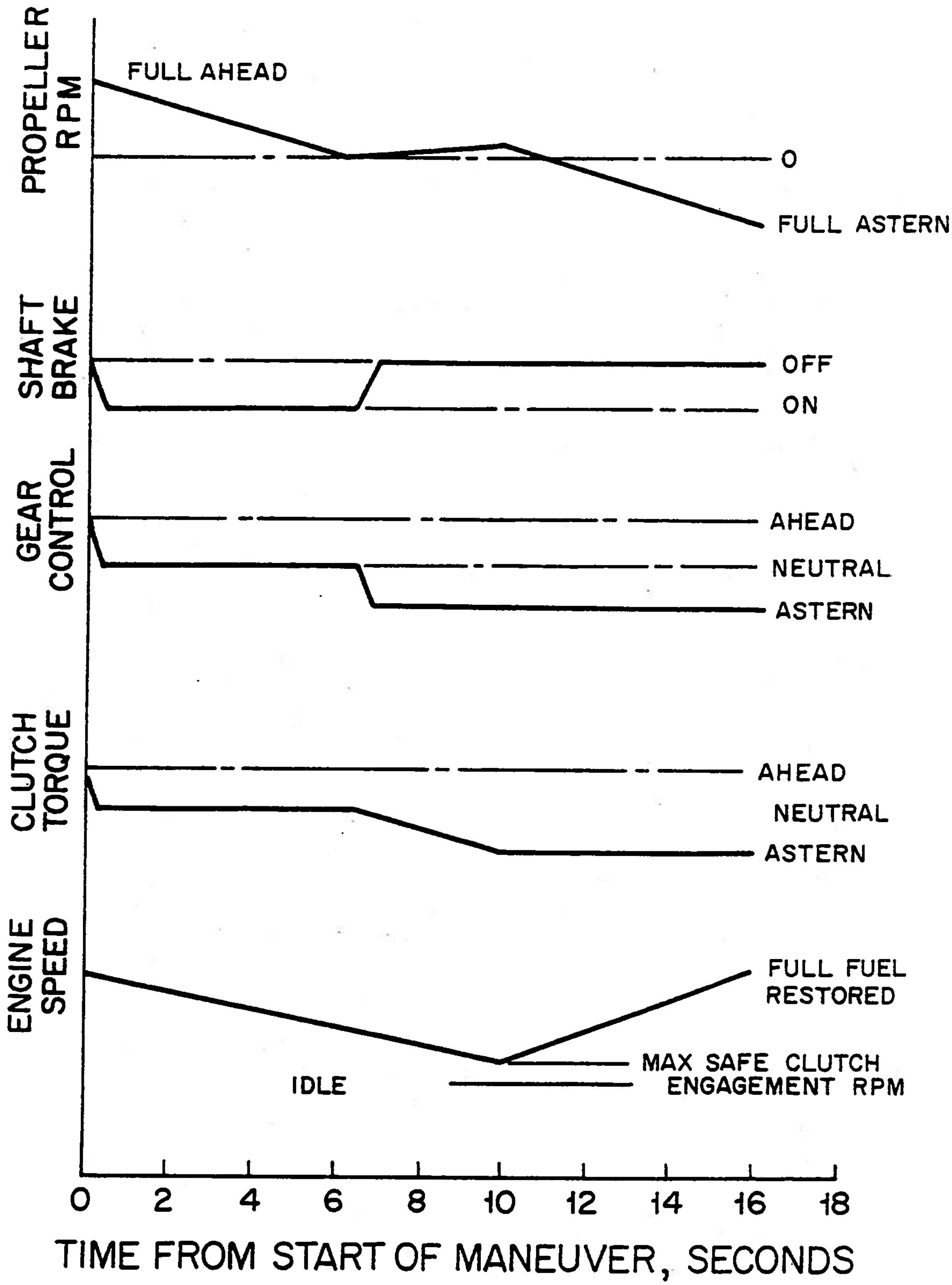
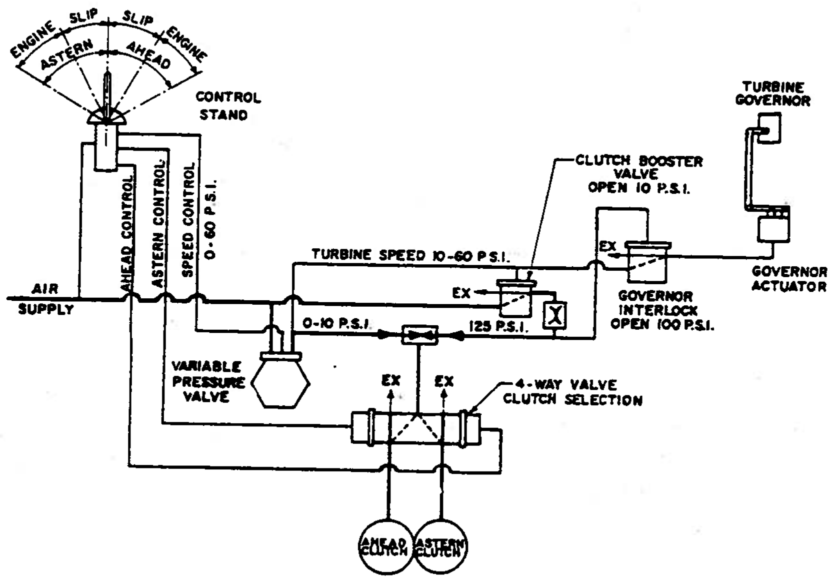


FIGURE 8.10 Crash Stop for a Tug with Reverse Gear and Shaft Brake (from Reference 2)

to attain the necessary heat absorption and dissipation. Slippage is allowed by reduced actuating pressure, this pressure being proportional to position of the control handle between fully disengaged and fully engaged points. Figure 8.11 sketches a pneumatic system that incorporates this feature, and a description is quoted below from the paper that furnished the figure 5 . This system is actually for the 20,000 shp plant of the gas turbine ship Admiral Callaghan but serves quite well as an example for small craft also.

> It was determined that 10-psi air supplied to the clutches was sufficient to transmit maximum propeller torque at turbine idling speed. The clutches could therefore, be controllably slipped for lower propeller speed by regulating the clutch air pressure below 10 psi.
>
> Stripped of frills and reduced to essentials, the basic pneumatic control circuit is as shown in Figure 8.11. The fundamental element is a cam-operated pressure-regulating valve that, located in the engine-room console, clutch air pressure for the slip mode and, via an actuator linked to the turbine governor, controls turbine speed in normal operation. Two slide valves actuated by the same cam shaft provide a 125-psi pressure signal to one or the other of two lines to operate a four-way valve for clutch selection whenever the control handle is moved from neutral to any position in either the ahead or astern quadrants.
>
> On the Callaghan, remote bridge control was accomplished by interconnecting the control-handle shafts in engine-room and bridge consoles by an electrical Selsyn system which gave some initial difficulties--since corrected--with sensitivity and repeatability; but a mechanical connection, or a duplicate controller station as normally used on tugboats, would also have been feasible.
>
> In any case, the variable pressure pneumatic signal from the control stand is reproduced by the variable pressure valve on the gear panel and supplies working air pressure to the clutches and the turbine governor actuator. At 10 psi, corresponding to both clutch "lock-up" at turbine idling and minimum governor speed signal, the output of the panel variable pressure valve also opens an on-off booster valve, which thereupon supplies the clutches with air at the full 125-psi line pressure.
>
> A governor-interlock valve, sensing clutch air pressure, intercepts the speed signal to the turbine governor, delaying any action to increase turbine



from Reference 5

FIGURE 8.11 Pneumatic Control Circuit with Slipping Clutch Feature

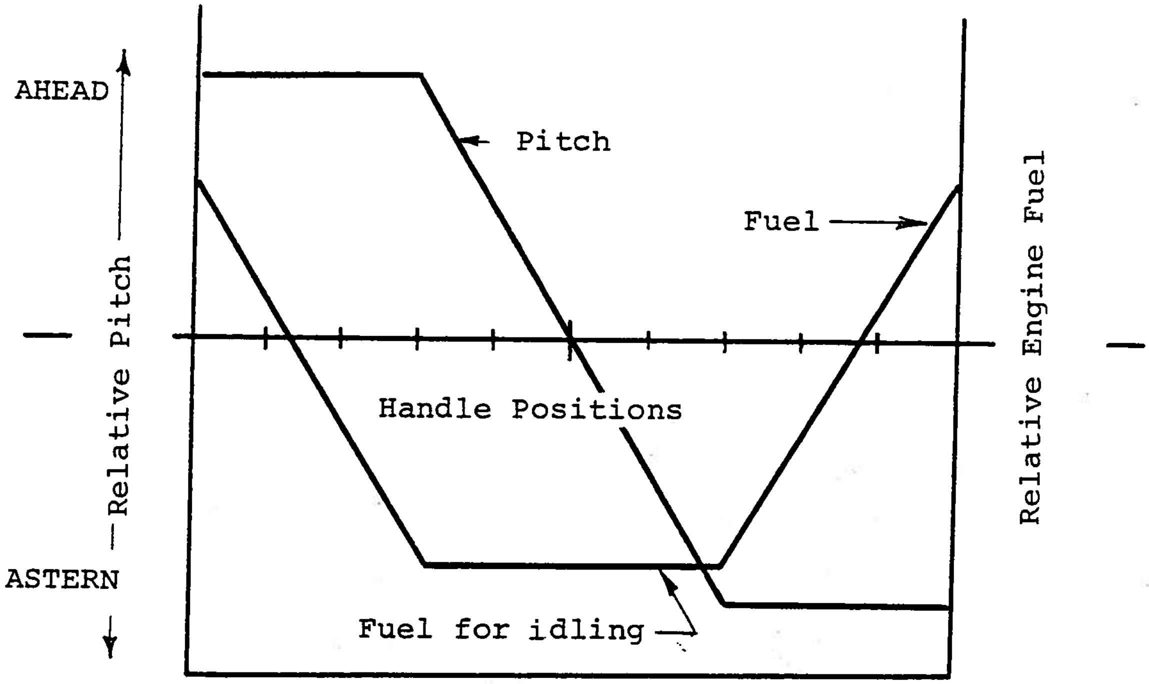


FIGURE 8.12 Elementary Program for Propeller Pitch and Engine Fuel as Functions of Control Handle Position

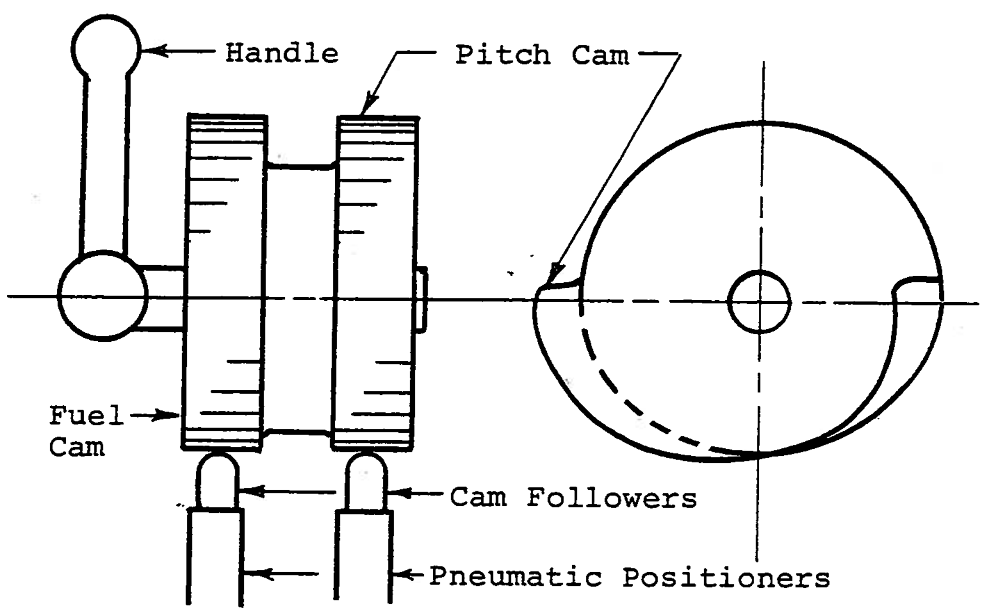


FIGURE 8.13 Cams to Effect Input to Pneumatic System -- Program as in Figure 8.12

power until the clutches have reached a suitable operating pressure. Conversely, any loss of clutch air pressure, for whatever reason or at whatever rate, to a value less than the setting of the intercept valve will return to governor position to turbine idle.

A number of refinements were also included in the basic system. These included a double boost valve to ensure snap action, biasing the interlock to speed up governor action for low-speed maneuvering, and a boost system for the turbine governor. The latter involved a second actuator, operating in parallel with the governor speed actuator through a mechanical linkage, and an additional valve or two in conjunction with a "memory tank." The functions are, first, to put the turbine on governed speed control whenever the throttle is advanced beyond the clutch slip range, while allowing it to operate ungoverned in the slip range; and, secondly, to "remember" for a period of several minutes when the turbine has been at full power so that, in the event of a reversal from full ahead to slow astern, the turbine rotation of the turbine due to propeller windmilling. The latter was probably an unnecessary precaution but cost little extra in complication.

8.4 INTEGRATED CONTROL OF ENGINE AND CONTROLLABLE-PITCH PROPELLER

If a controllable-pitch propeller is used, the clutches associated with a reverse gear train are absent, since reversal is accomplished by reversing pitch. In effect, control of pitch replaces control of clutch engagement. As noted in Chapter 6, pitch and fuel can be controlled individually, but for reasons mentioned there, these two propulsion plant variables are usually linked together in an integrated control system. A simple mechanical system might be like that of Figure 8.6, with the clutch cable instead operating the pitch actuator. The cams, or whatever, that translate handle movement into cable movement might be designed to produce a combination of pitch and engine fuel setting as it appears in Figure 8.12; Figure 8.13 is a sketch of cams to show how the handle, cams, and cables might be related. It is apparent that in this example pitch change is used only to reverse, and to provide low vessel speed below the normal operating range of an engine-fixed-pitch combination. (You might recall the discussion of pitch-fuel programs in Chapter 6, and try as an exercise to plot Figure 8.12 on the power-RPM plane used in that chapter. As another, you might try plotting Figure 6.11 in the manner of Figure 8.12.) More complex design of the cams can produce combinations of pitch-fuel such as those discussed in Chapter 6. And the same thing can be accomplished with pneumatic components, or hydraulic, as actuators and relays.

The missing performance element in the above discussion (again recalling Chapter 6) is the automatic response to changes in external conditions. There may be no such response provided, but when one is, it is usually the constant-RPM response, and this requires at least a pitch-control governor to provide the automatic element. The Escher Wyss/Greiner system, discussed in Chapter 6, makes use of a fuel governor and a pitch governor with characteristics matched as explained there (Figure 6.9). The command inputs to the governors could be accomplished by push-pull cables, with the program being physially present in the cams, as before. Or hydraulic or pneumatic (more common) elements might be used to relay the orders of the cam surfaces to the respective governors.

8.5 CLOSURE

There are some topics in propulsion plant control that cannot be covered in a single chapter such as this. The governor, for example, is an automatic control device, and whenever such elements are incorporated in a system, problems of speed of response and stability should be examined. This is a complex and lengthy topic for which a variety of modern textbooks can be consulted. I hesitate to make recommendations, though have already mentioned one book as a good starter [1].

Speed of response may be especially important with a gas turbine because of the low inertia of the engine itself. Overspeed protection, as mentioned in Chapter 5, must be designed with respect to potential changes of speed from pitch change, opening time of clutches, or emersion of propeller. Papers of Rubis [6] are about the only source of analysis of these effects.

Where pneumatic or hydraulic components are used, the unique characteristics of their components must be understood. Reference 7 gives an elementary description (lots of pictures). References 8, 9 treat the subject on a level appropriate to the engineering student.

8.6 REFERENCES

1. D. B. Wellbourn, The Essentials of Control Theory for Mechanical Engineers, the American Elsevier Publishing Company, 1963.

2. C. R. Ford, "Aspects of the Mechanical Propulsion of Tugs," Transactions, Institute of Marine Engineers, Vol. 78, No. 1, 1966.

3. W. R. Forster, R. C. Brooks, M. F. McAllister, "Interlocked Marine Reverse Gear Pneumatic Control," Society of Automotive Engineers Paper 528c, 1963.

4. John B. Woodward, The Diesel Engine: To Drive a Ship, Report 105, Department of Naval Architecture and Marine Engineering, University of Michigan, 1970.

5. W. S. Richardson, "The Friction Clutch Reverse-Reduction Gears for the GTS," ASME Paper 69-GT-5, 1969.

6. C. J. Rubis, "Transient Response of a 25,000-HP Marine Gas Turbine Engine," ASME Paper 71-GT-61.

7. Dudley A. Pease, Basic Fluid Power, Prentice-Hall, 1967.

8. J. F. Blackburn, Gerhard Reethof, F. J. Shearer, Fluid Power Control, the MIT Press, 1960.

9. B. W. Anderson, The Analysis and Design of Pneumatic Systems, John Wiley, 1967.

CHAPTER 9

DRIVELINE COMPONENTS

9.1 INTRODUCTION

The shaft is the most obvious component of the driveline between engine and propeller. It is indeed a simple device, but it demands a fair share of attention in the design process. A diameter sufficient to transmit the torque from engine to propeller and the thrust from propeller to thrust bearing must be chosen, though only an elementary calculation is required. Careful alignment of the shaft is important, since poor alignment can put excessive loads on bearings and on gear teeth. Alignment in a small vessel usually requires little in the way of design calculation, even though it is an item for careful attention by installation personnel. Nonetheless, the vessel designer should be aware of the problems possible, and of calculation techniques that might have to be used. The biggest shafting problem, if it does occur, is vibration. Torsional vibration is the potential bane of all shafting systems, especially those associated with reciprocating engines, and will occur if the properties of engine, propeller, gears, and shaft fall into the wrong combination. A whipping vibration is also possible, and is one of the major determinants of bearing spacing.

Of these shafting problems, this chapter gives a reasonably complete story (with help of references) on all but the torsional vibration; it is a book-size rather than a chapter-size problem. Some hints and explanation of the problem are included here, however.

The reverse-reduction gear is not quite the essential that shafting is, but is an adjunct found in practically all small craft, since there is a great disparity between the rated speeds of available engines and the best-efficiency speeds of propellers. Reversal of the shaft is essential except when thrust reversal is provided by a controllable-pitch propeller. Changes in direction in the shaft line are often accomplished by gearing also, as with the fairly common V-drive, and in the modern outboard motor.

The design of gearing is a complex matter, almost a field of engineering in itself. But just as with the engine, the marine designer is unlikely to be involved with component design; his gears will usually be stock models, or at the very least designed by specialists. Some hints as to the specialists' problems are touched on here, however, to aid in the marine designer's understanding of what he can and can't have in reduction gearing.

Clutches, couplings, bearings, brakes, and universal joints are the other items frequently found in propulsion drive lines. Some of their aspects are touched on here.

9.2 REDUCTION GEARING

There is good incentive to make the rotational speed of propulsion engines as high as practical. For the reciprocating engines, the P L A N formula makes the reason clear: for a given power, higher N means a lower displacement (i.e., L x A), hence a lighter engine. The trend toward higher N is limited by acceptable values of piston speed and bore/stroke ratio, and generally the engine designer pushes against these limits, so that the typical reciprocating internal-combustion engine is about as "high speed" as is reasonable to make it. As a result, engines are usually of much too high an RPM for good propeller efficiency. The almost universal answer is the reduction gear, which is produced in reduction ratios ranging up to 6:1. The alternatives are to accept a very poor propeller efficiency because of greatly excessive RPM, or to build a big (large L x A), heavy, engine with an N to suit the propeller. Neither is usually worth considering in the face of competition from the high-speed stock engine and the efficient and relatively cheap stock reduction gear.

Much the same can be said for the gas turbine, though to be sure the P L A N formula does not apply. For acceptable turbine efficiency, RPM must be such that linear blade speed is roughly one-half the jet speed of the hot gas spinning the turbine. The result is again an RPM that is much too high for the propeller to use efficiently, and a reduction gear is the answer. A low-RPM turbine could be built, but a linear blade speed giving good efficiency could be had only by increasing wheel diameter to compensate. That would spoil the very appeal of gas turbines: their compactness. So as with the reciprocating engines, the answer is to pursue high RPM in the engine design, using the reduction gear to obtain a reasonable propeller RPM.

The typical small craft reduction gear uses a single pair of helical-tooth spur gears in a casing that bolts directly to the engine flywheel housing. Its input member (drive ring) is bolted to the flywheel, though there is usually a flexible element of some sort between it and the pinion (pinion = the input, or high speed, gear) shaft. A reverse gear train is nearly always included, it consisting of an idler gear that reverses the rotation of a second pinion that also meshes with the bull gear (bull gear = the output, or low speed, gear). Both pinions are driven by the input member, each via its individual clutch. The gear set drives ahead or astern, depending on which clutch is engaged. If neither is, the gear is, of course, in neutral. Clutches are usually of the disc type, and are actuated either mechanically by a lever, or hydraulically. If the latter, the set's lube oil, pressurized by an internal pump, is the actuating medium, but with its pilot valve in turn positioned manually.

Figure 9.1 is a cut-away view of a typical small craft gear (Twin Disc) of the type discussed above.

An alternative to the simple pinion gear arrangement is the planetary gear, sketched in Figure 9.2. The virtue of this scheme is that it spreads tooth loads over several paths, meaning in consequence that individual gears can be smaller, producing an overall more compact set. Another feature that is an advantage for some installations is the in-line configuration of input and output. In a typical marine application, [1] the ring gear is fixed to the gear case, sun gear is input, and output comes from the planet gear carrier. With the ring fixed, the reduction ratio is $1 + r$, where r is the ratio of number of ring teeth to number of sun teeth.

Figure 9.3 is a cut-away view of a gear used with diesel engines of the 1000-3000 hp range (figure is from application literature of Electromotive Division, General Motors). Although it is in principle the same as the Twin Disc gear of Figure 9.1, several detail differences are apparent. The clutches are pneumatic, and are mounted external to the gear case. The unit is not to be mounted on the engine flywheel housing, but is mounted separately on its own foundation. The gear set could be remote from the engine, coupled to it by a high-speed shaft, though the more usual installation finds engine output flange and gear input flange directly mated, perhaps with a flexible coupling interposed.

Figure 9.4 illustrates a gear used to accomplish the V-drive arrangement of engine and shaft (from literature of Precision V-Glide Corp.). The main purpose of such gears is to change direction of the shaft line. Therefore they are likely to be used in addition to the reduction gear, as in the figure. V-type units are also available with reduction ratios, however.

9.3 GEAR DESIGN AND SIZING

The main concern in gear design is the tooth. Put crudely, it has to be strong enough to transmit the load to a mating tooth without failure. "Strong enough" implies two things: resistance to gross breakage, and resistance to surface damage in the area of contact with the other tooth.

The tooth is a cantilever beam, loaded at its line of contact, and with the maximum stress occurring at its root.

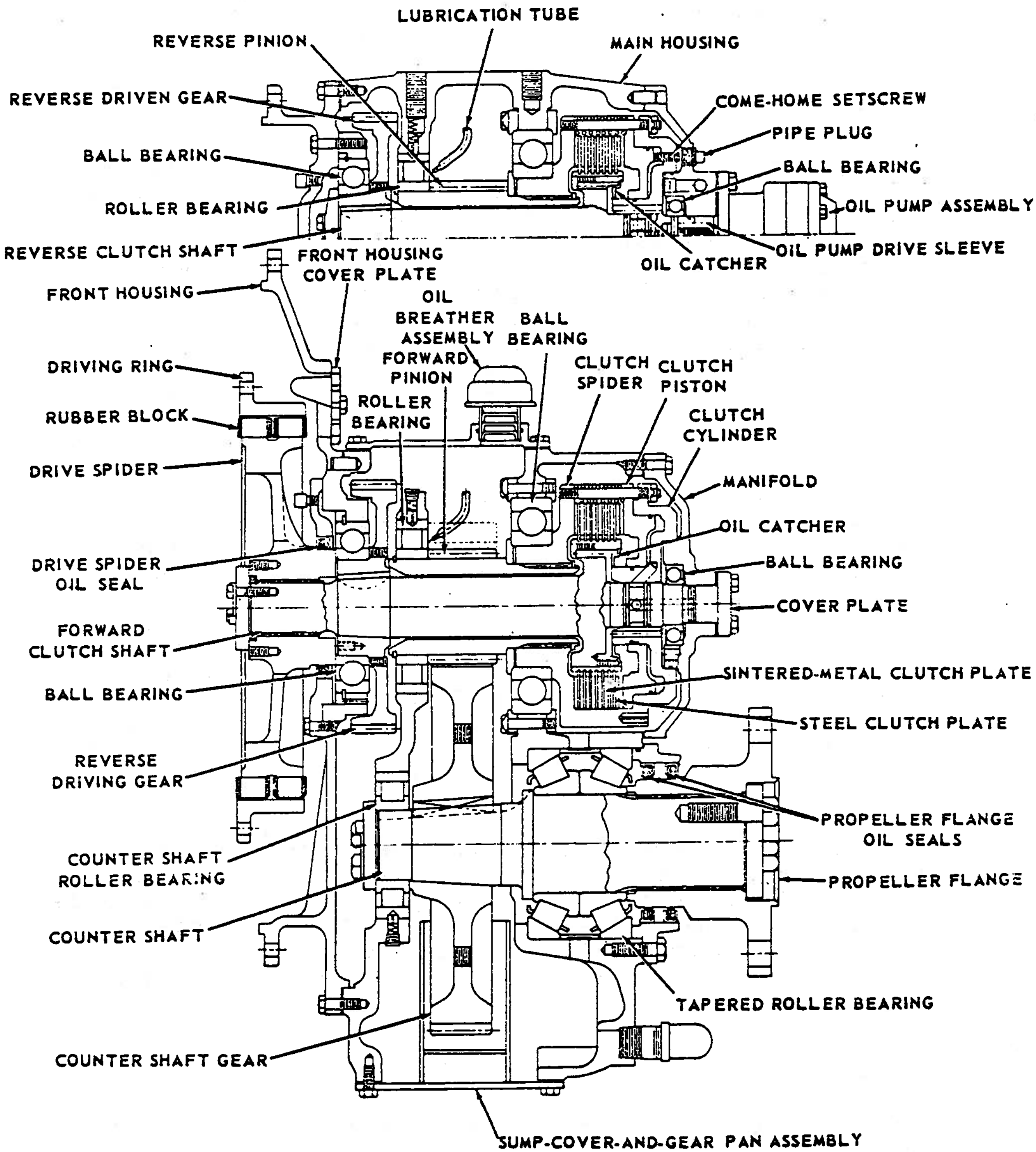
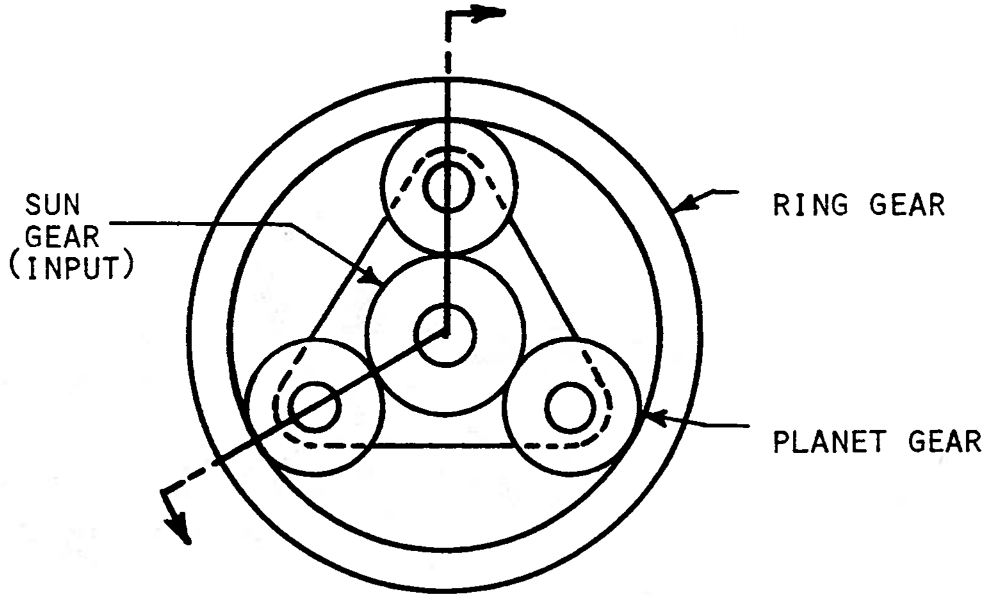
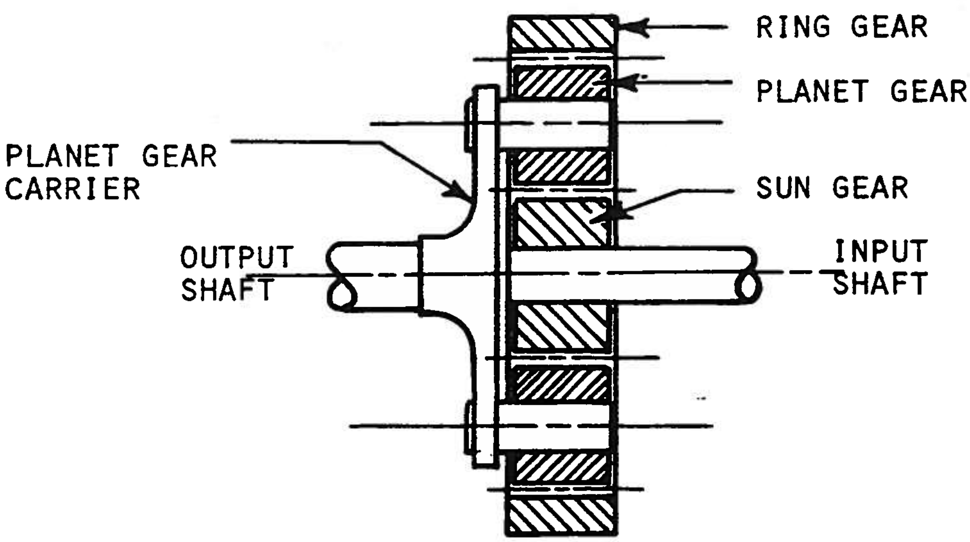


FIGURE 9.1 Cross-Section View of Reduction Gear for Internal Combustion Engine (Twin-Disc MG514)

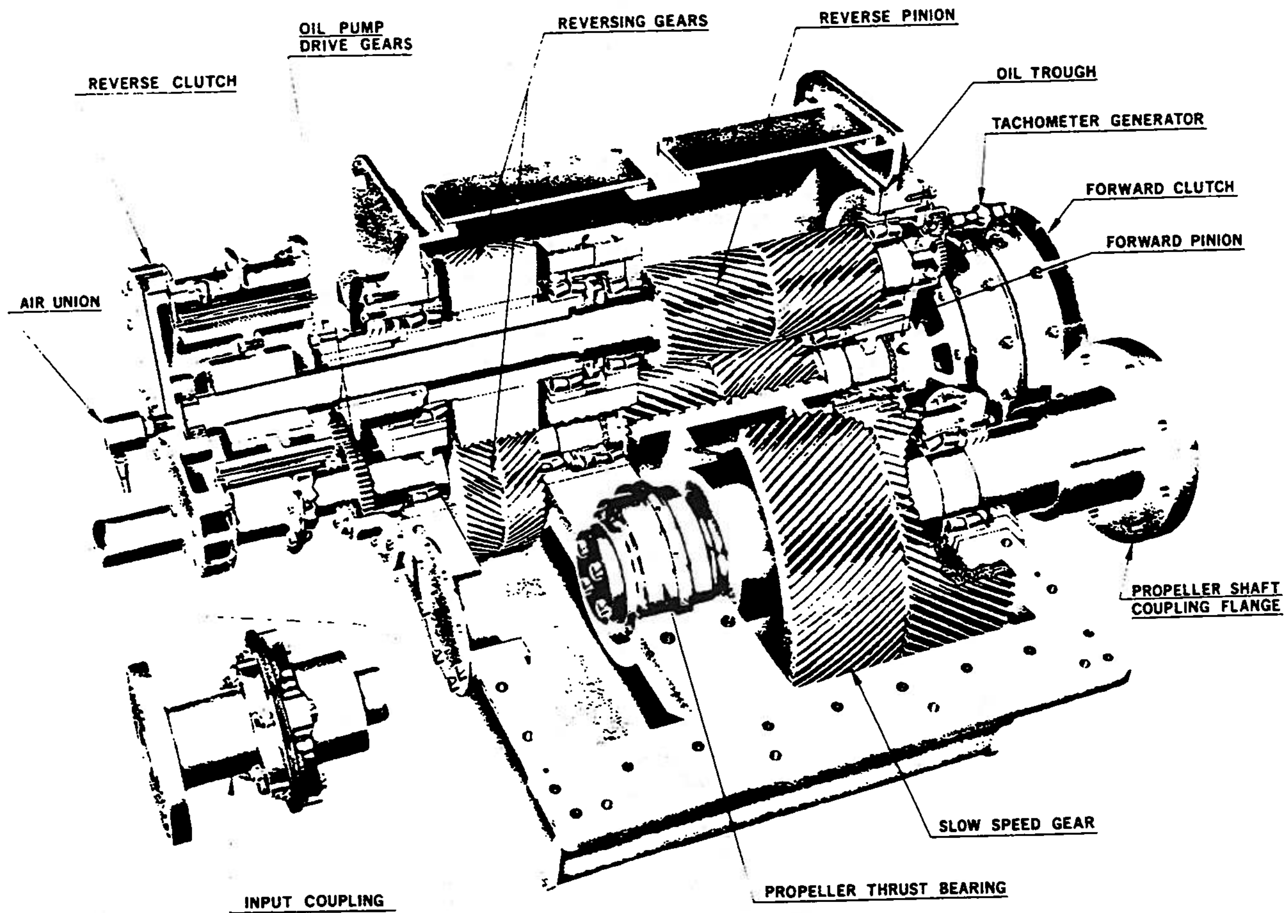




IN A TYPICAL MARINE APPLICATION (REFERENCE[1]), THE RING GEAR IS FIXED TO THE GEAR CASE, SUN GEAR IS INPUT, AND OUTPUT COMES FROM THE CARRIER SHAFT. WITH RING FIXED, THE REDUCTION RATIO IS $\frac{T_R}{T_S} + 1$ WHERE T_R = NUMBER OF RING TEETH

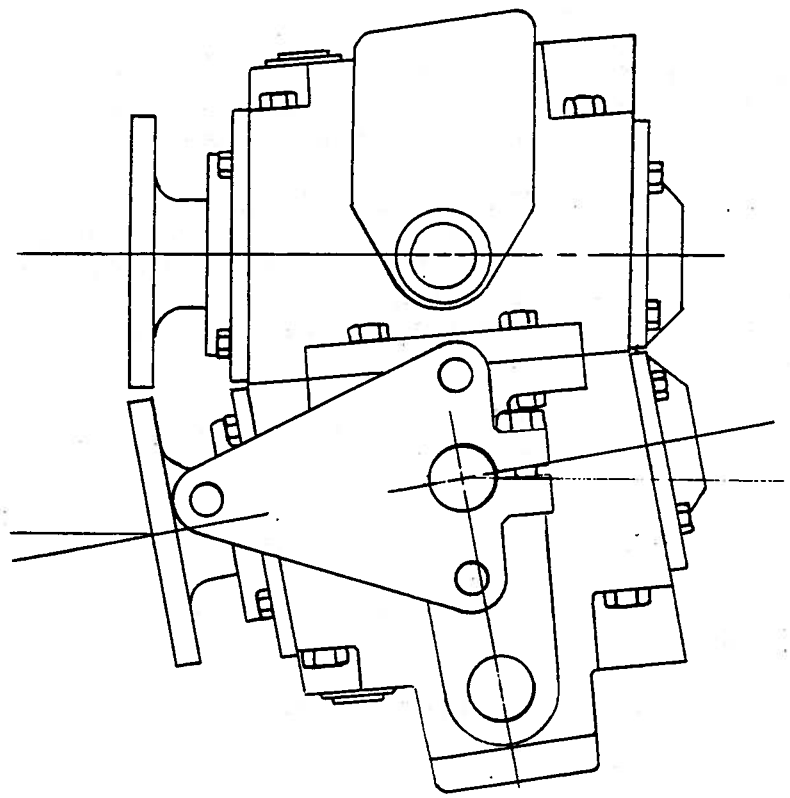
T_S = NUMBER OF SUN TEETH

FIGURE 9.2 Elements of Planetary Reduction Gears



Gears

FIGURE 9.3 Reduction Gear for Medium-Speed Diesel
(Courtesy Electromotive Division, General Motors)



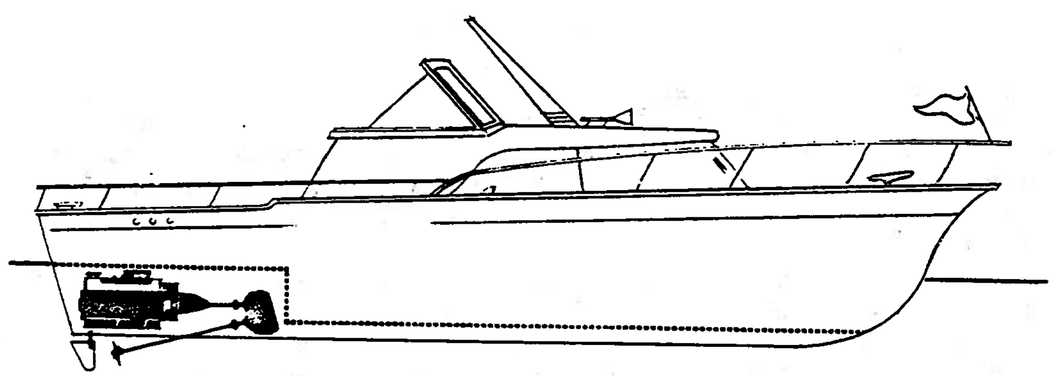


FIGURE 9.4 Marine V-Transmission (courtesy Precision V-Glide Company)

When such a beam is of simple shape, and loaded statically by a known force, calculation of the stress is an elementary matter. The gear tooth, on the other hand, does not have a constant cross-sectional area, and is subjected to a load that is varying in direction and magnitude. The second of these complications is perhaps more severe, because part of the varying load is caused by errors in manufacture, deflections that change contact area, and torsional vibration; these things and their effects are difficult to assess. Actual design formulas must account for them, but are able to do so only by means of correction factors based on experience, and semi-empirical analysis. To give an example, the AGMA (American Gear Manufacturers' Association) formula for bending stress is: [2]

$$\sigma = \frac{W_t K_o}{K_v} \frac{P}{F} \frac{K_s K_m}{J} \tag{9.1}$$

where σ = bending stress at base of tooth, lb/in^2

P = diametral pitch (number of teeth per inch of pitch diameter)

F = face width of tooth, inches

W_t = load transmitted, lbs

J = geometry factor

K_o = overload factor

K_v = velocity correction factor

K_s = size correction factor

K_m = load correction factor

A significant thing here is that the equation is largely a collection of "factors." Their values must be available to the gear designer from experience with previous designs or from tests. No attempt is made to reproduce values here, there being no intent to teach the use of this formula.

Gear teeth frequently fail in the area of tooth contact, rather than by complete tooth breakage. Pitting occurs along the line of contact, for example. Although it may be difficult to say exactly what the pit-digging mechanism is, it is an over-stress of some nature in the vicinity of the pit. Hardness of the tooth material is the most significant resisting property. To demonstrate, the AGMA surface durability formula for helical gears is: [3]

TABLE 9.1 Material Hardness

MATERIAL HARDNESS (MIN BHN)		Q-FACTOR
Pinion	Gear	
210	180	1.00
245	210	1.18
265	225	1.27
285	245	1.43
300	255	1.51
315	270	1.66
335	285	1.80
350	300	1.97
365	315	2.15
380	335	2.38
500	350	2.08
500	500	3.30
550	550	3.50
600	600	3.70

TABLE 9.2 Service Factors

Application	Operational (1) Mode Rating	Service Profile Based on 20 yr. Life	Motor or Turbine	Internal Combustion Engine (Multi-Cyl.)
Merchant Ship	Max Load	Continuous	1.5	1.75
Naval Ship	Full Power Max Load	5 percent Max Continuous	1.0 1.5	1.5 1.75
River Craft (Towboat, Tugboat)	Max Load	Continuous	1.5	1.75

(1) Max Load condition does not necessarily occur at full power. With multiple power plants, shaft cross-over operation, and controllable pitch propeller, single engine operation might well be the max. load condition for certain elements in the power path and they must be rated accordingly.

$$P = \frac{110\ C^2\ F\ n\ M\ Q}{S} \tag{9.2}$$

where
P = horsepower transmitted

C = center-to-center distance of mating gears, inches

F = face width, inches

n = pinion speed, RPM

M = a factor that is a function of gear ratio

Q = material hardness factor

S = service factor

Again, the formula is given for illustrative purposes, so that values of all the factors is not reproduced here. However, a subsequent discussion will make use of Q and S, so that it is appropriate to quote their values, and this is done in Table 9.1 and Table 9.2. Both are taken from Reference 3, and apparently the service factors in the latter table are not from AGMA, but originate with the authors of the reference, or with their organizations (Western Gear Corp. and NAVSEC-Philadelphia).

A significant point in the present context is the higher service factor throughout for the reciprocating engines. The cause is the unsteady torque of these engines which imposes a "clashing" load on the teeth.

So the gear designers have their problems which may be simple in principle, but require major inputs from experience factors for actual design solutions. In considering marine reduction gears, the local failure of the tooth along its contact line has been the more serious of the two main problems, and marine engineers have developed the "K-factor" to determine acceptable levels of loading with respect to local failure. This factor is a number expressed in terms of gear ratio, pinion diameter, and tooth face width, and thus gives a size of gear if a value of K is known. This in turn allows an estimate to be made of the overall size of the gear set. The formula for K is related to equation (9.2), since it represents a solution to the same problem, and so serves as a link between the gear designer and the marine designer.

The basis for the K-factor formula is the Hertz equation for compressive stress developed in two elastic cylinders being forced together by a load W. The formula is

$$\sigma_{max} = \left[0.175\frac{W}{L}\,E\left(\frac{1}{r_1} + \frac{1}{r_2}\right)\right]^{-\frac{1}{2}} \tag{9.3}$$

where σ_{max} = max compressive stress near line of contact, lb/in^2

L = length of contact between cylinders, inches

E = modulus of elasticity, lb/in^2

r_1, r_2 = radii of the cylinders, inches

Gear teeth aren't cylinders, but their involute profiles can be approximated by circular arcs near the contact line by using the involute radii of curvature in the formula. See Figure 9.5 for geometry. In this figure, note also the following relations:

$$r_1 = R \sin \alpha$$

$$r_2 = r \sin \alpha$$

$$W = W_t / \cos \alpha$$

Putting these into equation (9.3), and using $R_1 = R/r$ (i.e., R_1 is the reduction ratio), $E = 30 \times 10^6$, and $D_p = 2r$ (D_p = diameter of pinion), gives

$$\left[\frac{\sigma_{max} \sin 2\alpha}{4582}\right]^2 = \frac{W_t}{L\,D_p}\,\frac{1 + R_1}{R_1} \tag{9.4}$$

The terms on the left are those that should be the same for all successful gears, while the terms on the right have values determined by the particular gear and its application. The left being nominally a constant, it is abbrevaited, and the formula becomes

$$K = \frac{W_t}{L\,D_p}\,\frac{1 + R_1}{R_1} \tag{9.5}$$

A value of K might be calculated from an allowable maximum stress, but the uncertainties mentioned previously make it difficult to pinpoint this value by calculation or theory. Rather, K is thought of more as an experience factor, i.e., gears of $K = X$ having given satisfactory service, X is subsequently specified as a value of K when ordering the next gear set.

Equation (9.5) is closely related to equation (9.2). It is shown in [3] that they are the same if

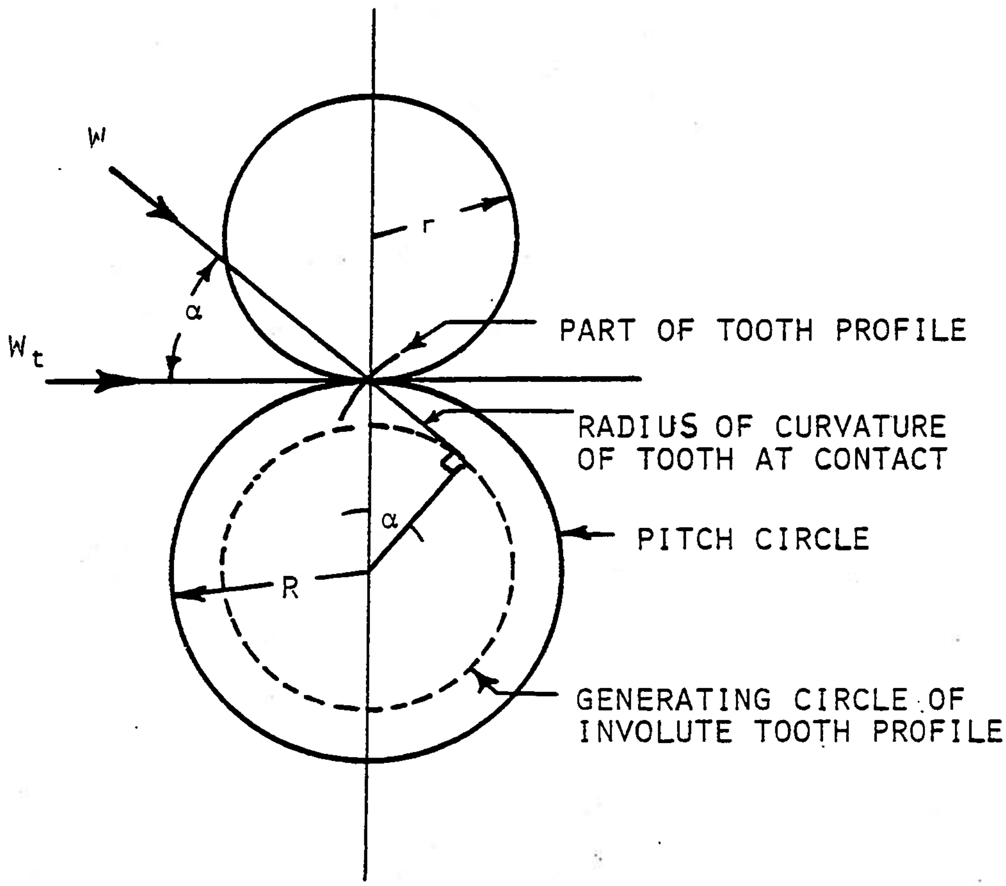


FIGURE 9.5 Geometry for K-Factor Derivation

$$K = 110 \frac{Q}{S} \tag{9.6}$$

K is thus seen to be a function of the gear and pinion surface hardness, and of the service, i.e., the harder the surfaces, the higher the K that can be allowed, and the more severe the service, the lower the K. Other factors can also influence the designer in his choice of K. For example, higher K is often specified for the reverse train gears because of their comparatively short use during the life of the vessel. Tables 9.1 and 9.2 have been included to facilitate the use of equation (9.5) via (9.6).

Having a reasonable value of K, the designer can use equation (9.5) to rough out the size of the gears, and hence, of the gear set. He must know the torque to be transmitted (to get W_t), the distance between pinion and gear centers that can be allowed by space considerations, and the gear ratio, but with these in hand, the equation ties everything together. Try it.

9.4 SHAFT SIZING

Shaft diameter is determined by the torque transmitted, and by the maximum stress that can be allowed in the material. From any elementary textbook [4] on strength of materials, the following basic formula for stress can be taken:

$$\tau_{max} = \frac{16Q}{\pi d^3} \tag{9.7}$$

where Q = torque, inch-lb

d = shaft diameter, inches

Or, making use of the relation between torque and power, this can be turned into a formula for diameter in terms of allowable τ_{max}:

$$d = 68.5 \left[\frac{P}{N \tau_{max}}\right]^{\frac{1}{3}} \tag{9.8}$$

where P is horsepower and N is shaft RPM.

Use requires, of course, values for the maximum stress. The yield stress in torsion is appropriate, but a modifying factor is introduced into the working values, mostly as the usual factor of safety, but also to make allowance for the type of service that the shaft is to receive. Thus an experience factor is involved, and one must resort to particular sources of marine design information, rather than to general handbooks.

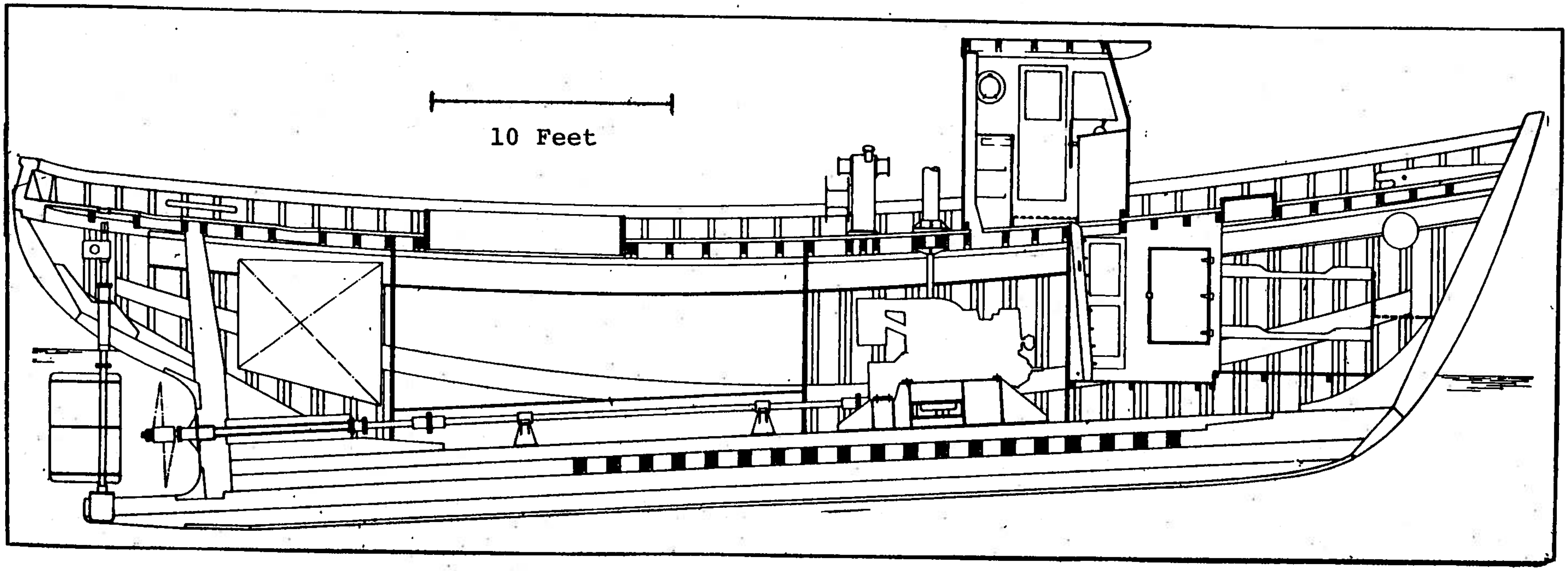
For pleasure boats, Standard P-6 of the American Boat & Yacht Council [5] contains tables of stress values, and also advice on the factor to use. This standard also presents details of shaft straightness, shaft couplings, and propeller-end machining, repeated from applicable Society of Automotive Engineers standards. For more rugged service, the shaft diameter must be greater than for pleasure craft, perhaps as much as 50% greater. Some diesel builders, in particular, publish tables of shaft sizes for their engines, and these give the details for the particular application. For more general use with commercial craft shaft sizing, the rules of the American Bureau of Shipping [6] are appropriate.

The propeller thrust bearing is nearly always a part of the reduction gear, so that the shaft transmits thrust as well as torque. The shaft is thus subjected to a combined torsional and compressive stress; formulas for the combined stress can be found in any standard strength of materials textbook, but are not really required for the usual design work. The thrust contribution is usually comparatively small, and is absorbed in the factor of safety, or is allowed for in a simple fashion. For example, the American Bureau of Shipping [6] specifies a 10% increase in diameter over basic formula size for the portion of the shaft carrying thrust.

9.5 SHAFT BENDING

In addition to being a transmitter of thrust and torque, the shaft is also a beam, i.e., a member that must resist bending. The bending loads are nominally the weight of the propeller and the weight of the shaft itself. In small vessels, both of these usually cause negligible stresses and deflections in the shaft, whereas they are significant with the heavier shafts and propellers of ships. A third possible source of load is misalignment of bearings. Take a look at Figure 9.6, an inboard profile of a fishing vessel with long shaft line, supported by two intermediate bearings. It should be intuitively obvious that if such a bearing is offset somewhat it will bend the shaft, causing reversing stresses as the shaft rotates, excessive pressure on the bearings, and possibly distorting moments on the reduction gear. Fatigue failure of the shaft, burning out of bearings, and rapid wear of gear teeth are the evil consequences that could follow in severe cases.

In a small vessel, the solution to these problems is elementary; make the shaft line absolutely straight, i.e., have the centerlines of propeller, bearings, stuffing box, couplings, and low-speed gear precisely coincident. Alignment of shafting in this situation is only a test of skill for the man with the tools. However, perfect straightness is not necessarily the best alignment, and often is not satisfactory in high-power propulsion plants; these are the two basic reasons:



Design by Fred Bates, Damariscotta, Maine. From the National Fisherman, June 1969. The line shaft is $3\frac{1}{2}$ inch AISIC steel, tailshaft is monel 400 of the same diameter. Engine is GM 12V-71, continuous RPM is 1800, with 4.13:1 reduction gear.

FIGURE 9.6 Shafting Illustration

1. For a uniformly loaded beam on multiple supports (more than two), placement of supports in line does not produce the minimum stress in the beam, nor the most even distribution of support loads. Lowering some bearings to follow the "natural bent" of the shaft produces the least bearing loads and the least bending stress.

2. Thermal expansion of the machinery as it heats to operating temperature can change the initial alignment. If the reduction gear coupling moves up significantly when hot, then it may be advisable to set it low in the cold condition to compensate.

Both of these problems are a function of size, i.e., they become of greater concern to the designer with increasing horsepower ratings of the propulsion plant. At the low end of the power scale, they are usually of no concern at all, and straight-line alignment is, to repeat, the satisfactory approach. On the other hand, there is no sharp dividing line between "large" and "small," between "ship" and "boat," and so the marine designer should be able to calculate bearing loads and shaft stress in order to estimate that the need for any special alignment. For this reason, there follows an exposition of several calculational techniques that may be useful.

It is assumed at this point that the reader is familiar with elementary beam theory, and the discussion therefore takes off immediately into the method of three moments, which is a method of analysis needed for beams supported on more than two bearings, a situation typical of most propulsion shafts. The method is briefly as follows:

1. The shaft is initially considered to be composed of individual spans, simply supported, and uniformly loaded. The standard formulas for bending moment along the length of each span, and for the slope at each end, are written.

2. Since the shaft must actually be continuous at each support, i.e., the slope of adjoining spans must be the same at this point, a moment is assumed to exist at the support that has just the direction and magnitude needed to make those slopes be the same. The magnitude is unknown, and must therefore be solved for.

3. An equation is written at each support, expressing the equality of slopes, with the moments just mentioned as unknowns.

4. These equations are solved for the moments mentioned in (2).

5. Since all bending moments acting on the shaft are now known, solution for stress and support reaction proceed as in an ordinary elementary beam problem.

Details of derivation can be found in Reference 4. Here the results only are listed. See Figure 9.7 for illustration. First, the equation for M_n, the bend-into-line moment at the n^{th} support:

$$M_n = \frac{-1}{2(\ell_n+\ell_{n+1})} [6EI(\beta_{n+1}-\beta_n)+\frac{w}{4}(\ell_n^3+\ell_{n+1}^3)+M_{n-1}\ell_n+M_{n+1}\ell_{n+1}] \quad (9.9)$$

where β = angular misalignment (see below)

E = modulus of elasticity, lb/in^2

$I = \pi d^4/64$ = plane moment of inertia, in^4

ℓ = length of shaft span, inches

$w = \pi d^2 \rho/4$ = unit weight of shaft, lb/in

ρ = shaft density, lb/in^3

d = shaft diameter, inches

The angle β is zero when all supports are in line. Otherwise

$$\beta_n = (\Delta_{n-1} - \Delta_n)/\ell_n \quad (9.10)$$

where Δ = displacement of support, inches (positive if up). Note that equation (9.9) will contain three unknowns, M_n, M_{n-1}, and M_{n+1}. If it is written in turn for each support, a set of equations results that is solved simultaneously for all M.

If the ends are supported, as in Figure 9.7, the end moments are zero (i.e., $M_0 = M_3 = 0$ in this figure). If an end supports an overhanging weight (e.g., a propeller), the end moment must be the cantilever moment required to support this weight and the overhanging shaft. For example, if the shaft in Figure 9.7 is extended ℓ_0 to the left and a weight W_0 hung on, then $M_0 = (W_0 + w\ell_0/2)\ell_0$.

The reaction at each support is

$$R_n = R_n' + \frac{M_{n-1}-M_n}{\ell_n} + \frac{M_{n+1}-M_n}{\ell_{n+1}} \quad (9.11)$$

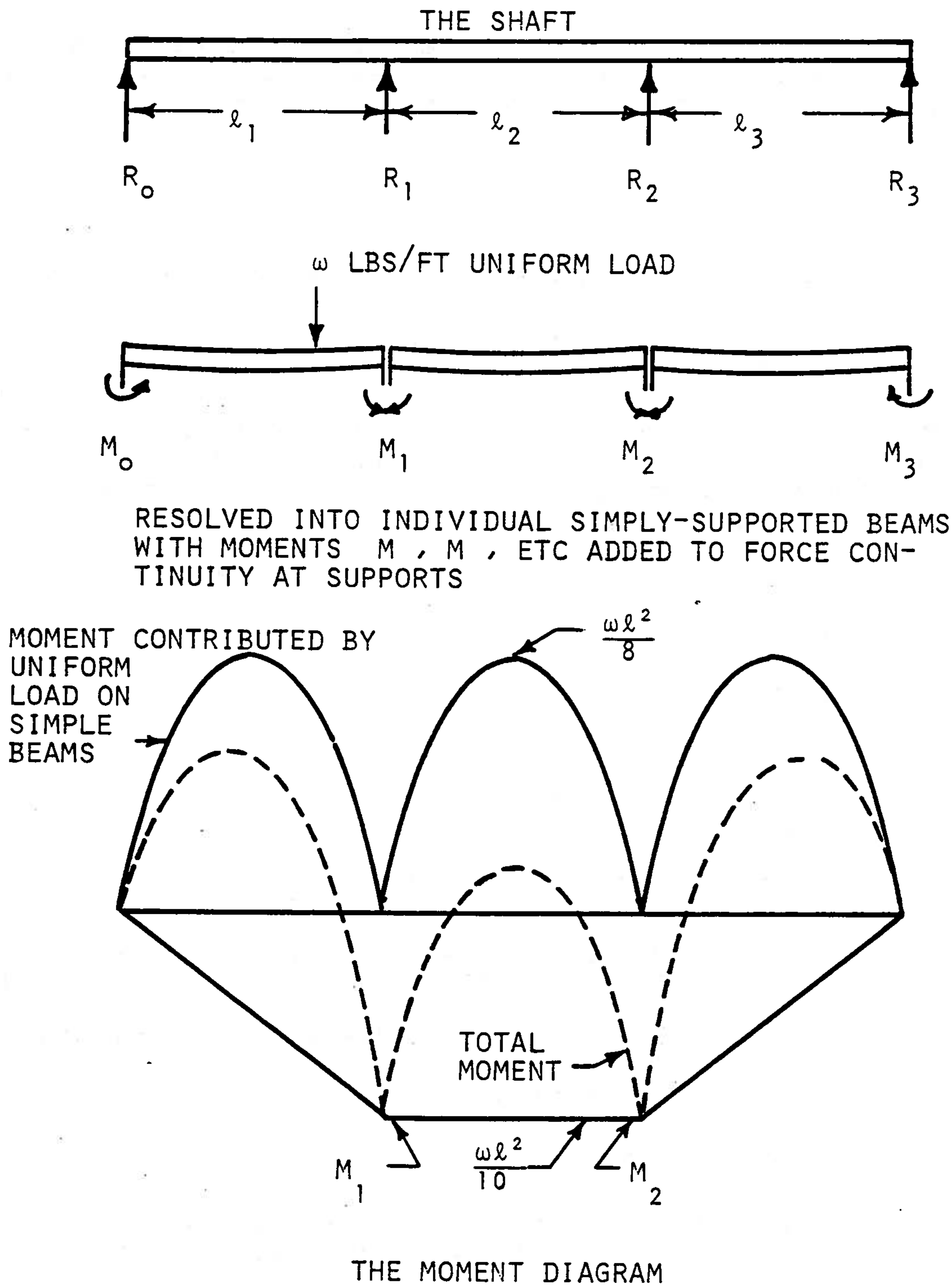


FIGURE 9.7 Illustrating the Method of Three Moments

where R_n' = the reaction due to the individual spans considered simply supported.

As an example, consider a uniform shaft having three equal spans, as shown in the lower part of Figure 9.7. Supports are in line. At support 1, by equation (9.9)

$$M_1 = -\frac{1}{4_\ell} \left[\frac{w^3}{2^\ell} + M_2 \ell\right]$$

Since $M_1 = M_2$ in this example, because of symmetry,

$$M_1 = M_2 = \frac{w\ell^2}{10} = -\frac{\pi\rho d^2 \ell^2}{40}$$

The figure shows this moment plotted along with the moment for the simply-supported, uniformly-loaded spans. The total moment diagram is sum of these two. Maximum moment occurs at the supports and is $-w\ell^2/10$.

Bending stress is $M\,d/2I$ and is therefore $-4/5\ \rho\ell^2/d$ (at 1 and 2) after w and I are written in terms of d.

Bearing reactions are

$$R_0 = \frac{w\ell}{\ell/2} + \frac{M_1}{\ell} = \frac{\rho\pi d^2 \ell}{10} = R_3$$

$$R_1 = w\ell - \frac{M_1}{\ell} = \frac{11}{40}\ \rho\pi d^2 \ell = R_2$$

As further example, assume a bearing span of $\ell = 40d$, and $\rho = 0.28$ lbs/in^3. Then

$$\sigma = -1280\ d \text{ (at supports 1,2) } (\sigma = \text{bending stress})$$

$$R_0 = R_3 = 3.56\ d^3$$

$$R_1 = R_2 = 9.79\ d^3$$

A quick substitution for d is of interest, giving, for instance (negative sign dropped from stress):

d inches	σ lb/in^2	R_0, R_3 lbs	R_1, R_2 lbs
3 1/2	4,480	153	421
6	7,680	770	2,110
24	30,700	7,330	20,150

Now, an appropriate maximum value of cyclic stress in steel shafting not exposed to sea water corrosion might be 9000 lbs/in^2, [7] irrespective of diameter. The shaft in Figure 9.6 corresponds to the first one above, i.e., 3-1/2 inch diameter, 40 diameters span, and apparently is quite safe from bending stress (neglecting any vibratory contribution to cyclic stress). Not so the 24-inch shaft (its bearings would need to be much closer than 40 diameters). The general message is that smaller craft have less concern with shaft bending.

Moving a support out of direct line with the others can be either good or bad. Let's illustrate again with the uniform shaft of three equal spans, and also assume that supports at 1 and 2 are moved up or down the same distance. Still $M_1 = M_2$ because symmetry is preserved. Equation (9.9) becomes

$$M_1 = M_2 = -6EI \frac{\Delta}{I} - \frac{\pi \rho d^2 \ell^2}{40}$$

Again use $\ell = 40d$, and put in terms of stress at supports 1,2:

$$\sigma = \frac{3E\Delta}{40} - 1280\ d = -2.25 \times 10^6 \Delta - 1280\ d$$

$$R_0 = R_3 = 3.56\ d^3 - \frac{3}{1280}\pi E \Delta d^3 = d^3(3.56 - 0.221 \times 10^6 \Delta)$$

$$R_1 = R_2 = 9.79\ d^3 + \frac{3}{1280}\pi E \Delta d^3 = d^3(3.56 + 0.221 \times 10^6 \Delta)$$

It is obvious that pushing these supports up (Δ positive) increases the stress at the supports, and transfers load to the raised supports.

It is also obvious that pushing the supports down relieves the bending stress at the supports, and indeed, if $\Delta = -1280 \times d/2.25 \times 10^6$ for the example above, this stress is zero. It will, however, increase bending stress at other points, such as span midpoints, in particular. A glance at Figure 9.7 shows why. The minimum stress condition occurs when the stress supports 1,2 equals stress at the midpoints of the other spans, or in terms of moments

$$\frac{1}{2} M_1 + \frac{w\ell^2}{g} = -M_1$$

Now consider the effects of an overhung weight (i.e., propeller) on the problem of lining up the bearings. Assume that the equal-span shaft in Figure 9.7 is extended to the left a distance a, and a weight W is hung on the end

$$M_0 = -Wa + \frac{wa^2}{2}$$

$$M_1 = -\frac{1}{4\ell}\left[\frac{w\ell^3}{2} + M_0\ell + M_2\ell\right]$$

$$M_2 = -\frac{1}{4\ell}\left[\frac{w\ell^3}{2} + M_1\right]$$

$$M_3 = 0$$

Putting w in terms of d, and making the indicated substitutions, reduces these to

$$M_0 = -a[W + \frac{\pi}{8}\rho d^2 a]$$

$$M_1 = -\frac{4}{15}[\frac{\pi\rho d^2}{32}(3\ell^2 - 4a^2) - aW]$$

$$M_2 = -\frac{1}{4}[\frac{\pi\rho d^2}{30}(3\ell^2 - 4a^2) + \frac{4}{15}aW]$$

To simplify further discussion, let $\ell = 40d$ and $a = 1/2$.

$$M_0 = -20d\ [W + \frac{5}{2}\pi\rho d^3]$$

$$M_1 = -\frac{4}{15}\cdot[100\pi\rho d^4 - 20dW]$$

$$M_2 = -\frac{1}{4}\ [\frac{320}{3}\pi\rho d^4 + \frac{16}{3}dW]$$

The overhung weight tends to lift the shaft off all bearings except 0, and in particular, lightens the load on bearing 1. So let's check reaction at 1.

$$R_1 = w + \frac{M_0 - M_1}{\ell} + \frac{-M_1 + M_2}{\ell}$$

The appropriate substitutions give

$$R_1 = \frac{113}{120}\rho\ d^3 - \frac{11}{15}\ W$$

Thus the reaction would be zero if $W = (113\rho d^3)/88$, and the shaft would be lifted clear (as far as top half of the bearing allows) for greater values of W.

A propeller is never overhung in the extreme manner of this example, but still is always overhung to some extent, and so there is a tendency for the shaft to lift off the second bearing inboard, or off the forward edge of the stern tube bearing. This means that other bearings are overloaded, and that the span is longer than intended, hence possibly allows whipping of the shaft. Here then, is incentive for raising the unloaded bearing, or for lowering the after one, which usually implies boring the stern tube at a slight angle downward from the shaft line.

9.6 SHAFT ALIGNMENT

The entire shafting system must be aligned so that bending stresses are minimized, so that bearings are loaded equitably, and so that reduction gear mesh is not distorted. The simplest alignment is, of course, the ruler-straight line. There are two possible difficulties with this scheme, however.

First, several examples above have shown that minimum bending stress and equitable bearing loadings do not come from straight-line alignment, but from raising or lowering certain bearings. As you may have deduced from the examples, the best condition is obtained when the supports are aligned to follow something like the "natural bent" of the shaft. Such a notion is the basis of the "fair-curve" alignment practice now widely followed.[13]

Second, thermal expansion of the reduction gear can lift it with respect to the nearest shaft bearing, causing not only bearing overload and shaft bending stress, but distortion of the gear mesh. The main remedy for this is to set the gear low by the amount it will expand in operation. A further remedy is to allow flexibility in the shaft, i.e., don't locate a shaft bearing close to the gear but back off as far as practicable to allow the shaft to follow gear-coupling movements. In any case, the general idea is to get approximately equal loading on both bearings within the gear housing, since this assures good mesh. See Reference 14 for complete discussion.

Now, the question of small vessels versus large ones in alignment practice. It has been shown that, for a given ℓ/d bending stress is proportional to diameter, so that there is less incentive to attempt to relieve it. The question of gear case thermal growth is not specifically analyzed here, but it can be shown that the shaft stress and the resulting load on the gear are of lesser magnitude for small power plants. The general conclusion is that the ruler-straight alignment is probably satisfactory for propulsion plants of, say, a few hundreds of horsepower per shaft, while careful offsetting of bearings and gear cases is necessary in large ships. The dividing line is not well established. If in doubt, a designer can and should make a careful analysis of shaft stresses and bearing loads in the expected operating condition. Formulas can be found in this chapter, with some help from Reference 4. However in practice the job is quite laborious, especially if there are many bearings, and if the shaft is not of uniform size. A digital computer is greatly to be desired. Programs are in existence; see Reference 8 for example.

It is well known that hulls deflect from a variety of causes. One of the causes is the change in bottom loading experienced as the vessel moves afloat from dry dock or launching ways. Thus it is that alignment should be performed afloat. Deflections of the hull in service might also seem worrisome, but practice has shown that they do not noticeably disturb an originally well-aligned shaft. The shaft is usually so much more flexible than the hull that it can take any deflection that the hull can.

The actual process of alignment for the small craft is usually one of aligning the coupling that joins propeller shaft to reduction gear output flange, and ideally consists of reducing offset and angular misalignment to zero. See Figure 9.8 for illustration. Since perfection is not to be expected, engine builders publish tolerances for their engines. Caterpillar, for example, [10] advises that the flange faces should not be out of parallel by more than 0.0005 inches per inch of flange diameter, measured across the outer edges of the flange.

Engines typically have leveling screws to facilitate aligning to the shaft coupoing. They are apparent on the Mercury spark-ignition engine shown in Figure 4.3. In some instances, these screws are intended for permanent support of the engine; in others it is intended that after alignment liners be fitted between the engine and the foundation to provide the actual in-service support of the engine.

9.7 TORSIONAL VIBRATION

Torsional vibration is a potential hazard to any rotating shaft system to which torque is applied in pulses, as it is when the shaft is driven by a reciprocating engine (the propeller is also a source of such excitation, as mentioned later). The possibility of such a vibration existing can be seen from consideration of two massive discs at opposite ends of an elastic shaft, as in Figure 9.9. This assembly might be an idealized picture of a propeller, engine flywheel, and their connecting shaft. If the discs are twisted in opposite directions, then released, a torsional vibration occurs because of the cyclic interchange between kinetic energy of the rotating discs and elastic energy of the shaft. The vibration continues until the energy put in by the initial twist is dissipated by friction. The frequency of the vibration is given by the following equation:

$$F = 2\pi\ k\ \frac{(J_1 + J_2)^{1/2}}{J_1 J_2} \quad \text{cycles/sec} \tag{9.12}$$

where J_1, J_2 = polar moments of inertia of the two discs

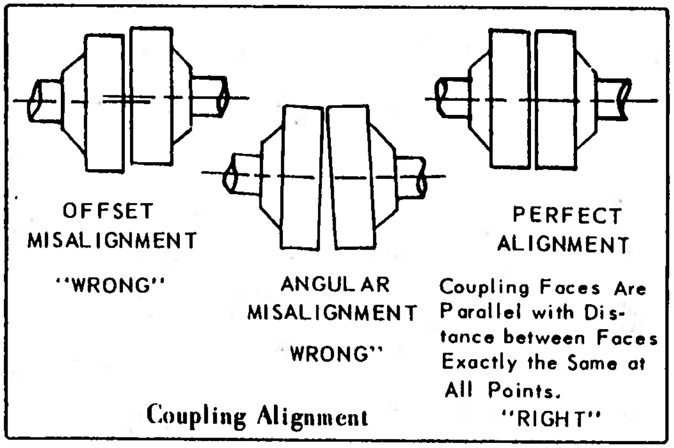
k = torsional spring constant of the shaft

There will be a node, or point between the discs where relative motion is zero, as indicated in the figure.

If input energy is repeatedly applied, as from the power strokes of a piston connected to the shaft, the vibration continues at the frequency of the input. If this frequency is the same as that given by equation (9.12), a condition of resonance is said to exist, and the angle through which the discs vibrate is very large. Torsional stress in the shaft is proportional to this angle; consequently resonance can cause failure of the shaft.

The analysis of a shaft system for its frequency is usually done by analyzing a disc-shaft combination as in Figure 9.9. For reasonably accurate results with an engine-propeller system, however, it is usually necessary to have more than two discs, because each piston-connecting rod assembly has inertia that makes it contribute to the phenomenon as a disc would. The system might thus be represented by a shaft with a disc for each cylinder, for the flywheel, for the reduction gear, and for the propeller. Of course, some of these are not actually discs at all, so a fair part of the analysis goes into finding the inertia of an equivalent disc that has the same inertia as the actual non-disc parts. Finding the spring constant of a straight shaft equivalent to a crank shaft is also an entertaining exercise. Both of these can be accomplished with fair success, nonetheless. See Reference 11 for details.

When more than two discs are present, the finding of frequency is no longer a matter of simple formula such as equation 9.12. The methods are detailed in most textbooks on mechanical vibration, such as [11]. A significant result that must be mentioned here is that the analysis predicts more than a single frequency, in fact, it predicts an additional frequency for each degree of freedom (each additional disc-plus-spring) added to the system. Each is a possible natural frequency of torsional vibration, and is traditionally designated by the term mode, i.e., the lowest frequency is mode I, or the first mode, the next highest is mode II, or the second mode, etc. The analysis also predicts the relative amplitude of vibration at each disc. When plotted, these amplitudes show the nodes associated with each mode. The first mode will show one node, the second mode will show two nodes, etc., so that the terms "one-noded mode, two-noded mode,. . ." are sometimes used to identify the modes.



The engine must be aligned to the propeller shaft within .003" (.076mm) or less when measured between the mating surfaces of the transmission coupling flange and the propeller shaft coupling flange. To obtain correct engine alignment, insert a feeler gauge between the coupling mating faces and adjust engine position as required to place the mating surfaces parallel to each other within .003" (076mm).

FIGURE 9.8 Coupling Alignment as Advised for Kiekhaefer-Mercury Engines

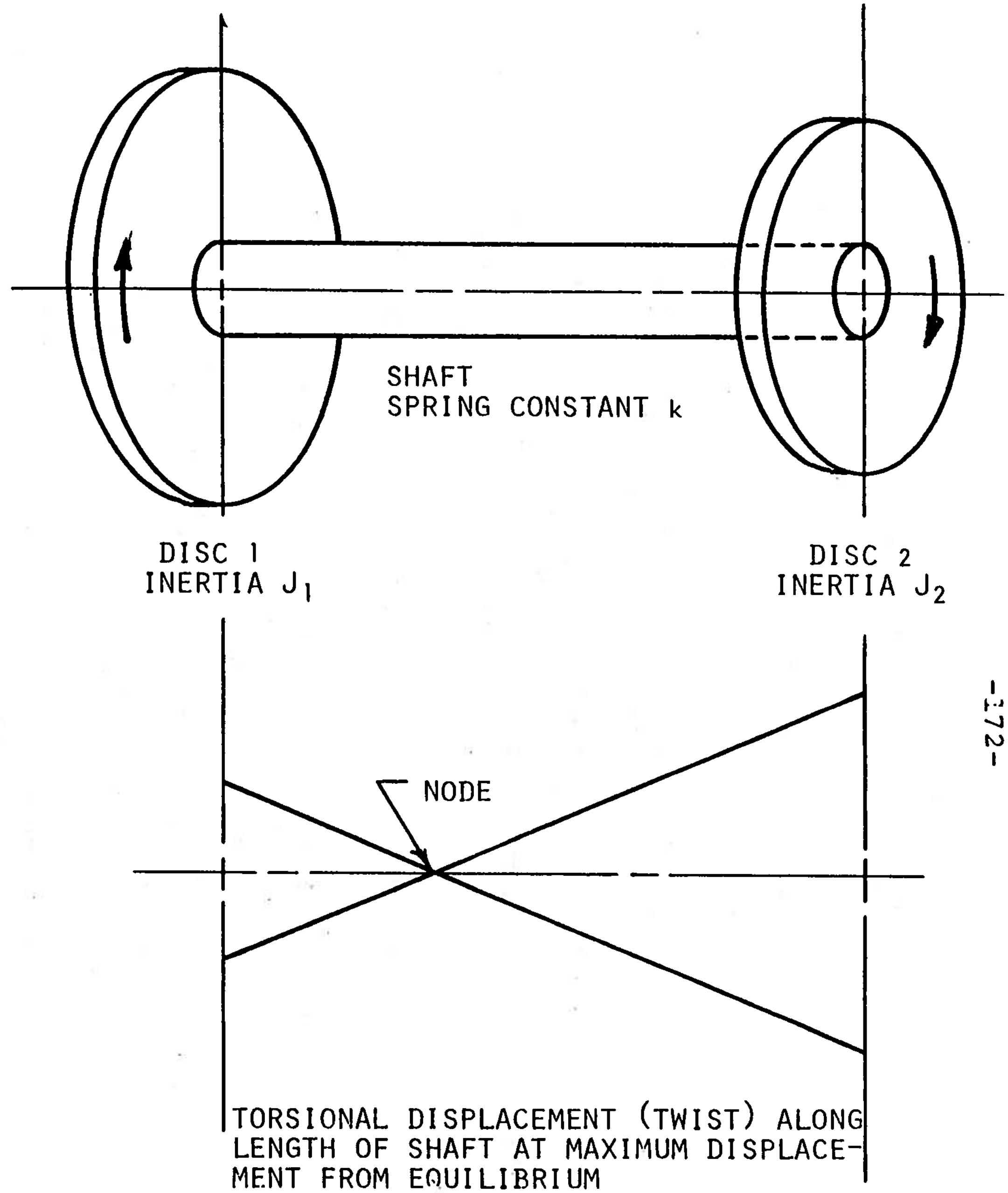


FIGURE 9.9 Disc-Shaft Combination to Illustrate Torsional Vibration

Figure 9.10 shows a plot of nodes for the first three modes from a discussion in Reference 12. Note that the first mode produces a large twist in the propeller shaft, and relatively little twist between engine cylinders. If the vibration is severe enough, the result is thus likely to be a broken propeller shaft. Also, the crankshaft vibrates as a unit relative to the propeller, so that rough running of engine accessories may also result. On the other hand, danger to the crankshaft arises from the higher modes, which show neighboring portions vibrating against each other.

All, or none, of the modes of torsional vibration may exist in a rotating shaft, depending on the presence or absence of the necessary excitation. The excitation is a periodic torque input having a frequency resonating (coinciding) with one of the natural frequencies. The propeller is a source of such excitation; as each blade passes near the hull it enters and leaves the hull boundary layer, and this momentarily changed condition is reflected as a torque jog in the shaft. With a reciprocating engine, however, the more important excitation comes from the engine, originating in its highly irregular torque curve. Figure 9.11 shows a typical torque variation for one cylinder of a four-cycle engine. The torque is high during the power stroke, falls quite low during exhaust and input strokes, then is moderately high again (negatively) during the compression stroke. This curve is periodic, being repeated with each cycle of the cylinder. Unfortunately, it is not a pure sinusoid, and so must be a mixture of many sinusoidal harmonics. Figure 9.11 also includes sketches of several of these. Note the term order used to designate them; order refers to the number of cycles completed by the harmonic during one revolution of the crankshaft. (The two-cycle engine can have only integral orders, since the first harmonic must complete in one revolution, rather than two.)

There are thus many natural frequencies, or modes, and many torque harmonics that can resonate with them, so that it appears virtually impossible to avoid some conditions of resonance, and hence some torsional vibration with a reciprocaring IC engine. Fortunately, all of these resonances are not necessarily significant. The ultimate consideration is usually one of stress, and some of the resonances produce negligible stress. The higher modes generally have lower amplitudes of vibration, and the higher orders likewise have lower amplitudes of torque, so that the resonance between a high mode and a high order is usually of little consequence. Also, some of the orders will be cancelled, or at least much reduced in magnitude, by the differences in phase among the cylinders of the multi-cylinder engine. Figure 9.12, from Reference 12, illustrates this with a typical example. Note that by far the largest stress is produced by interaction of order 4 with mode I (orders in Arabic numerals, modes in Roman). A designer would want to do something about this one, but could probably safely ignore the others shown in the figure.

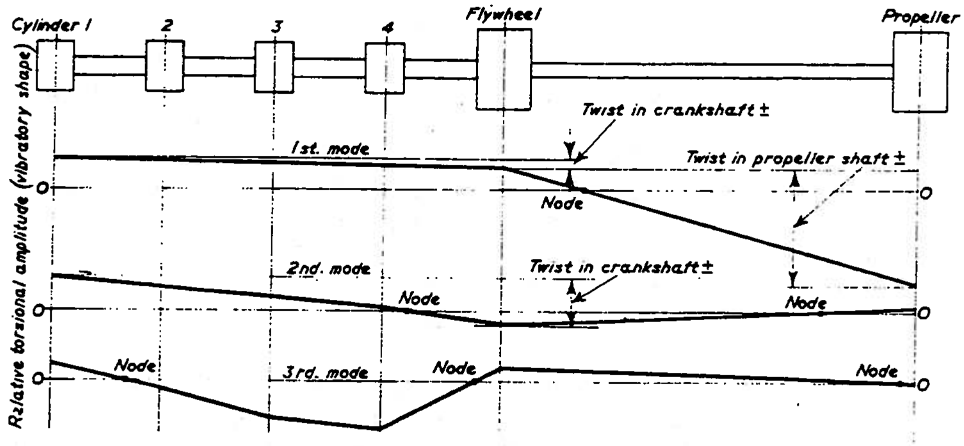


FIGURE 9.10 Torsional Displacements for Six-Disc System (Four-Cyliner Engine, Flywheel, Propeller) (Reference[12])

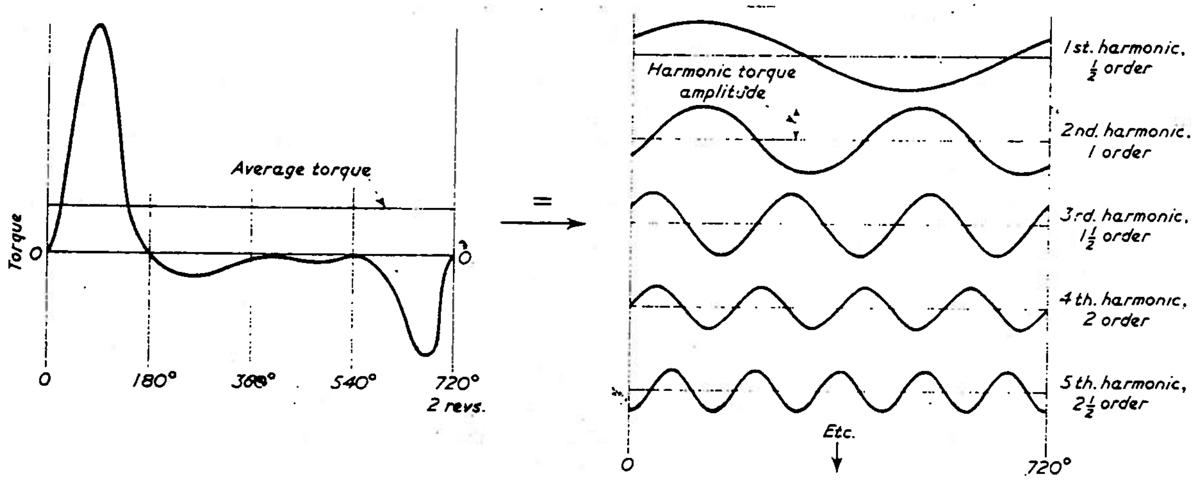


FIGURE 9.11 Torque Curve for Four-Stroke Engine Cylinder and First Five Harmonic Components (Reference[12])

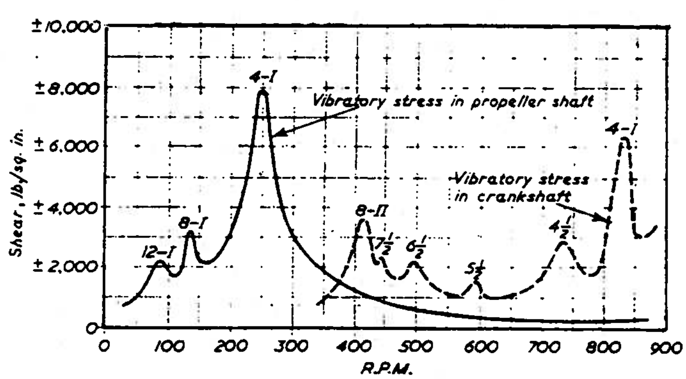


FIGURE 9.12 Typical Torsional Stress vs RPM, showing Several Resonances (Reference[12])

The analysis of interactions of engine torque harmonics and the vibration modes for the purpose of deciding which are significant is another task that is a little too complex for a treatment here. You can get the story from Reference 11.

Cures for Torsional Vibration

Now suppose that a high-stress resonance between engine torque harmonic and vibration mode is found, either in design analysis, or in service; what do you do about it? There is no simple answer, since there are a number of things that can be done, and the choice will have to be made in view of the overall power plant situation. Some of the techniques that are frequently used are:

(1) The barred speed range. RPM's near the resonance forbidden, perhaps by simply painting a nasty red line on the tachometer at the appropriate spot.

(2) "Tuning" the vibrating system to a different frequency. For example, increasing shaft diameter raises its spring constant, and so increases natural frequency. Usually, load-carrying considerations forbid reduction in shaft diameter, but the same effect on frequency (lowering) can be gotten by use of flexible couplings. Going a step further, fluid couplings are sometimes used, particularly between engine and reduction gear pinion. These devices pass so little of the vibratory torque that they essentially separate the system into two systems.

(3) Adding damping. Damping is the mechanism by which the surroundings (water around the propeller, bearing oil films, etc.) absorb the vibratory energy. The greater this absorption, the smaller the vibratory amplitude, and hence the lower the stress magnitude. Torsional dampers are usually added to the engine internally, but sometimes the propeller can be changed to increase its contribution to damping. Reference 12 cites a case in which propeller damping was increased by changing to a smaller propeller. The new propeller had a lower inertia, and so tended to have a greater amplitude of vibration. Since the damping energy transfer is proportional to the square of the amplitude, the net result was a smaller amplitude, and hence a lower shaft stress.

9.8 LATERAL VIBRATION

Lateral vibration, or whirling, of the shaft occurs if the length of shaft between bearings is such that its natural frequency as a beam vibrating laterally equals the rotational speed. It doesn't depend on external excitation, as from engine or propeller, but is self-excited, i.e., is excited by small eccentricities of mass in the shaft itself, or by some other inherent non-uniformity that can exert a cyclic force as the shaft rotates.

Analysis of whirling frequency is usually based on the standard analysis for a slender beam vibrating laterally. See Reference 13 for details; the result is

$$w_n = \frac{C}{\ell^2}\sqrt{\frac{gEI}{w}} \quad \text{radians/sec} \tag{9.13}$$

where g = acceleration of gravity

E = modulus of elasticity

I = plane moment of inertia about diameter

w = weight per unit length

ℓ = length of span

C = a constant whose value depends degree of fixity of ends

For a simply supported steel shaft, the frequencies in cycles/minute for the first modes are (value of C taken from Reference 13, numerical values put in, I and w written in terms of shaft diameter, etc.)

$$F_1 = 4.78 \times 10^6 \frac{d}{\ell^2} \text{ cpm} \tag{9.14}$$

$$F_2 = 19.2 \times 10^6 \frac{d}{\ell^2} \tag{9.15}$$

$$F_3 = 43.2 \times 10^6 \frac{d}{\ell^2} \tag{9.16}$$

for diameter d and span length ℓ in inches.

If this is to be used on an actual shaft, the user must decide if the continuous shaft supported by more than two bearings, perhaps, can be represented by the simply-supported model. For example, if a beam is clamped at the ends, its frequencies are approximately twice those given by equation

(9.14) et al. The condition of continuity with the next span adds some fixity, but assumption of simple support seems to be closest approximation. It is recommended by Reference 13, for example. Another difficulty is that the frequency of all spans must be the same, so that if spans are of unequal length or unequal diameter, the frequency found by consideration of the individual span will be incorrect. Nonetheless, considerable guidance can be obtained from the simple approach; let's illustrate with the boat pictured in Figure 9.6.

The shaft in Figure 9.6 is 3-1/2 inches diameter throughout its speed at continuous rated power of the engine is 436 RPM, and the longest span is about 120 inches. Equation (9.14) gives a frequency of 1160 cpm. The other spans are shorter, so that if the frequency could be calculated for the shaft as a whole, it would doubtless be greater than 1160. Since even this figure is more than twice the rated shaft RPM, whirling vibration does not appear likely in this example.

9.9 FLEXIBLE COUPLINGS

There are many types of flexible couplings--too many for a listing and description to be attempted here. A simple form might be a rubber disk clamped between two shaft flanges, and arranged so that all torque must be transmitted by the rubber rather than via coupling bolts or other comparatively rigid members. The rubber would transmit the torque, but with an angle of twist that would equal that of a long piece of steel shafting. This, indeed, is the purpose of the coupling--to obtain the flexibility of a long shaft within a limited space. The effect on torsional vibration, that of lowering natural frequencies, has been mentioned in the section on shaft vibration.

The flexible coupling also eases the alignment problem if it is flexible in the bending direction. The usual admonition is not to use a flexible coupling in lieu of good alignment, but it is a respectable solution where alignment is difficult to maintain in service. Such cases would exist if thermal expansion is difficult to compensate for or to predict, or if the engine is flexibly mounted and so subject to movement relative to the shaft. Another problem arises if there is axial movement within the propulsion system from any cause. Alignment of shafting cannot take care of this, and so a coupling with axial flexibility is essential.

The universal joint might also be classed as a flexible coupling, and is used, especially in smaller draft, when engine shaft and propeller shaft must be displaced from one another. These joints do have significant friction losses which are a function of the angle between driving and driven shaft. For example, at 30° angle, losses can be as much as 10 percent of the power transmitted. Losses imply friction, and friction implies wear, so occasional replacement may be in order for a vessel in continuous service. Figure 9.13 shows an example of universal joint use.

Hydraulic couplings and several types of electromagnetic couplings can also be put under the heading of flexible couplings. They seem to be rarely used, but sometimes are the solution for severe vibration or shock isolation problems. They can also be used as clutches by draining and filling the hydraulic fluid or by changing the electrical excitation. Reference 14 gives a thorough discussion of the use of these devices, and of their characteristics.

9.10 BELTS AND CHAINS

A belt or chain drive can furnish the speed reduction function of a reduction gear by the difference in diameter between their sheaves. Chain drives were used in some WWII military craft because of the unavailability of gear sets, but haven't been heard from since. If you are interested, check Reference 15.

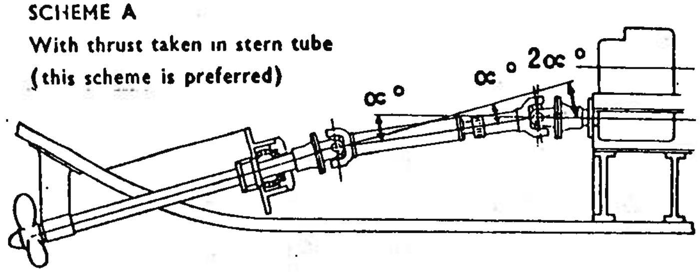
Belt drivers are occasionally used in lieu of reduction bears. The appeal is presumably comparative cheapness. There are problems; however, the load usually requires many belts in parallel, and getting and keeping a uniform load distribution among them can be troublesome. Likewise the need for adjustment for wear, and replacement of failed belts. See Reference 16 for further discussion.

9.11 RIGHT-ANGLE DRIVE

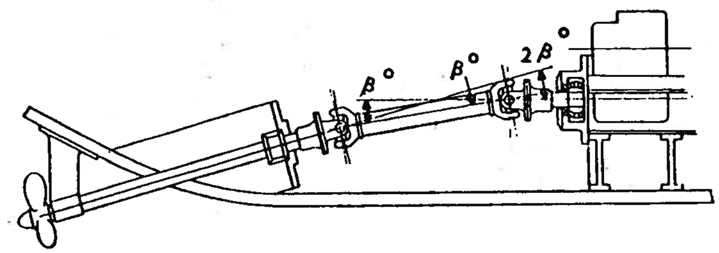
A right-angle drive is pictured in Figure 9.14. This particular unit is one commercially available as a package including CP propeller, gearing, drive line, pitch-actuating mechanism, and housing. Also, it is steerable, and so eliminates the need for a rudder. One advantage of such a unit is that it provides the features of controllable pitch and steering in a package, thus relieving both designer and builder of a small vessel of several individual burdens. If the CP propeller, and high maneuverability at low speeds are not otherwise called for, however, the cost of this unit would seem hard to justify. The resistance of that bulb ahead of the propeller is also significant, although I have no publishable data on this point.

Another possible advantage is the placement of the engine almost directly above the propeller, which means it can be tucked well back in the stern. There are other ways of accomplishing the same thing, however, the inboard-outboard drive and the V-drive (Figure 9.4).

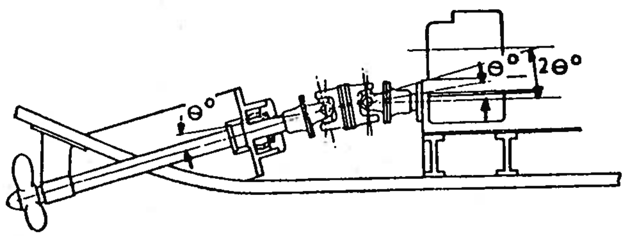
SCHEME A
With thrust taken in stern tube
(this scheme is preferred)



SCHEME B
With thrust taken in gearbox

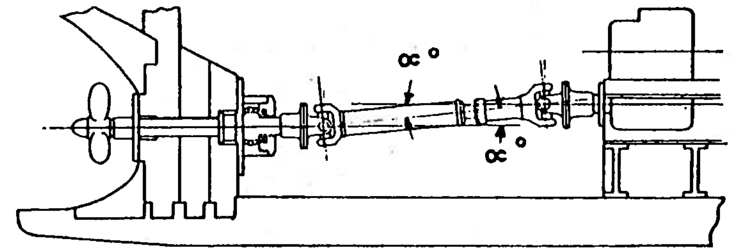


SCHEME C
For use in confined space
Thrust taken either in stern tube or
gearbox (but preferably in stern tube)

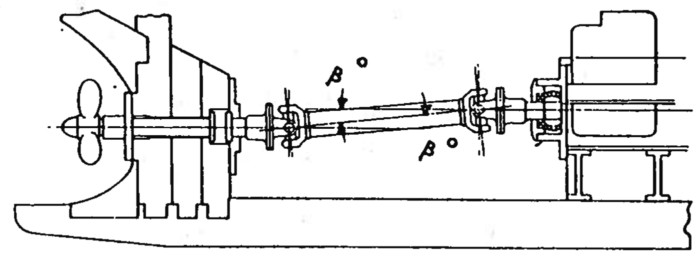


NOTE ! (SCHEME C AND SCHEME F)
When thrust is taken in stern tube
a sliding action should be incorporated
into one of the companion flanges

SCHEME D
With thrust taken in stern tube
(This scheme is preferred)



SCHEME E
With thrust taken in gearbox



SCHEME F
For use in confined space
Thrust taken either in stern tube or
gearbox
(but preferably in stern tube)

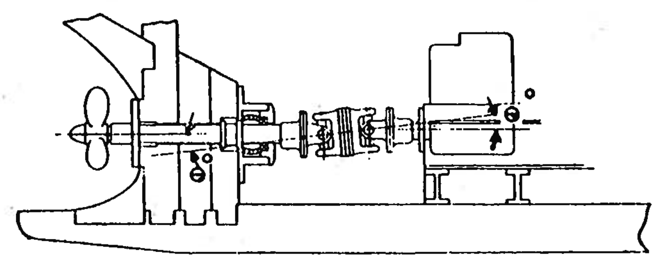


FIGURE 9.13 Applications of Universal Joints in Propulsion Shafts (from application literature of the Perkins Engine Company)

An analysis of right-angle drives is given by Reference 17. It does not include any comparison of this drive with the conventional, and so is of no assistance in making such a choice.

9.12 INBOARD-OUTBOARD DRIVE

Since many thousands of small pleasure boats have been built with this drive, I feel it unnecessary to describe it, nor to give a picture. In fact, it is mentioned here only to head off questions about why I left it out.

Its main virtues are supposedly that it gives to the inboard boat the beachability, steerability, and trailerability of an outboard boat. Doubtless it does, but it's used on boats where these features are not of interest indicates that perhaps the package concept here also has a large appeal to designer and builder. No ruddder, shaft, log, etc. to worry about--just cut a hole in the transom according to the manufacturer's template. If the transom is stout enough to serve as a foundation, all is well.

9.13 REFERENCES

1. Vito Cascia, "A New Marine Transmission for Diesel Engines in the 800-HP Range," ASME Paper 63-GP-9, 1963.

2. Joseph Shigley, Mechanical Engineering Design, McGraw-Hill, 1963.

3. T. A. Gugliuzza and W. H. Hargett, "Gear Design and Laboratory Test Experience--Marine Gas Turbine Propulsion," ASME Paper 69-GT-3, 1969.

4. S. Timoshenko, Strength of Materials, Second Ediction, D. Van Nostrand, 1940.

5. American Boat and Yacht Council, Safety Standards for Small Craft.

6. American Bureau of Shipping, Rules for Building and Classing Steel Vessels, 1970.

7. Sterling A. Fielding, "Design Improvements and Standardization of Propulsion Shafting and Bearings," Marine Technology, April 1966.

8. Graeme Mann, "Design of Propulsion Shaft, Systems, Using Fair Curve Alignment Theory," Naval Engineers Journal, December 1964.

9. H. C. Andersen, and J. J. Zrodowski, "Coordinated Alignment of Line Shaft, Propulsion Gear and Turbines," *Transactions*, Society of Naval Architects and Marine Engineers, 1959.

10. Caterpillar Tractor Company, *Cat Marine Engine Application and Installation Guide*.

11. J. P. den Hartog, *Mechanical Vibrations*, Fourth Edition, McGraw-Hill, 1956.

12. T. W. Spaetgens, "Torsional Vibrations--Some Actual Marine Problems and Solution," *Transaction*, Institute of Marine Engineers, Canadian Division Supplement, No. 11, March 1963.

13. I. K. Mott and R. Fleeting, "Design Aspects of Marine Propulsion Shafting Systems," *Transactions*, Institute of Marine Engineers, Vol. 79, page 177 (1967).

14. Joseph d'Ottavio, "Some Aspects of Indirect Diesel Drives," *International Shipbuilding Progress*, No. 9, 1955.

15. N. C. Bremer, "Heavy Duty Chain Drives for Marine Propulsion," *Proceedings*, ASME Oil and Gas Power Division, 1946.

16. Ullmann Kilgore, "Belt Drives for Marine Propulsion," (unpublished, Professor Woodward has copy).

17. H. M. Chang, J. B. Hadler, H. J. Bernaerts, and John Bader, "Analysis fo Right-Angle Drive Propulsion System," *Transactions*, Diamond Jubilee Meeting, Society of Naval Architects and Marine Engineers, 1968.

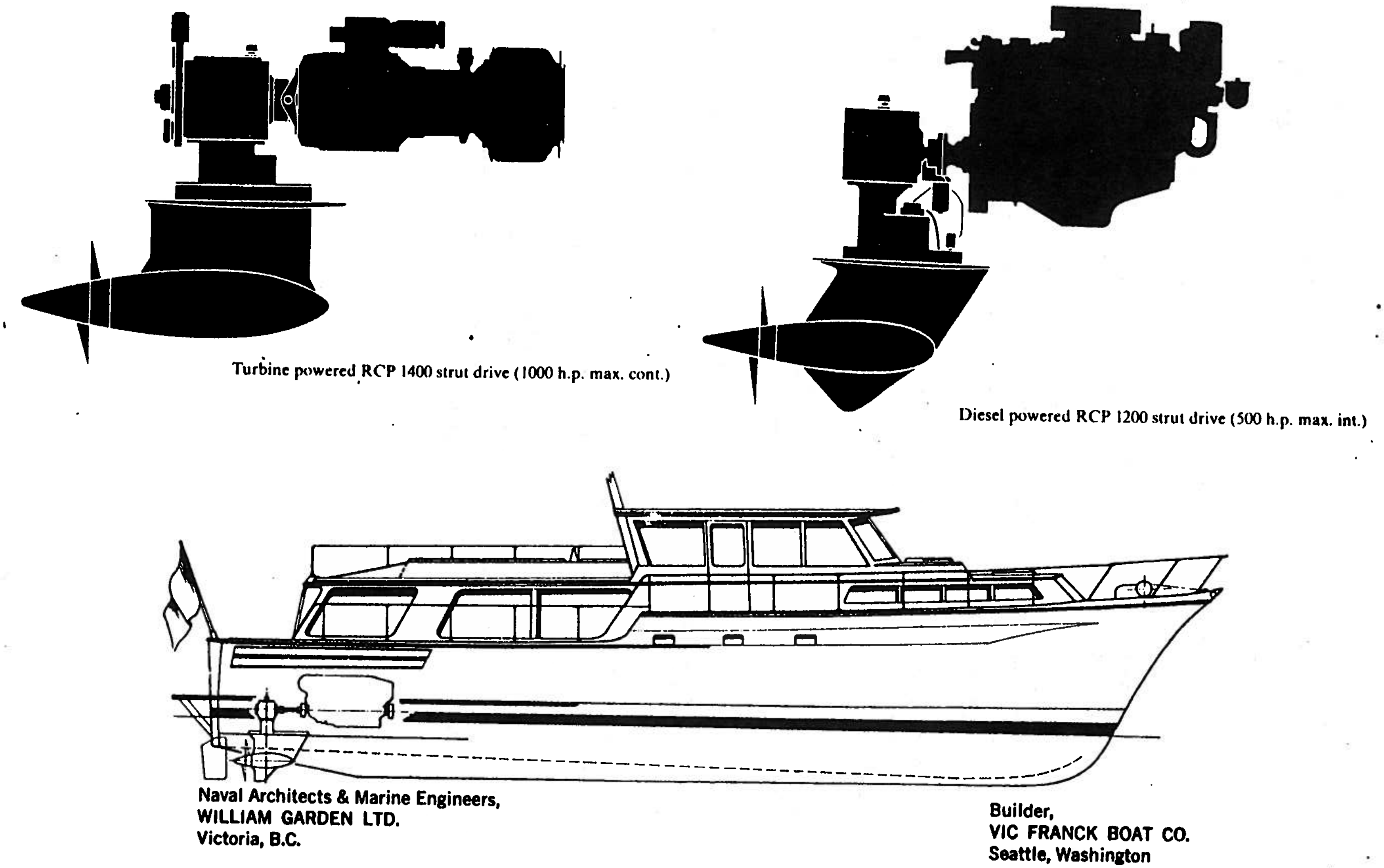


FIGURE 9.14 Illustration of Strut Drive (from literature of Hydro-Drive, Seattle)

CHAPTER 10

AUXILIARY SYSTEMS FOR DIESEL AND SPARK-IGNITION ENGINES

10.1 INTRODUCTION

The propulsion engine must be provided with a happy home; it must be kept at the proper operating temperature, supplied with air, have its exhaust removed, and be supplied with fuel and lube oil. It must be started, its compartment must be ventilated, and it must be supported by an adequate foundation. Such are the usual auxiliary needs of an engine.

This chapter discusses cooling, intake and exhaust, and fuel supply for the diesel and spark-ignition engines. Other auxiliary topics are continued in the following chapter, and special considerations for the gas turbine follow in the chapter after that.

The various auxiliary details for a particular engine are usually specified by the engine builder, e.g., temperatures and flows of cooling water, and these chapters therefore make no attempt at providing handbook-type information for use by designers. Caterpillar, Detroit Diesel, Cummins, and Perkins (these are the ones I am acquainted with; doubtless there are others) publish booklets giving quantitative installation details for their engines. Some of the material here is taken from these sources, but when numbers are quoted, they are intended for illustration only; see the primary sources if you are involved in a particular design. The American Boat and Yacht Council [1], Boating Industry Association [2], and U.S. Coast Guard [3] also publish installation or auxiliary-system information, generally in the guise of safety rules or standards, and they, too, should be consulted as primary sources.

10.2 ENGINE COOLING

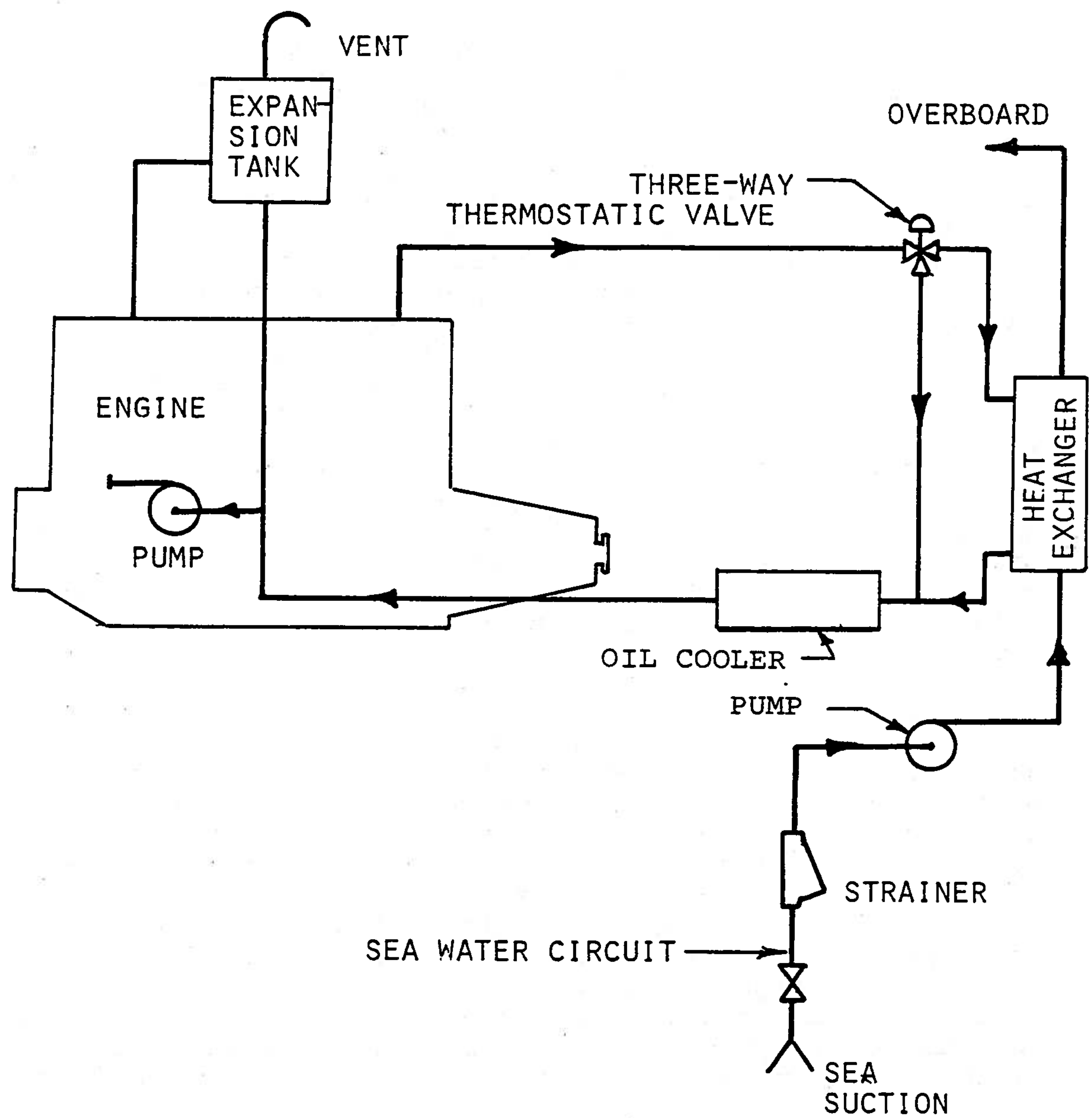
Keep it cool, but not too cool. The cylinder walls and head, as well as the piston crown, must be kept at temperatures well below the peak temperature of the combustion gas, but these comparatively cool surfaces quench the combustion process in their vicinity, lowering combustion efficiency, and possibly condensing corrosive combustion products. On the other hand, if too hot, the lube oil on cylinder walls is destroyed, and in the extreme leads to destruction of the engine through seizure of the pistons. Gruesome! Thus there is a limited range of temperature for optimal conditions within and around the cylinder; to the operator it is manifested as a temperature of the cooling water circulated through the engine cooling passages. Typically this should be in the 180F-200F range for water leaving the engine.

Several types of system are used to transport the engine heat to the sea. The simplest of these is to pipe sea water directly through the engine circuits. In effect, the sea itself replaces the radiator universally found in automotive service. The appeal is simplicity, and hence cheapness, but there are several drawbacks. Most obvious is the corrosive nature of the cooling medium when the vessel is to be used on salt water. Thermostat, drain plugs, and pump impeller must be made of corrosion-resistant alloys, and even so, the life of the engine is likely to be shortened by corrosion of the cooling passages. Another fault is the need to run the engine at a fairly cool temperature because sea salts begin to precipitate from the water at about 140F. This problem is not entirely absent on fresh water--look inside a domestic hot water heater sometime for an idea of the gunk that can drop out of fresh water as it is heated.

When low first cost is not the overriding consideration, indirect cooling via a heat exchanger is preferred. Figure 10.1 depicts such a system schematically, and gives an idea of its features. Engine water is circulated in a closed loop, and chemical additives are therefore feasible to protect against corrosion. Heat is transferred to sea water via a heat exchanger. The heat exchanger and the additional pump are features not found with the simpler direct system, although these components are usually furnished with the engine, and attached thereto. The sea water circulated can be kept well below the temperature of engine water, so that a satisfactory engine temperature and a sea water temperature below 140F are both possible.

Note the termostatic valve, a feature commonly found. It bypasses engine water until it reaches the specified temperature, thus allowing the engine to warm up quickly following startup, and keeping it hot under light loading. Note also the oil cooler. As the figure shows it, the cooled engine water is used to cool the lube oil (oil circuit not shown). A separate sea water cooler for the oil could be used, but the scheme shown is usually preferred because it prevents overcooling of the oil; the cooler does, in fact, serve as a lube oil heater during warm-up. The reduction gear usually has a lube oil cooler also, which may be cooled in similar fashion by the engine water, or may have a separate sea water cooler.

Turbocharged diesel engines often have a cooler for the compressed inlet air (an intercooler or aftercooler), and this, too, can be cooled by the engine water just as the lube oil in Figure 10.1. Alternatively, direct sea cooling of this air can be used, and is preferred in so far as the engine performance is concerned, for the air can be rendered much cooler, therefore more dense, and hence more of it can be forced into the cylinder.



Adapted from P G Tomalin, "Marine Engineering as Applied to Small Vessels," Transactions Society of Naval Architects and Marine Engineers, 1953

FIGURE 10.1 Typical Engine Cooling Circuit, with Heat Exchanger between Engine Water and Sea Water

Another frequently-found feature of an engine cooling circuit is a waste-heat evaporator, a device that uses the engine heat to distill fresh water. These are discussed separately in the next section.

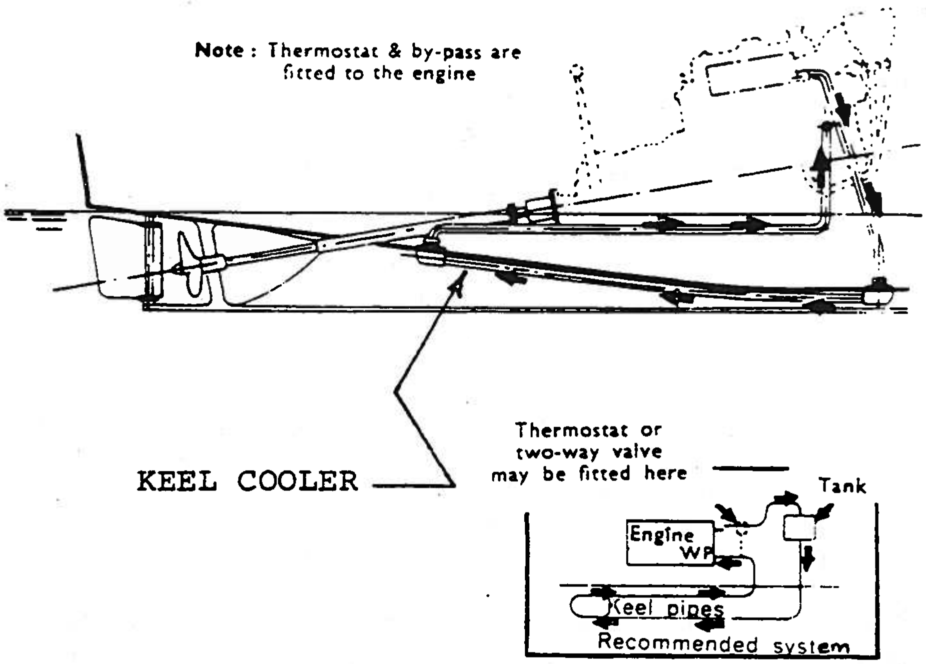
Some of the advantages of direct sea cooling, e.g., the absence of a second circulating water pump, can be had by use of a <u>keel cooler</u> or <u>shell cooler</u>. With the first of these, the heat exchanger for the engine water is placed outside the hull, it consisting of thin-walled tubing exposed to sea water. Heat is thereby exchanged with the sea water through the tube walls. With the shell cooler, which is practical only with a metal hull, engine water is circulated against the inside of the hull, and heat transfer takes place through the shell to the sea water flowing by outside. The keel cooler has the disadvantage of being exposed to damage from grounding or striking submerged objects; it also suffers the lesser hazard of being slathered thickly with anti-fouling paint, a process that adds resistance to heat flow. The shell cooler is no more vulnerable to damage than the hull itself, but must also suffer from whatever coatings are added to the outer hull surface. In general, the area of the shell cooler is much larger than that of the keel cooler because ths shell thickness is usually much greater than that of the keel cooler tubes. Figure 10.2 illustrates both types of cooler.

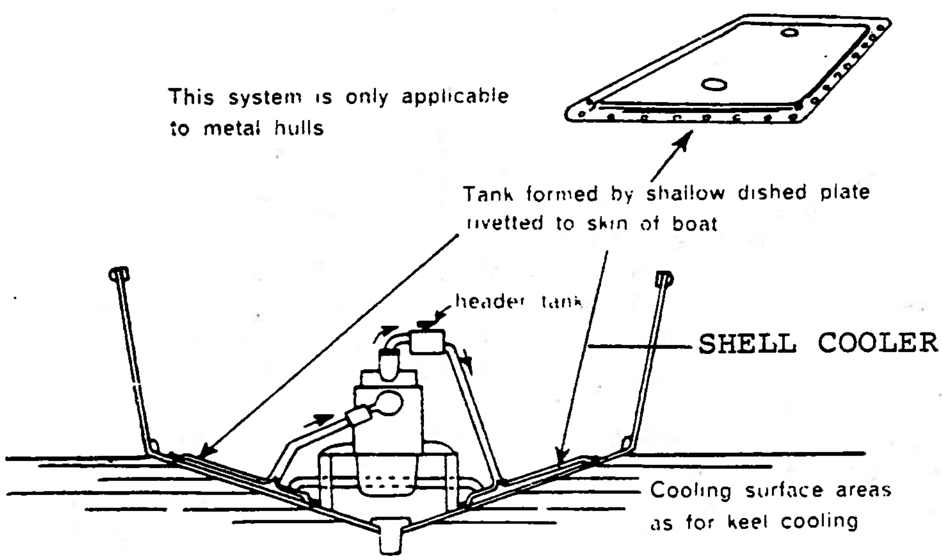
The fundamental question in any heat exchanger design is the area of heat transfer surface to be provided. The question can be approached quite rationally via the elementary principles of heat conduction and convection, as any textbook on heat transfer is likely to explain. This approach is scarcely practicable in the case of the keel cooler or shell cooler because the major resistance to heat flow is the coating (paint, slime, etc.) of unknown characteristics that may build up on the outside surface. The more usual design practice is to rely on experience from past installations, perhaps as advised by the engine builder. An example of typical figures from diesel engine builder [4] is the following set of rules for heat transfer surface:

for stationary vessels (e.g., dredges)	1.0 ft^2 per hp
towboats	0.5 "
free-running, but slow, vessels	0.4 "
fast (above 8 knots) free-running vessels	0.25 "

These surface areas are intended for both keel coolers and sheel coolers.

The amount of heat to be dissipated into the cooling water is sometimes of interest, as for example, when the





from Perkins Engine Company literature

FIGURE 10.2 Illustrations of Keel Cooling and Shell Cooling heat available for distilling sea water in a waste-heat evaporator is being estimated. As a rough rule, between 20 and 30 percent of the heat released by combustion of the fuel usually ends up in the cooling water. The more exact percentage depends on the type of engine and the details of its cooling system. For example, in the two-stroke cycle, some of the heat that would to into the engine cooling water is carried away by the scavenging air. If the exhaust manifold is water-cooled, a higher fraction of heat will appear in the cooling water, and so on.

Table 10.1, taken from Reference 5, is an example of waste heat data as furnished by an engine builder. Figure 10.3 is a heat flow diagram for a turbocharged two-stroke diesel engine with aftercooler and intercooler (i.e., two stages of cooling of the inlet air), giving a graphic idea of the several ultimate destinations of the fuel energy, and also of the interchanges that take place among the various streams along the way. Figure 10.4 shows heat balance information for a four-stroke turbocharged diesel engine, and is of interest because it indicates how the distribution of heat typically changes as the engine load changes from idle speed up to full power.

10.3 USE OF WASTE HEAT

As just seen, a significant percentage of the heat released by combustion of the fuel is wasted by the engine. In diesel-powered ships, it is common practice to make use of the heat in both exhaust and cooling water. The former, which may be from 600F up to perhaps 1200F at full load, is hot enough to produce steam at a reasonably high pressure, and so it is that a "waste heat boiler" is commonly found in the uptake space of a diesel ship. The steam produced can be used for heating domestic hot water, heating the living spaces, heating fuel oil, and running pumps. If sufficient steam can be made, it is often used to drive a steam turbogenerator, and the resultant electric power is used to run the auxiliary machinery as well as lighting, communications, etc. The cooling water, on the other hand, being at low temperature has limited use. It can be used to heat domestic hot water, and to run a distilling plant to produce fresh water from the sea.

The waste heat boiler is rarely found in smaller vessels, because there is usually scant need for the steam, other than electric power production, and small vessels typically do not experience the several thousand hours per year at full load that would make the complication and extra first cost of a steam system economically attractive. Note that an engine-driven generator must be installed anyhow, or a fired boiler in addition to the waste heat boiler, for use when the propulsion engine is not providing sufficient

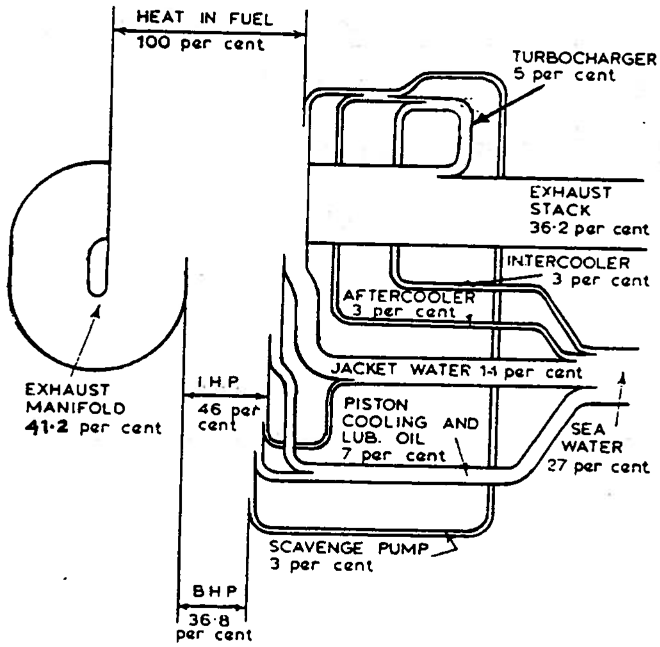
TABLE 10.1 Heat Rejected to Cooling Water by Several Caterpillar Engines (all data from Reference 5)

			Heat rejection Btu/min		
Engine	BHP	RPM	Jacket	After-cooler	Coolant used in aftercooler
D330 NA	85	2000	3200	-	
D342 NA	170	1225	8100	-	
D343 TA	335	1800	12600	-	Jacket water
D343 TA	365	1800	12000	2300	85F water
D399 TA	1300	1300	58500	-	Jacket water
D399 TA	1425	1300	56700	12300	85F water

NA = naturally aspirated

TA = turbocharged and aftercooled

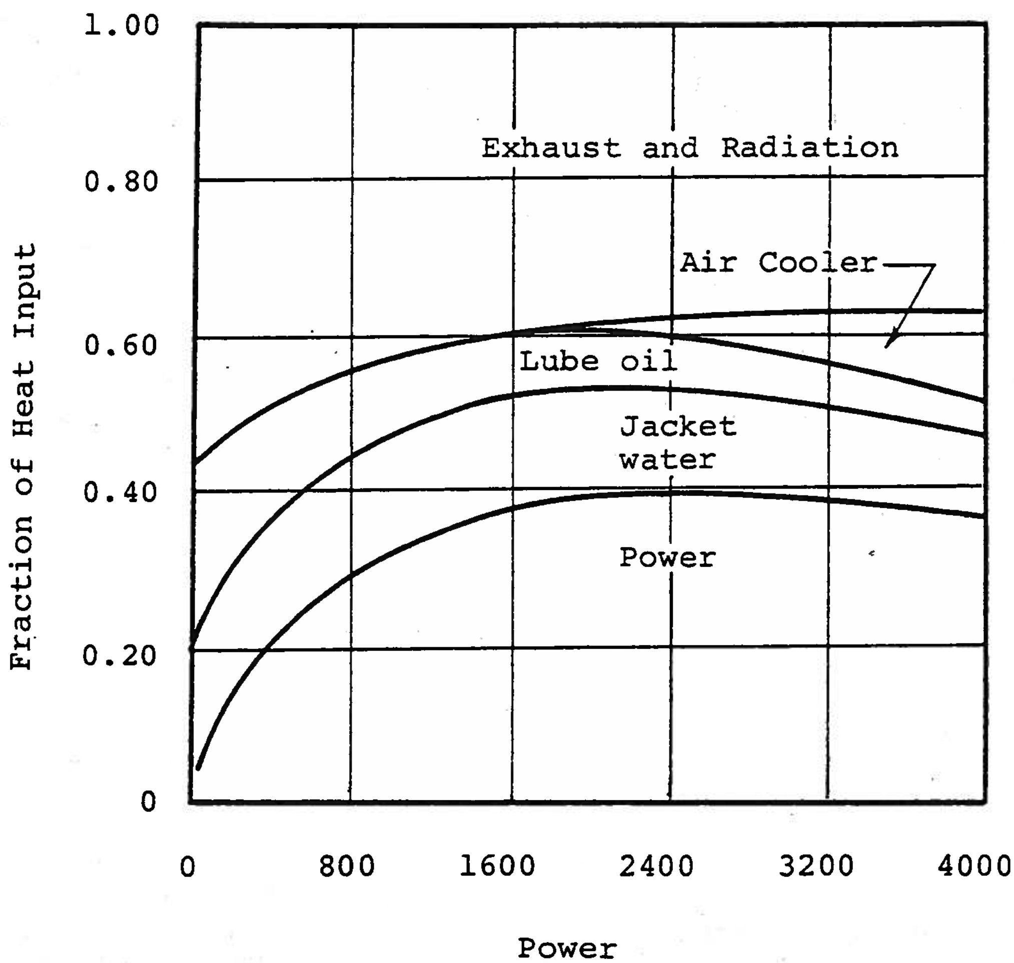
Where engine jacket water is used in the aftercooler, all heat rejected appears in the jacket column.



2. Heat-flow Diagram for a Turbo-charged Engine on Full Load

Reproduced from <u>Shipbuilding and Shipping Record</u>, April 1963

FIGURE 10.3 Heat Balance for a Two-Stroke Turbocharged Engine at Rated Load



from Reference [7]

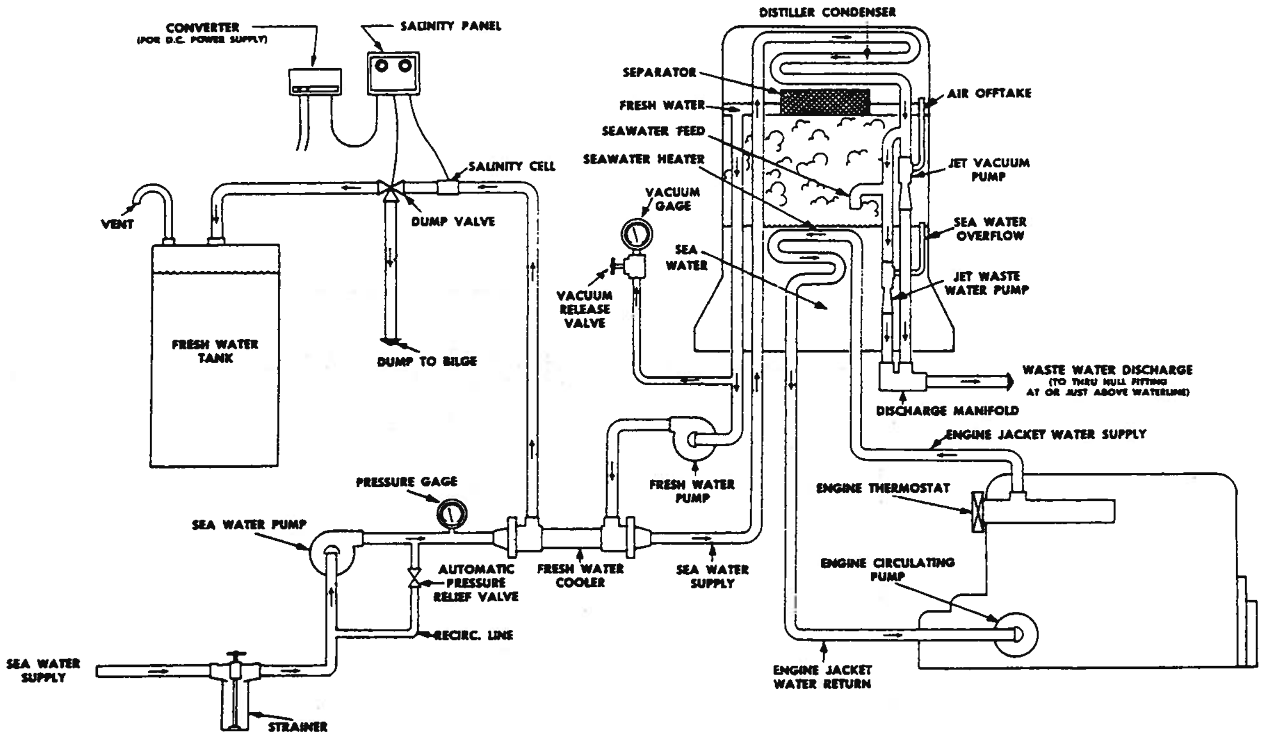
FIGURE 10.4 Heat Balance for a Four-Stroke Turbocharged Diesel Engine exhaust, as in port. For many small vessels, the situation is much the same with respect to use of the engine cooling water to run an evaporator; the fresh water requirements can easily be met by a modest amount carried from shore, making the first cost of an evaporator unattractive. On the other hand, the carrying of fresh water can be a serious burden in some cases, as with a yacht of long cruising range. Thus it is that waste heat distilling plants are occasionally found on small vessels, and appears to be coming more common due to the availability of small stock-model plants in recent years.

Figure 10.5 is a semi-pictorial diagram of a distilling plant made for attachment to an engine cooling water circuit. Boiling takes place at about 115F, the necessary vacuum being drawn by the jet pump shown. The unit is said to produce 10 gal/hr when attached to a 50 hp engine (diesel or spark-ignition) running at full load, to weigh 55 lb, to have length-width-height dimensions of about 15-15-22-and-one-half inches, and to require 0.3 kw for operation of its pumps.

Figure 10.6 is a heat balance for a similar unit (a larger model of the same manufacturer). The numbers represent my calculations, and confirm approximately the claimed production rate; compare the 0.26 gal/hp-hr with the 10 gal/hr from 50 hp mentioned in the preceding paragraph. This ratio should be about the same for all sizes of distilling plant, provided the engine is of comparable size. The ratio is, however, a strong function of engine water temperature. For example, a 15F decrease in water temperature cuts the production rate approximately in half.

Figure 10.7 is a heat balance for a unit that works on the "humidification principle." No vacuum is drawn, the pressure is atmospheric, and the vapor transport to the condensing tubes takes place through the forced circulation of air. Note that the gal/hp-hr is distinctly less than for the vacuum unit just discussed, although the engine water temperature is distinctly higher.

An important consideration in applying a distilling plant is protection of the engine cooling system from disarrangement by operation of the distilling plant circuit. Recall the earlier point that the engine be run neither too hot nor too cool, and both hazards are potentially present. For example, Figure 10.8 shows a piping arrangement suggested by a distilling plant builder. Note that the engine thermostat is bypassed by the distiller so that the engine might run cool without the thermostat being able to do anything about it. Figure 10.9 shows an alternative arrangement that meets this objection, but suppose someone shuts valve C to maximize heat input to the distiller, then forgets to open it when the distiller is not running? Of course, no cooling system is infallible anyhow, that's why temperature gages



The unit shown is the Cuno VJ-5, taken from sales literature of the Cuno Engineering Corp, Meriden, Conn.

FIGURE 10.5 Diagram of a Typical Waste Heat Distilling Plant

VACUUM IS PRODUCED BY A JET PUMP OPERATED BY THE SEA WATER FLOW (NOT SHOWN, SEE FIGURE 10.5)

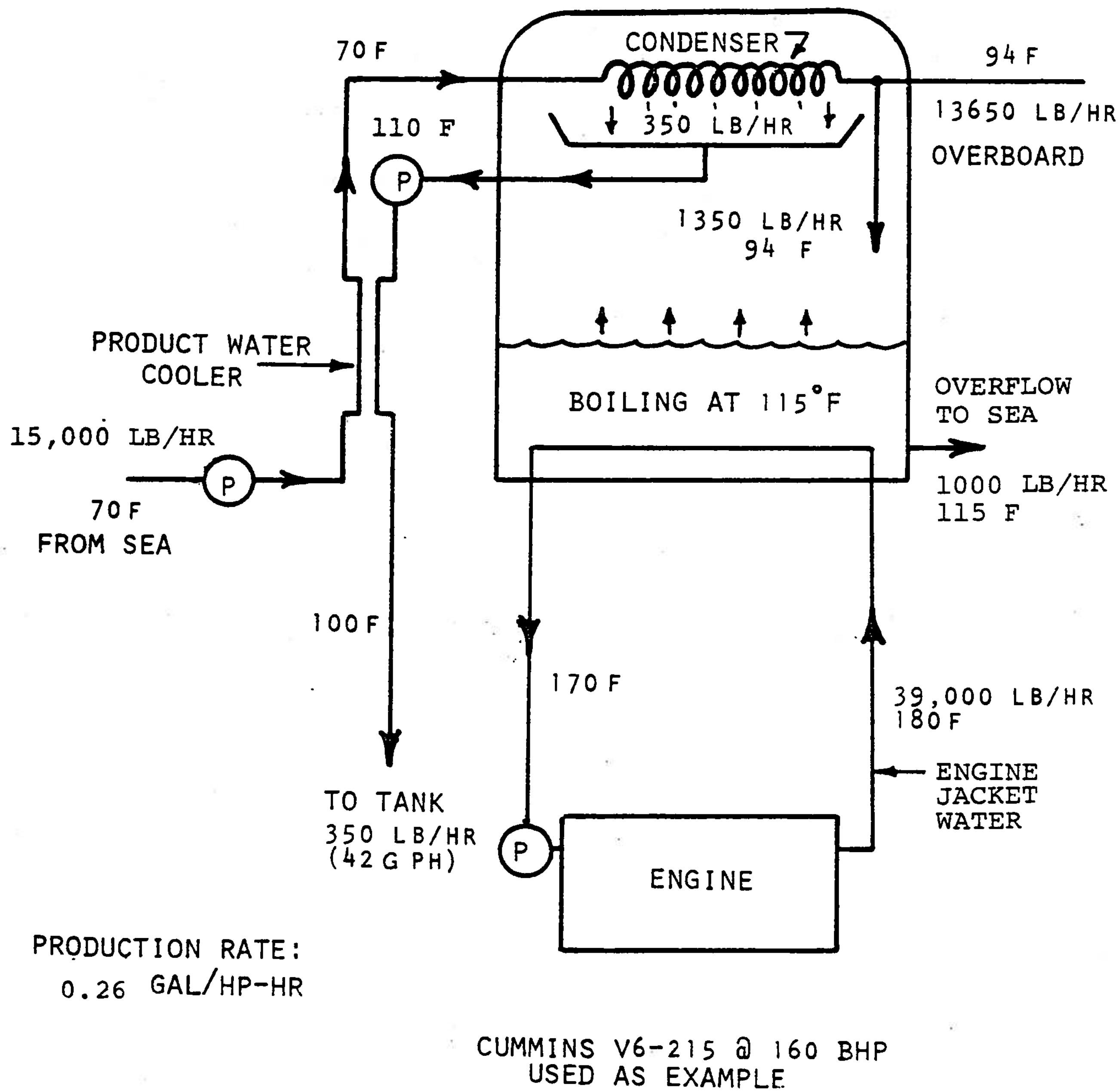
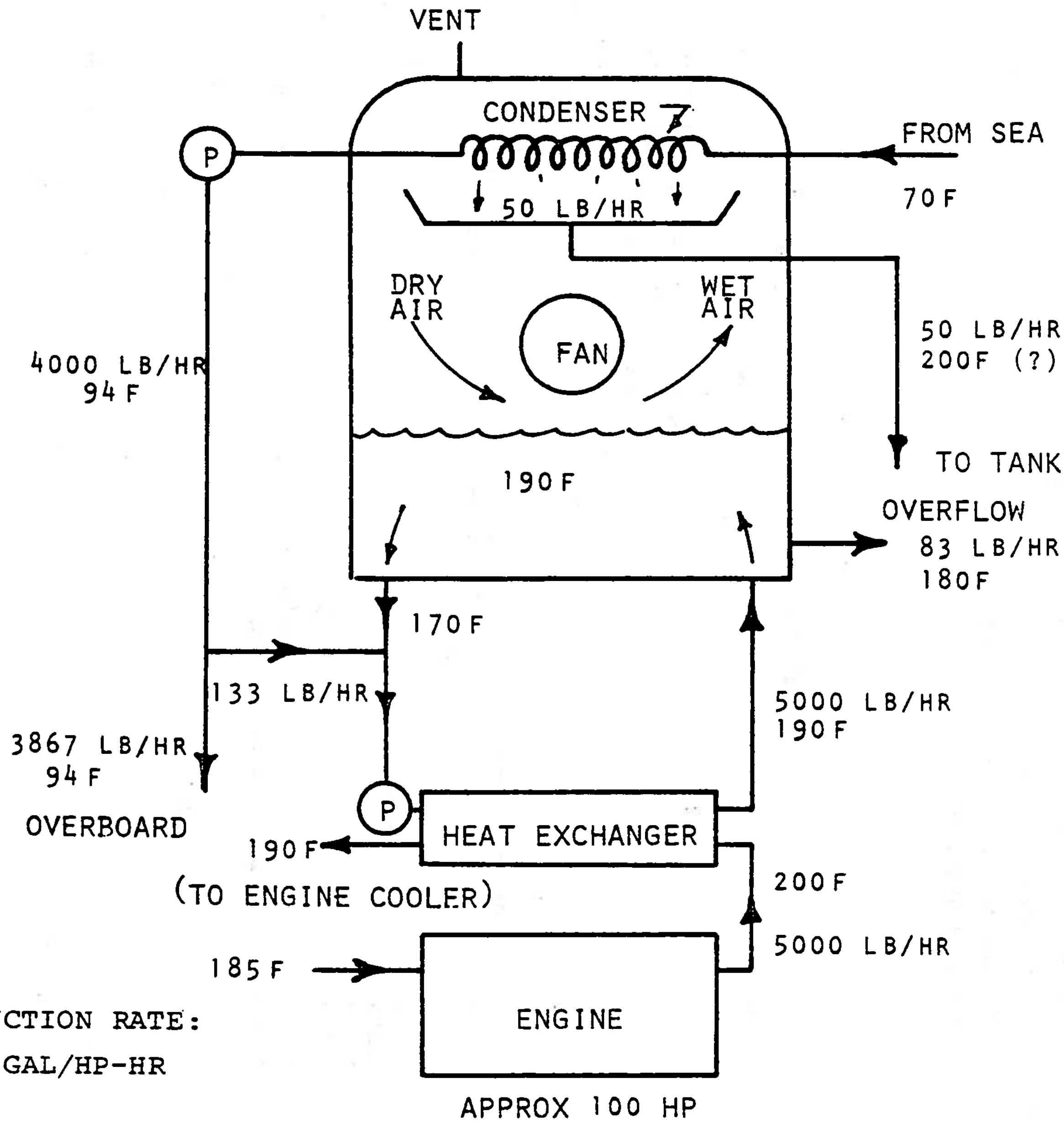


FIGURE 10.6 Heat Balance for a Waste Heat Distilling Plant Similar to Unit Shown in Figure 9.5



PRODUCTION RATE:
0.20 GAL/HP-HR

FIGURE 10.7 Heat Balance for a Waste Heat Distilling Plant of the Humidification Type

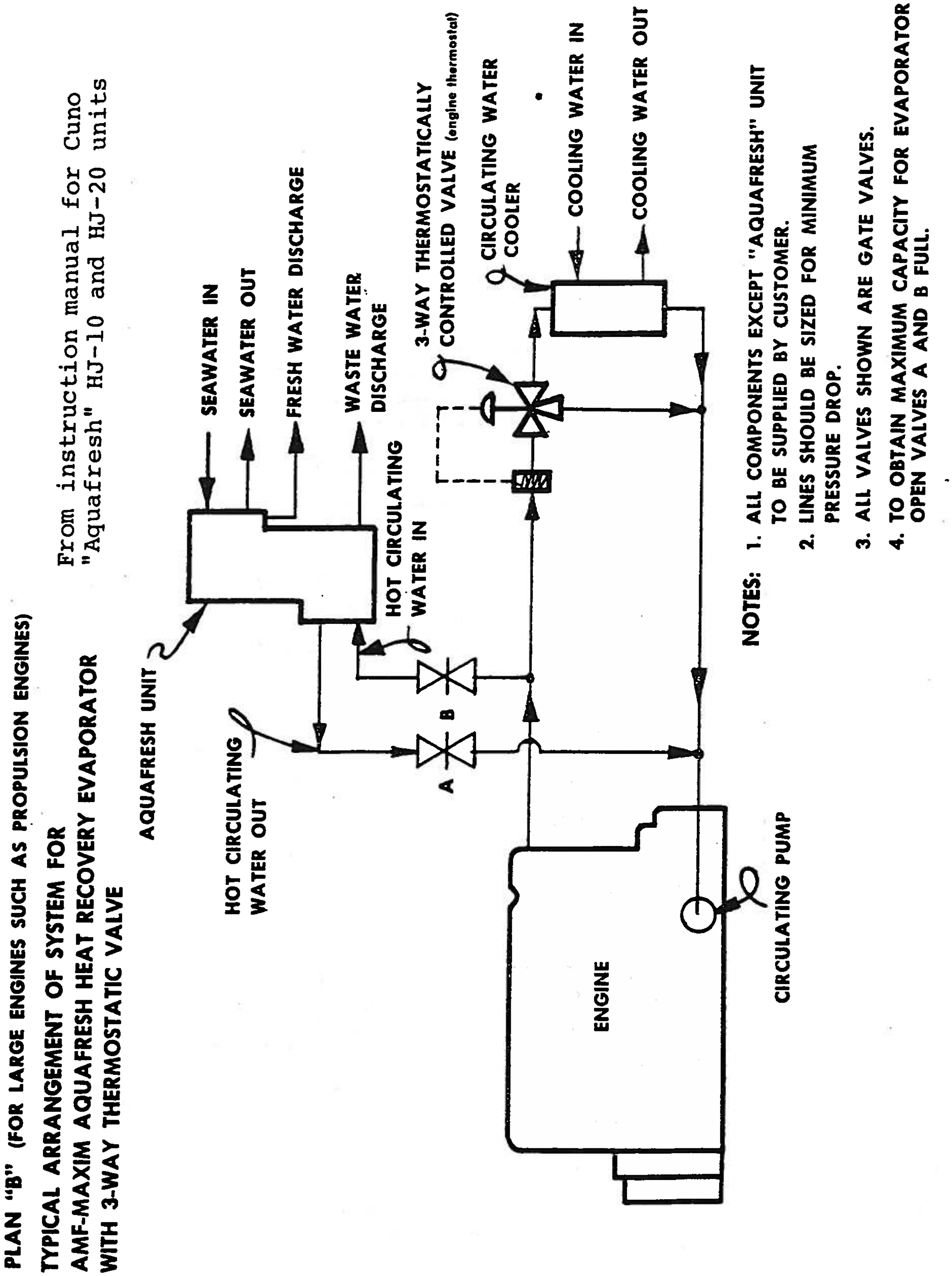


FIGURE 10.8 Typical Arrangement of Engine Cooling System with Waste Heat Distilling Plant

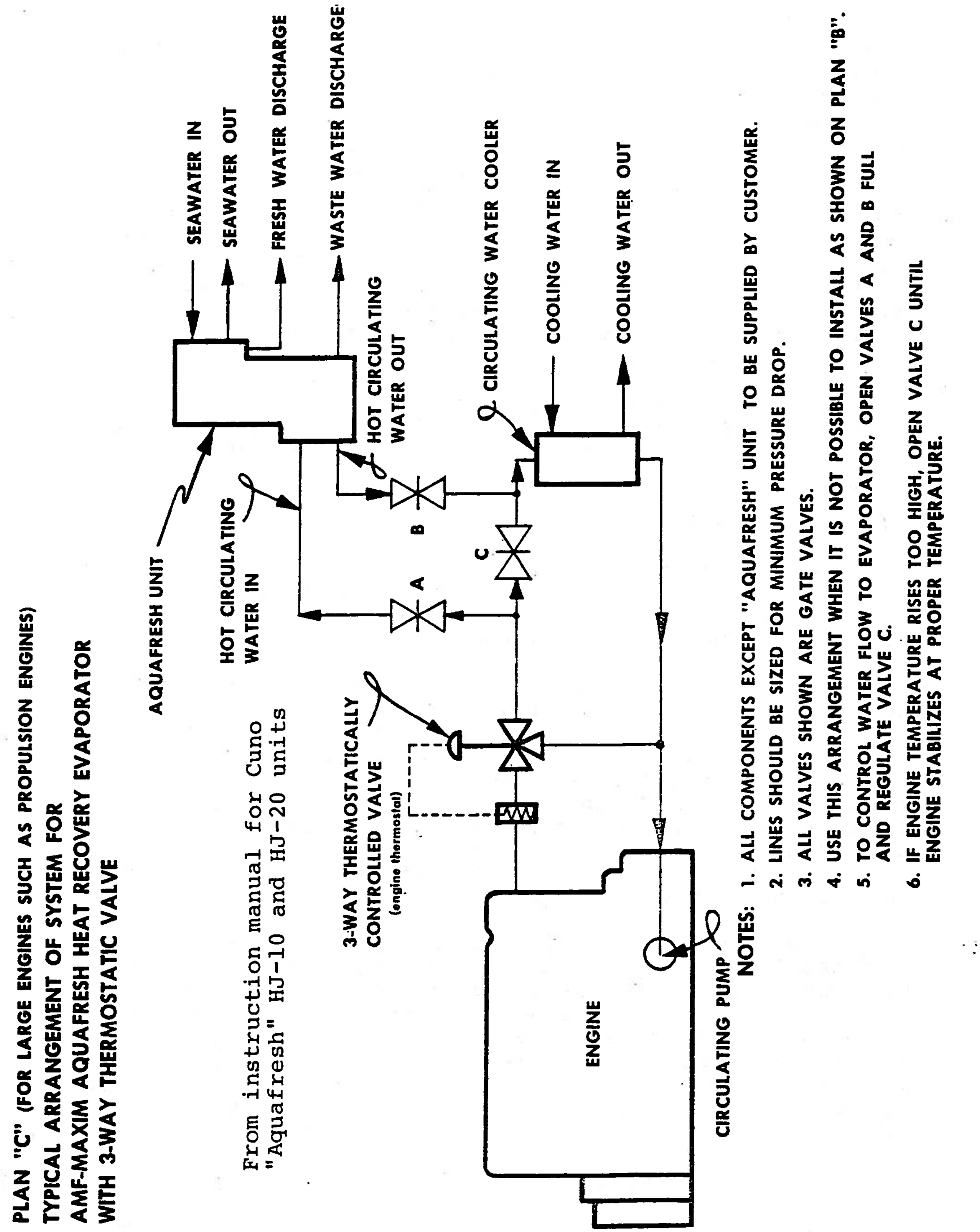


FIGURE 10.9 Typical Arrangement of Engine Cooling System with Waste Heat Distilling Plant

and alarms are used. In addition to these, an interlock scheme might be devised, using solenoid valves, such that C could be closed only when A and B are open. The piping arrangement suggested by an engine builder [5] is sketched in Figure 10.10; it is similar to Figure 10.9 except that valve C is omitted, and a pump is added to circulate the water through the distiller. This arragement is not as good from the standpoint of the distilling plant, since some hot water inevitably must bypass it, but better for the engine since its cooling circuit cannot be inadvertently blocked. It's an interesting question just how much water goes through each of the parallel paths available to it; the answer depends on the head of the distilling plant pump, and the frictional resistance to flow of each path. The general principle that underlies the analysis of these is that the total head difference between the junction points must be the same via both paths, the flows dividing such that this principle is obeyed. See Reference 8 for further discussion.

10.4 AIR INTAKE

Under favorable circumstances, an engine can simply take suction from the engine compartment, provided, of course, that the compartment is well ventilated. Several modifications are to be considered, however.

First, an air cleaner is usually recommended, especially for diesel engines. The cleaner is most often furnished with the engine, but some builders offer options as to the type of cleaner to suit the service. For example, a heavy-duty oil-bath type might be advisable for a tug that wrestles sand barges, while a dry paper type would be satisfactory for a pleasure boat. The air cleaner is often designed to function also as an intake silencer. For the spark-ignition engine, a spark arrestor at the intake is required by the 1940 and subsequent motor boat laws.

Second, the compartment may be too hot; recall the discussions in earlier chapters of the effect on performance of atmospheric density. The designer has the option of running a duct to the weather for engine intake. It can provide cooler air at the intake, but if the vessel operates in cold weather, this scheme may cause starting difficulties, though a means of diverting to an internal suction should be easy to provide. The outside intake also increases the danger of salt spray entering the engine, so that care in placing the weather intakes, and the use of at least one baffle in the duct are obvious suggestions.

The question of how much air is to be consumed faces the designer, however he is arranging the air supply, since he will have to design ducts or openings of sufficient size to pass the air at a minimum pressure drop. Table 10.2 will

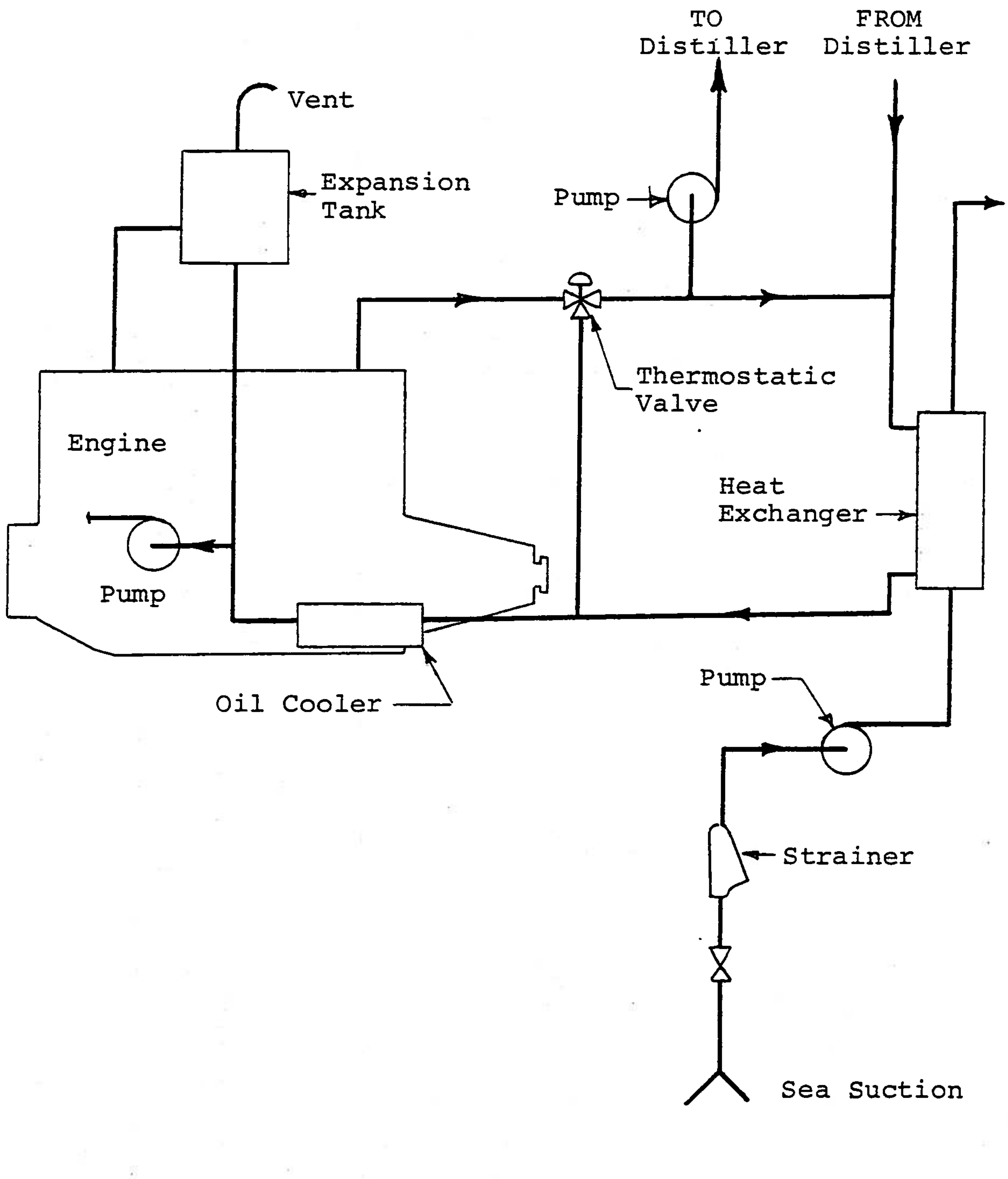


FIGURE 10.10 A Hookup for Waste Heat Distilling Plant Adapted from Caterpillar Recommendation [5]

TABLE 10.2 Air Consumption Rates for Typical Diesel Engines

MFGR	Type[2]	Size[3]	Displ	RPM	Air Rate[4]	Air Rate[5]	Source
EMD[1]	2S,NA	9-1/16x10x8	5160 IN^3	900	3.64	1.32	EMD Application Literature
EMD	2S,TC	9-1/16x10x8	5160	900	3.02	1.63	" " "
CUMMINS	4S,NA	5-1/2x4-1/6x8	785	3000	2.10	0.68	Cummins " "
CUMMINS	4S,TC	5-1/8x6x12	1486	2100	2.33	1.55	" " "
CATERPILLAR	4S,TC	4-1/2x5-1/2x6	525	2200	2.37	1.73	Note 6
CATERPILLAR	4S,TC	-	-	-	2.5		Note 7
-	4S,NA	Supposed Typical Values			2.7		
-	4S,TC				3.0		Note 8
-	2S,NA				4.1		
-	2S,TC				3.7		

Notes:
1 EMD = Electromotive Div., General Motors
2 2S = 2-stroke cycle, 4S = 4-stroke, NA = Naturally Aspirated, TC = Turbocharged
3 Bore-Stroke-Number of Cylinders
4 CFM/HP (HP used here is highest advertised BHP)
5 CFM/CFM where denominator is engine displacement x RPM
6 D. W. Knopf, M. B. Morgan, F. P. Butke, "Compact New Caterpillar Diesels," SAE Paper 254A, 1960
7 Reference 4
8 M. L. Baker, "Systems for Extracting and Utilizing Engine Rejector Heat," _Proceedings_, ASME Oil and Gas Power Division, 1963

give an idea of how this runs for diesel engines. The spark-ignition engine uses somewhat lesser amounts of air; the air/fuel ratio for diesels is seldom less than about 25/1, while the ratio for an SI engine is seldom greater than about 15/1. However, the air supply rate to the SI engine is based on the need for safely dissipating fuel vapors, and not on actual engine needs. Legal requirements [3] determine duct sizes etc. (as this is being written, the requirements of the Boat Safety Act of 1971 have not been promulgated).

In addition to combustion requirements, there is the need for air to cool the machinery compartment. Even if this space is unmanned, the machinery itself may suffer from excessive temperature. This is especially true for electrical motors and generators. The rate of air supply for this purpose is a function of the outside air temperature, the exhaust temperature desired, and the heat given off by the engine and other hot sources. It can be expressed by the handy formula

$$C = \frac{H(T_i + 460)}{9.6(T_o - T_i)} \tag{10.1}$$

where C = air supply rate, cfm (cubic feet per minute)

H = rate of heat liberation within the engine room, Btu/min

T_o = temperature of exhaust from the compartment, which is effectively the temperature of the compartment itself, F

T_i = temperature of the outside air supplied, F

The sticky part of this is knowing or finding the heat liberation rate, H. Rather detailed (and probably inaccurate) calculations can be made, based on estimates of engine surface temperatures. A better way is to use estimates supplied by the engine builder. For example, Caterpillar [5] publishes such data for all its marine engines. It should be noted however, that some hot things will not be under control of the engine builder. A prominent example is the exhaust pipe; complete is insulation of exhaust lines within the machinery space is essential if dry exhaust is used, and desireable even if wet exhaust is used.

As a rule of thumb, Reference 4 can be quoted as advising 1.25 cfm/hp for ventilation requirements, this in addition to air for the engine itself.

10.5 ENGINE EXHAUST

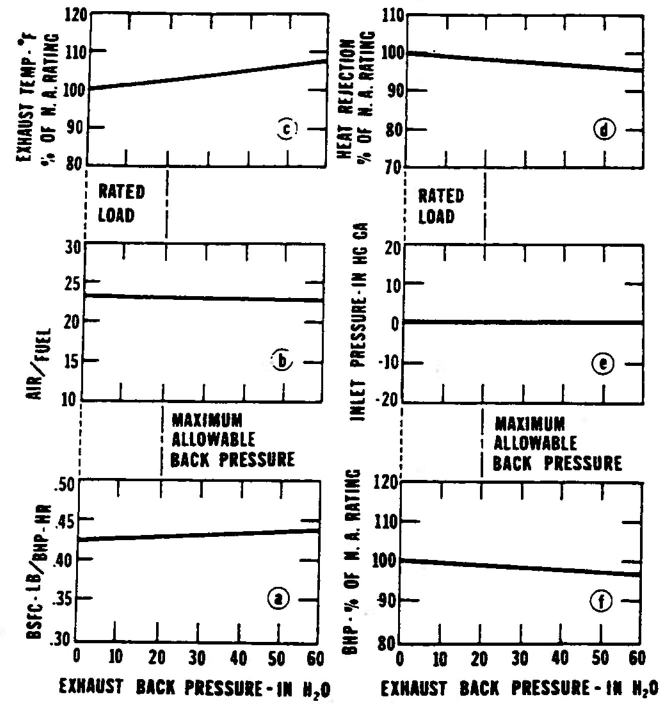
The thing to do with exhaust is get rid of it as expeditiously as possible. From the standpoint of the engine, this means with as low a back pressure as possible, since excessive back pressure adversely affects engine performance. Figure 10.11 (from [9]) gives an idea of the magnitude of the several back pressure influences. The engine builders generally specify a back pressure for which the engine is rated, and sometimes in addition a value that should not be exceeded. The specified value is usually easy to meet with the size of exhaust line that is usual (perhaps also specified by the builder) with the particular engine, but if the line is unusually long, or has an unusual number of bends, the designer may have to undertake a pressure drop calculation to establish pipe diameter. Although the problem is one of pulsating flow of a compressible fluid, the elementary methods used for steady incompressible flow give adequate results; readers not familiar with these methods can consult such sources as References 8, 10. But even when well acquainted with the methods, a designer needs to know the temperature and flow rate of the exhaust gas in order to calculate. This is information that the engine builder should supply; meanwhile Table 10.3 gives an idea of what to expect from the diesel engine.

In this table you will note the distinct difference in temperature between the two-stroke and four-stroke engines. The former run lower because of the cooling effect of the scavenging air. For a particular engine, higher rating means higher exhaust temperature. This point is illustrated by giving entries at two different ratings for the same Caterpillar engine. As a rough rule of thumb, an engine's exhaust temperature (^{o}F) is proportional to its BMEP.

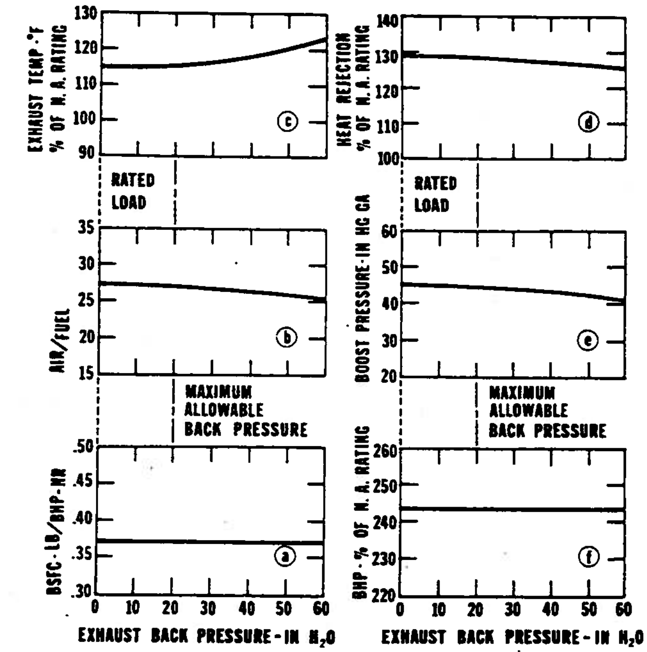
The weight of exhaust gas is generally higher for the two-stroke engines. The difference between two-stroke and four-stroke is less distinct in volume flow because of the higher temperature of the latter.

Builders of spark-ignition engines are less free-handed with engine data than their diesel rivals, so that I have no comparable figures to offer for this type of engine. For a rough estimate, it may be assumed to be the same, quantity-wise, as the four-stroke, naturally-aspirated diesel, but with temperature perhaps two hundred degrees higher because of the lower air/fuel ratio.

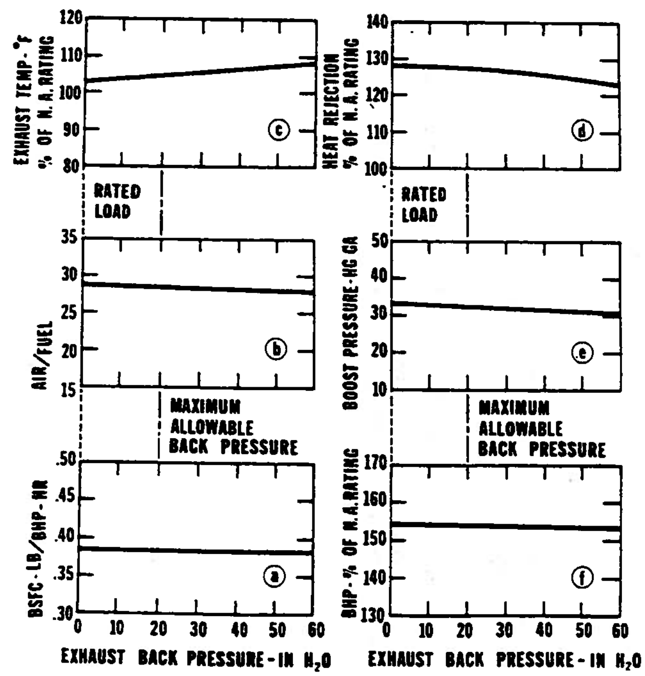
An interesting aspect of exhaust system design is the possibility of <u>tuning</u> the pipe for lower back pressure, and thereby obtaining better performance from the engine. The input to the exhaust pipe is actually pulsating (a pulse for each opening of an exhaust valve), and hence the pipe may resonate with the input frequency in the manner of an organ



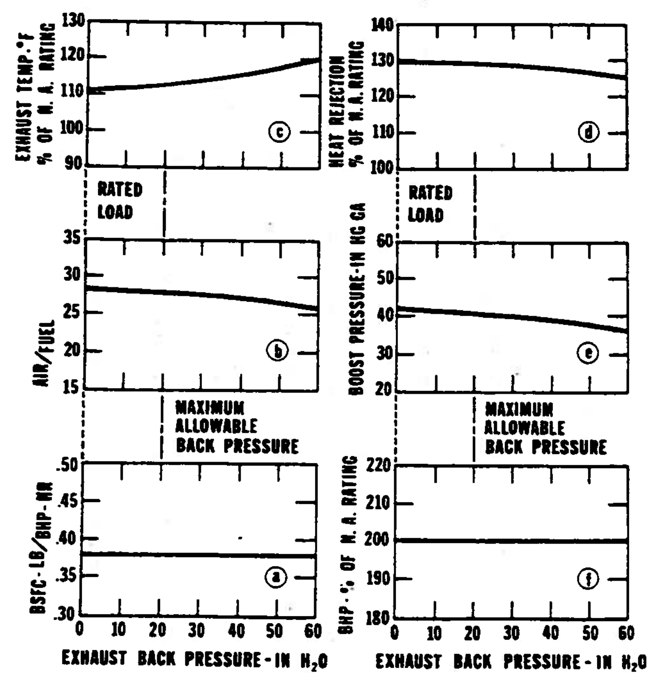
Influence of exhaust back pressure on engine performance: naturally aspirated engine; constant fuel rate



Influence of exhaust back pressure on engine performance: turbocharged-low temperature aftercooled engine; constant fuel rate



Influence of exhaust back pressure on engine performance: turbocharged engine; constant fuel rate



Influence of exhaust back pressure on engine performance: turbocharged-jacket water aftercooled engine; constant fuel rate

FIGURE 10.11 Example Curves Showing Influence of Back Pressure on Diesel Engine Performance (from Reference [9])

TABLE 10.3 Exhaust Gas Temperatures and Flows for Typical Diesel Engines

MFGR	Type[2]	Size[3]	BMEP[4]	Exhaust Temp	Exhaust[5] Flow	Source
EMD[1]	2S,NA	9-1/16x10x8	84 PSI	775 F	8.4	EMD Application Literature
EMD	2S,TC	9-1/16x10x8	124	737	6.5	" " "
CUMMINS	4S,NA	5-1/2x4-1/6x8	101	1100	5.9	Cummins " "
CUMMINS	4S,TC	5-1/8x6x12	152	1000	6.2	" " "
CATERPILLAR	4S,TC	4-1/2x5-1/2x6	170	1400	7.6	Note 6
CATERPILLAR	4S,TC	4-1/2x5-1/2x6	120	1200	7.1	Note 6

Notes:
1 See Note 1, Table 10.2
2 See Note 2, Table 10.2
3 See Note 3, Table 10.2
4 At Max Advertised BHP, except for last, which is BMEP at continuous marine rating
5 CFM/BHP at rating
6 See Note 6, Table 10.2

pipe. If it does, there exists a standing wave within the pipe. For example, for a two-stroke, single-cylinder engine, if the length of the pipe is $60C/4n$, where C is the speed of sound in the exhaust gas, and n is the impulse rate per second, a standing wave is established with the peak pressure at the engine when the exhaust valve is open. This is bad; the engine sees a higher back pressure than static design analysis would indicate. Such adverse tuning is particularly undesireable with a two-stroke engine, since it impedes the scavenging process. The remedy is to tune for minimum pressure at the exhaust valve. This condition is obtained when the period of the standing wave in the pipe is equal to the exhaust port opening period,i.e., when

$$L \approx 60\alpha\, C/4n \qquad (10.2)$$

where α is the fractional part of a revolution during which the exhaust valve is open.

Note that the discussion so far applies to a single-cylinder engine. In the usual multi-cylinder engine, the exhaust valve openings overlap, so that the formulas do not apply, and tuning is of negligible benefit. If you wish to take advantage of the tuning concept with a multi-cylinder engine, the thing to do is use an individual exhaust pipe for each cylinder. This complication is rarely worthwhile in the common commercial craft or in stock-model pleasure boats, but is quite the usual thing in streak-for-hell racing craft.

The discussion of tuning here is mostly from Reference 11. This source is recommended for further details.

Some words might be said about mufflers, although space doesn't allow a complete treatment. Reference 12 is the source used for the discussion here, and is recommended for further reading. Many textbooks on internal combustion engines also have discussions.

Placement of the muffler is perhaps the most likely problem to be faced by the marine designer, since he will usually buy the muffler itself as a stock item, while the total exhaust line must be designed to suit the boat. It is most desireable to place the muffler at the point where the pressure peaks of the pulsating gas flow are likely to occur. The best [5,12] place for the muffler is right at the engine exhaust discharge; the next best is at 40% of the length of the exhaust line from engine to outlet; at 80% is said to be almost as good as this second choice. The points halfway between those mentioned are the worst.

There are many internal variations in muffler design, far too many for enumeration here. If you are confronted with the need to design your own, consultation with a reference such as number 12 is advised, though the essential ingredient of the best mufflers is simply sufficient volume to dissipate the pressure pulses. This volume plus some baffling will make a good muffler even when designed without much attention to acoustic niceties. Several formulas for volume can be found; one recommended by Reference 12 is

$$\text{Volume} = \frac{Q \times \text{engine displacement} \times \text{RPM}}{1000 \sqrt{S \times N}} \qquad (10.3)$$

where the volume is in cubic inches for displacement in cubic inches.

S = 2 or 4, depending on whether 2 or 4 cycle engine

N = number of cylinders

Q = a coefficient = 2 for minimal quieting
= 3 for good quieting
= 4 for high-grade quieting
= 5 or 6 for automotive use

10.6 EXHAUST SYSTEMS

Figure 10.12 illustrates a typical wet exhaust system, i.e., one into which part of engine cooling water* is injected to cool the pipe and to assist the silencing. The principal concern in laying out such a system is to obviate the possibility of water entering the engine, either cooling water, sea water, or condensation. A definite slope away from the engine is a must, and unless the exhaust manifold is well above the waterline, an inverted U in the line is necessary. Typically the U is a water-jacketed "riser" furnished with the engine. Water injection takes place directly downstream of the riser.

A flexible section in the line is needed to absorb thermal expansion as the line heats. It can be rubber since the injected water will prevent excessive temperature.

Only a part of the engine cooling water need be injected into the exhaust. In fact, full flow is likely to cause excessive back pressure in exhaust lines of normal size. Since it is difficult to say in the design stage what the flow to exhaust should be, and difficult to design for a certain split even if known, the usual approach is to provide a balancing valve in the water line to the exhaust, and one in the overboard line, so that flow can be adjusted for adequate cooling and noise suppression after the vessel is built.

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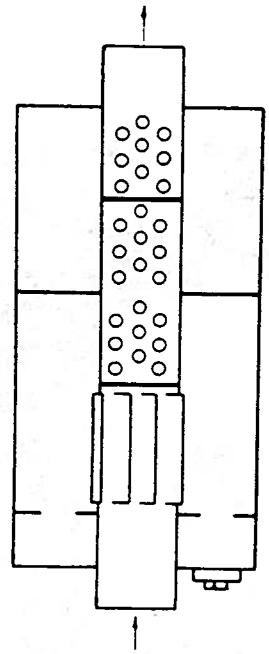
* In the case of indirect cooling, this would be the sea water, and _not_ engine jacket water.



Spinner type vertical spark arresting muffler

FIGURE 10.14 Illustrating a Spark-Arresting Silencer (from Reference[12])

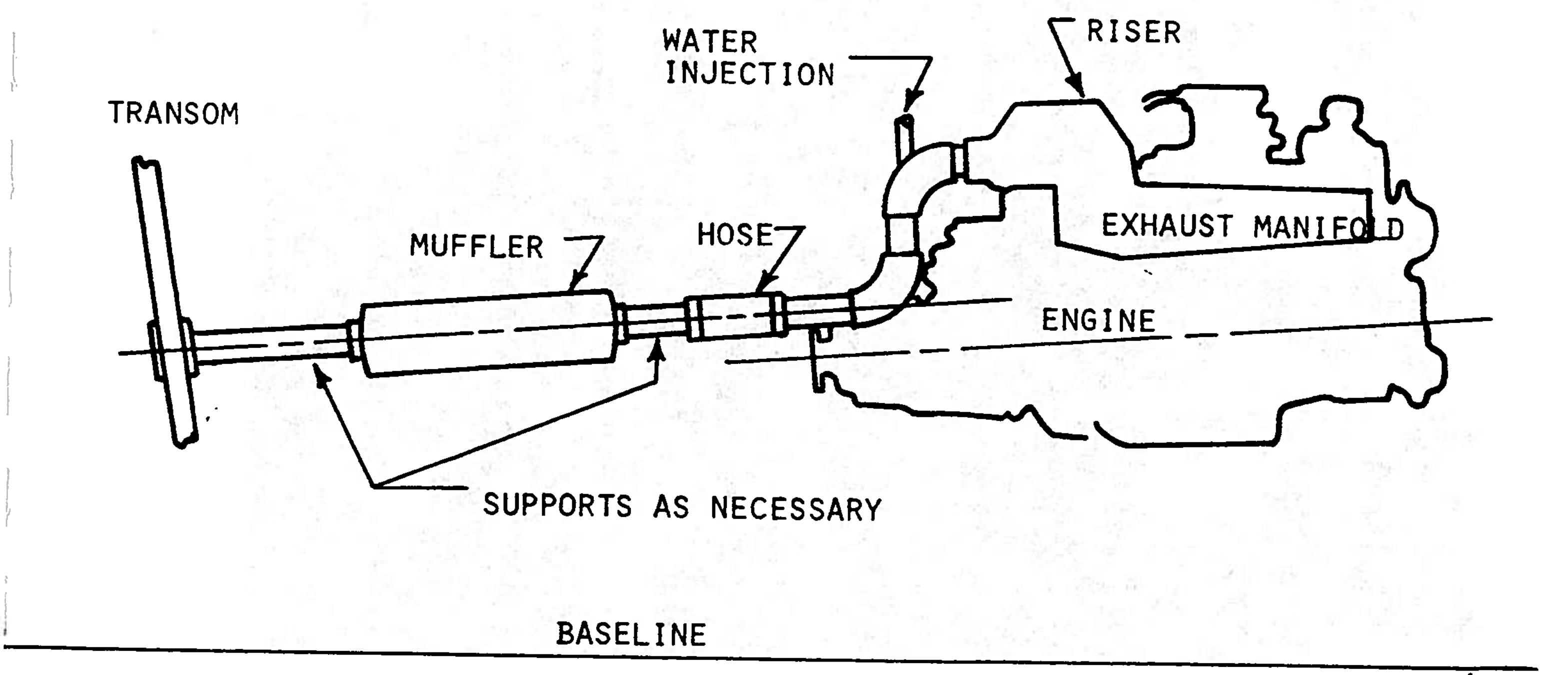


FIGURE 10.12 Elements of a Wet Exhaust System

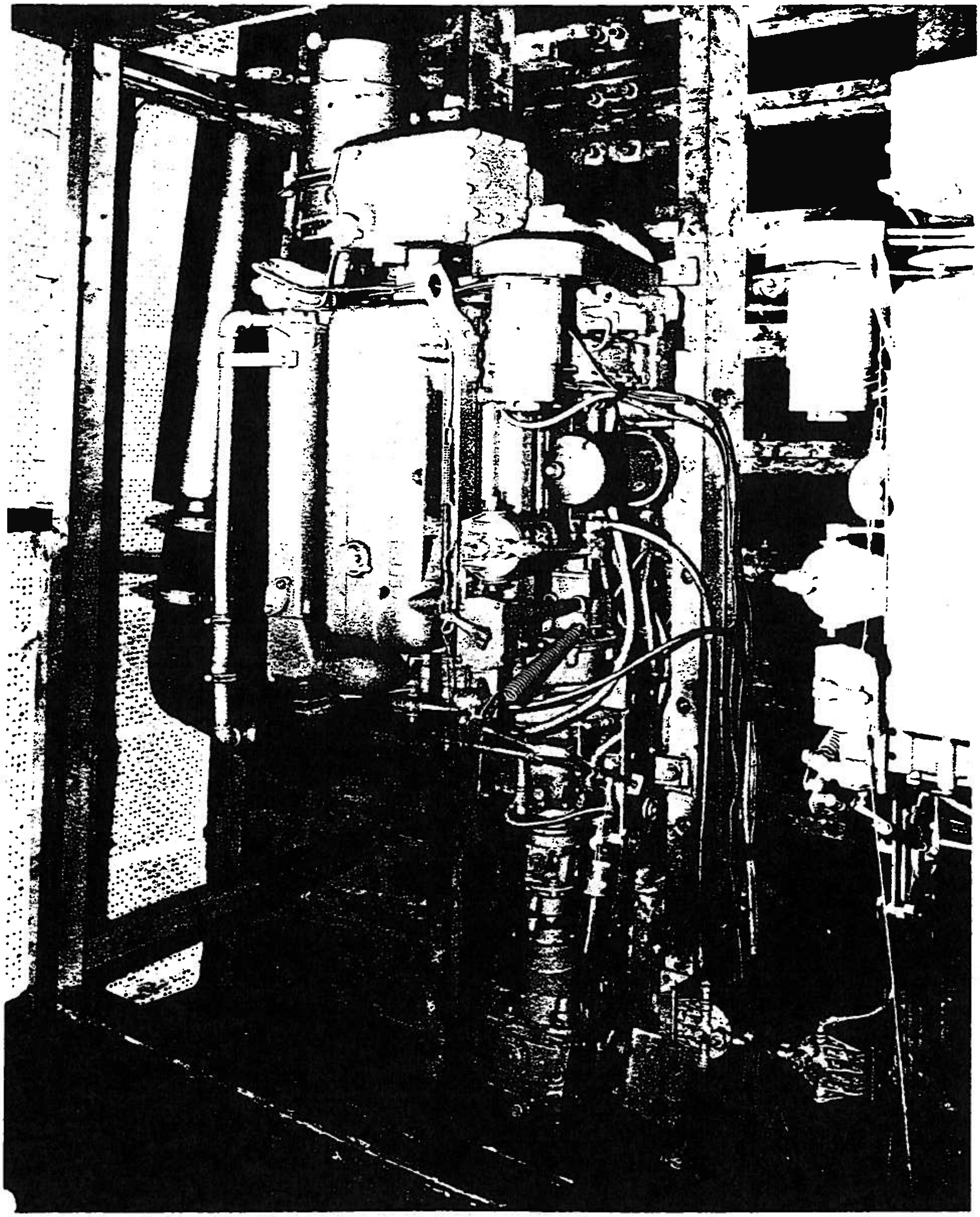


Figure 10.13 Engine Installation Showing Wet Exhaust
(Courtesy Grafton Boat Co)

The hot raw water and the combustion products are a nasty mix from the corrosion standpoint. Gray cast iron, copper-nickel, and monel are usually regarded as satisfactory metals [13]. Neoprene is sometimes used for mufflers and the flexible hose section.

The wet system has several advantages over its alternative, the dry system. It is usually quieter, less likely to deposit soot on deck, and due to its relative coolness is not a fire hazard to a wooden boat. It has potential defect of grave importance, however. It is nearly always run straight aft and out through the transom, and in some pleasure craft, this means that it runs through living spaces. Being an ugly old pipe, it is in such cases concealed behind panels. Deterioration may go unnoticed, and when the pipe subsequently leaks, the consequences are likely to be fatal for the occupants of the affected space. Regulatory and advisory bodies are oddly almost silent; the American Boat & Yacht Council [1] meekly observes "The exhaust system shall be gas-tight throughout its passage through the interior of the hull" (project P-1). A fine thought, but hardly sufficient, for what is gas-tight today may develop leaks tomorrow, and most boat owners feel that dying tomorrow is nearly as bad as dying today. A prohibition against running of exhaust lines through living spaces might well be in order. At the least, the piping through such spaces should be jointless, and made of a material that will not rust through, such as monel or copper nickel.

Figure 10.13 is a photograph of a small-craft diesel engine in place, with part of its exhaust line (wet), including muffler, clearly visible.

The alternative to a wet exhaust system is (who'd ever guess!) a dry system. They generally take the most direct path to the outside, namely more or less straight up from the engine. Because water is not injected, the exhaust line is hotter and noisier than the wet system. The high temperature is a hazard, especially on wooden boats, because of the danger of fire. Heavy insulation is a must, and careful measures must be taken to keep the pipe isolated from combustible material. Detailed safety rules have been published [14] and the designer is advised to heed them carefully, because fires started by hot exhaust pipes are not at all rare.

High temperature also implies significant thermal expansion. If the exhaust pipe is anchored to the vessel structure, a metallic expansion joint between the anchor point and the engine is essential to prevent excessive thrust on the engine, and possible overstressing of the pipe itself.

Dry systems are also said to dirty the decks because at least occasionally they will spew forth some soot. Thus they seem to have numerous points against them (noise, heat, dirt), but they are much less a hazard than the wet system from the leakage standpoint. Since they usually lie wholly within the machinery space or a trunk above it, the ventilation that is required for other purposes is likely to be sufficient to keep even those spaces safely clear of carbon monoxide in the event of a small leak.

The muffler used in a dry system should be of the spark arresting type, since an occasional fleck of hot carbon will break loose from engine or exhaust passage, and be a threat to anything combustible on deck. There are several designs of spark arrestors; one example is shown in Figure 10.14. It imparts a spinning motion to the gas passing through, thus centrifugally separating any solid particles into the outer chamber.

Figure 10.15 is a photograph of an installed marine engine showing part of a dry exhaust system. Sheet metal lagging around the pipe insulation is apparent.

10.7 FUEL SYSTEMS

The principal concerns in fuel systems is delivery of clean fuel, and the adherence to strict safety standards.

Safety is of special importance with the spark-ignition engine, and so has received extensive attention from regulatory and advisory bodies. Many details are specified, and since they appear in publications readily available to any designer, it is not worth the trouble to repeat them here. Let us, however, list the broad areas of consideration:

(1) Escape of liquid gasoline or its vapors is to be avoided by stern measures. For example, drain plugs in fuel tanks, or indeed, any connections to tank sides or bottom, are not allowed.

(2) All metal parts are to be bonded together and grounded to prevent a static electricity buildup that might lead to sparks.

(3) Tanks and tubing should be able to withstand an external gasoline fire long enough for extinguishing means to take effect.

(4) Tanks and tubing should be carefully supported to prevent working loose of mechanical joints and work-hardening failure of tubing.

(5) Tanks should be constructed of materials that do not promote the formation of gums and tars in

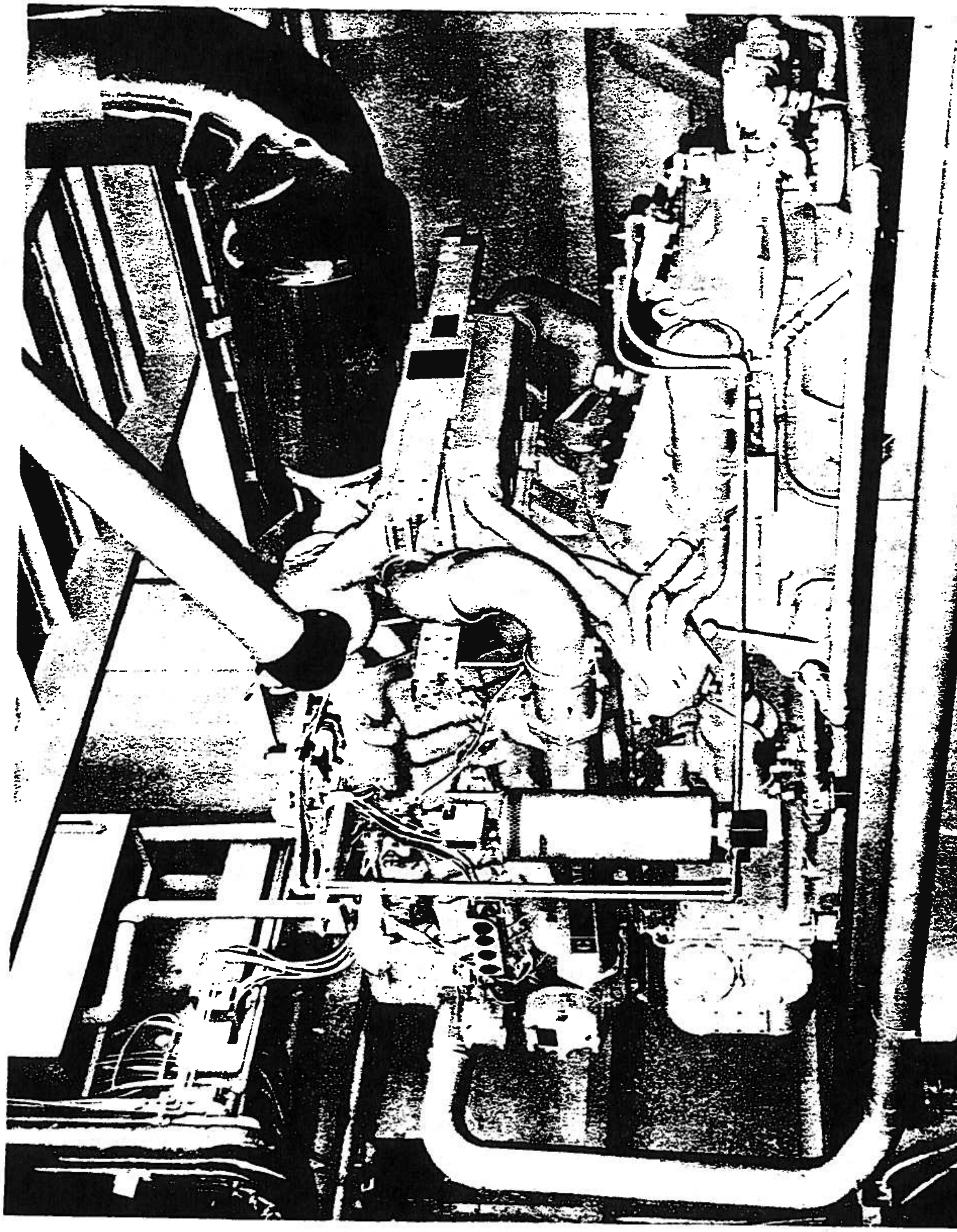


Figure 10.15 Engine Installation Showing Dry Exhaust
(Courtesy Grafton Boat Co)

gasoline. (This is more for protection of the engine than for boat safety.) Acceptable materials are listed in the references.

A diagramatic picture of a typical small-craft fuel system, taken from the Boating Industry Association manual [2] is shown in Figure 10.16.

Figure 10.17 shows a diesel fuel system, adapted from details given in Reference 5. A feature of this system is the standpipe from which fuel suction is taken, and to which excess fuel is returned from the injectors. This scheme is intended to avoid the heating of the tank contents that would occur if fuel, having picked up heat from the engine, were returned directly to the tank. An alternative is to return fuel to the tank, and have a water-cooled coil placed against one side of the tank.

Clean fuel is vital to the diesel engine; consequently a fuel filter is always an integral part of the engine. These filters are not intended to cope with large amounts of sediment or water, however, so that a trap in the line ahead of the filter is advisable. This trap is basically a chamber of sufficient size that water and heavy particles can settle to the bottom as the fuel is drawn from the top. A strainer is usually incorporated.

The best level for a diesel fuel tank relative to the engine is one that places the half-full liquid level at the height of the injectors. Although putting it higher helps the fuel pump suction, it also leads to possible leakage of fuel into the engine while it is shut down. If the tank is placed high, a shutoff valve in the supply line, and a check valve in the return line, are advisable.

Black iron or steel is the recommended material [5] for diesel fuel ines, and for tank construction also. Copper lines are sometimes used, though some authorities [13] recommend against it because of its possible reaction with the fuel. Galvanized material should be avoided because zinc reacts with sulfur in fuel to form compounds that will deposit within the fuel injection system.

10.8 REFERENCES

1. American Boat & Yacht Council, Safety Standards for Small Craft, 1971-72.

2. Boating Industry Association, BIA Engineering Manual (annual).

3. United States Coast Guard, Rules and Regulations for Uninspected Vessels, Subchapter C.

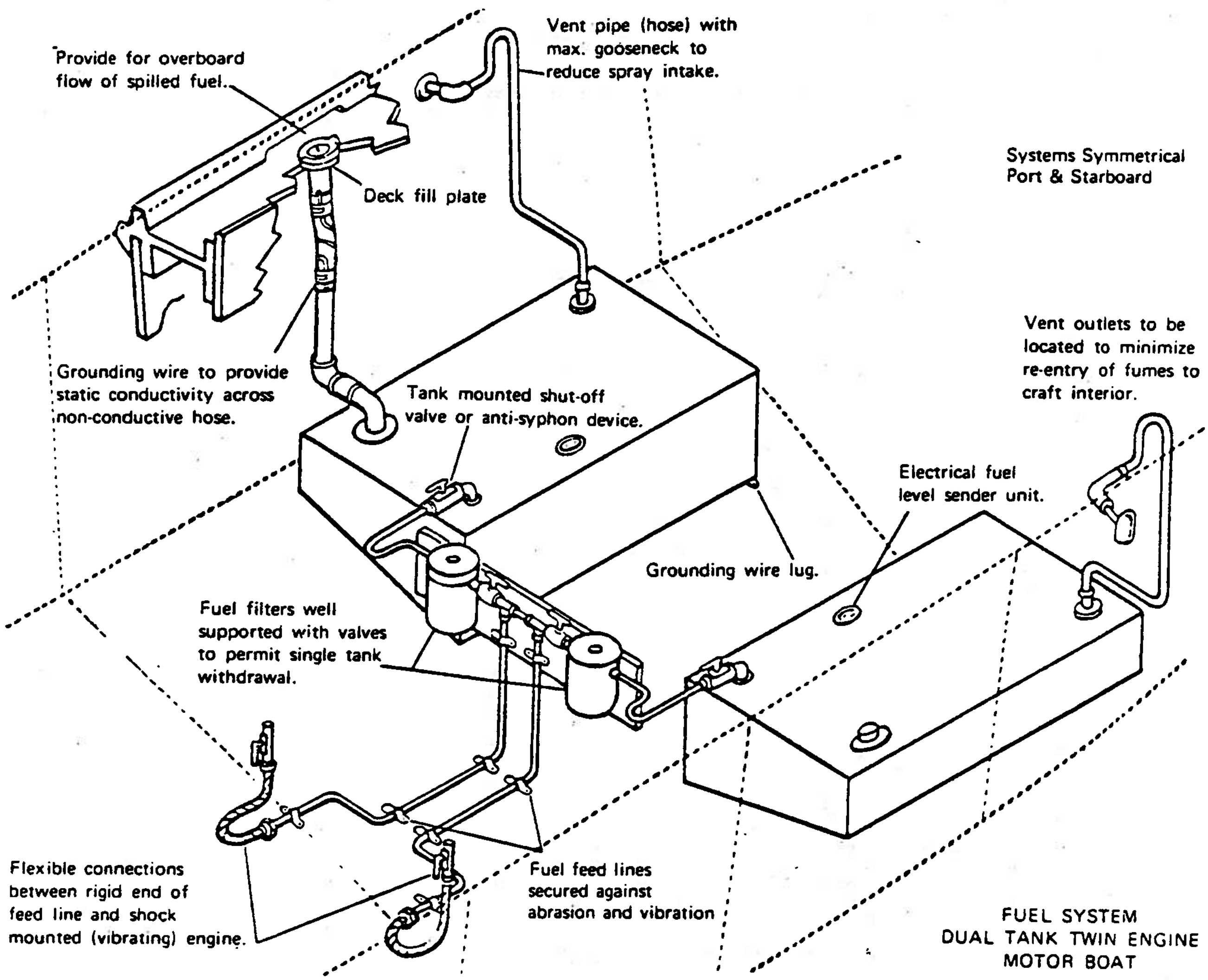


FIGURE 10.16 Typical Fuel System for Spark-Ignition Engine (borrowed from BIA Yacht Engineering Manual)

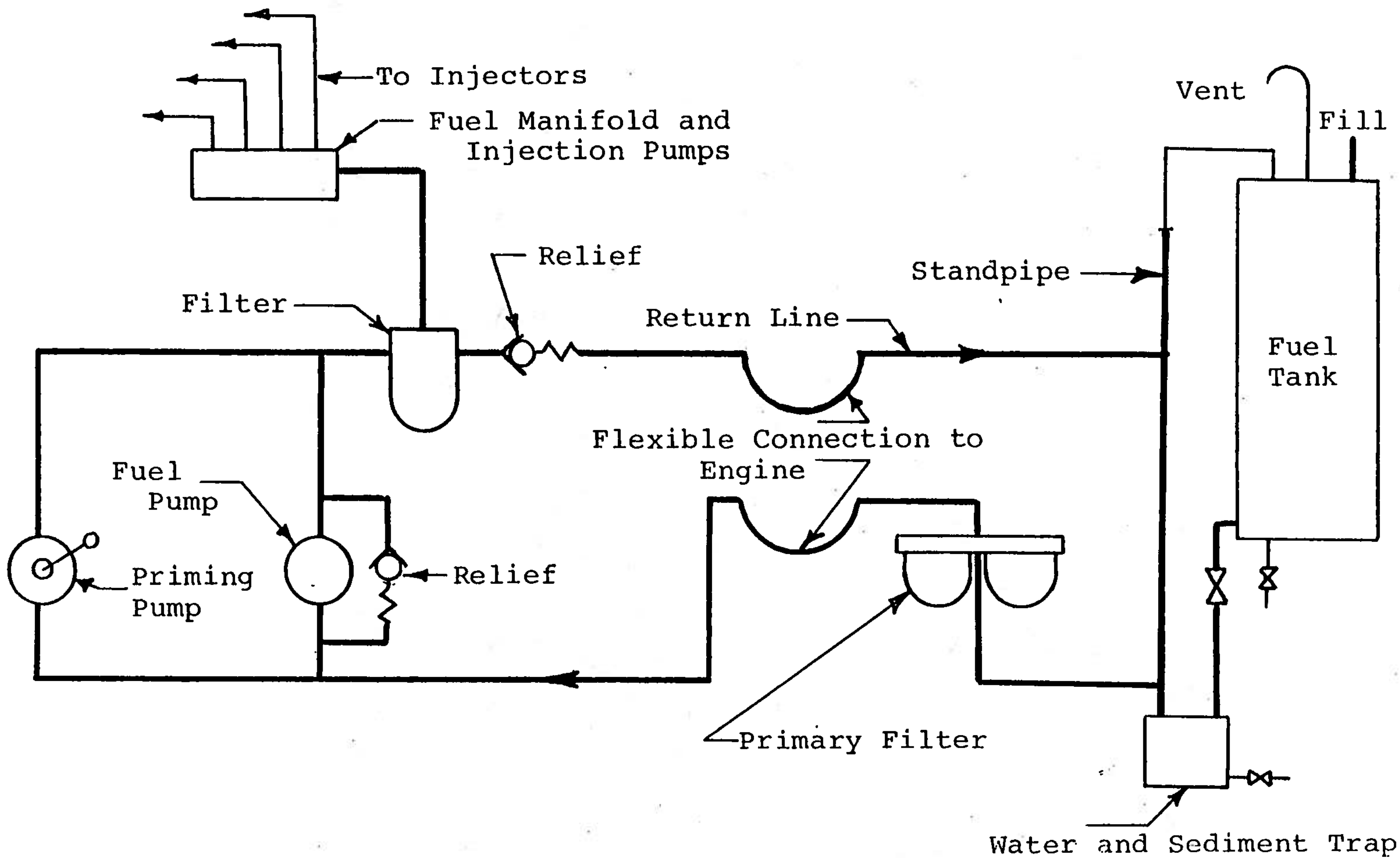


FIGURE 10.17 Diesel Engine Fuel System (adapted from Reference[5])

4. Alex Bryner, Caterpillar Tractor Comapny, "An Introduction to Marine Diesel Engine Installation."

5. Caterpillar Tractor Company, Cat Marine Engine Application and Installation Guide.

6. The Shipbuilder and Marine Engineer, April 1963, page 265.

7. L. Wechsler and H. Holler, "Development of Light-Weight High-Output Four-Cycle Diesel Engine for Naval Service," ASME Oil and Gas Power Proceedings, 1958.

8. John B. Woodward, Hydraulics for Piping Design, Report 103, Department of Naval Architecture and Marine Engineering, University of Michigan.

9. Harold G. Holler, "The Influence of Induction and Exhaust System Design on Power Producing Characteristics of Diesel Engines," Society of Automotive Engineers Paper 700535, 1970

10. The Crane Company, Technical Paper 410, Flow of Fluids Through Valves, Fittings, and Pipe.

11. Paul H. Schweitzer, Scavenging of Two-Stroke Cycle Diesel Engines, MacMillan, 1949.

12. Dean G. Thomas, "Muffler Selection and Design for Internal Combustion Engines," Society of Automotive Engineers Paper 700537, 1970.

13. Detroit Diesel Division, General Motors, Marine Engine Installation Facts.

14. NFPA 302, Motor Craft, 1966, National Fire Protection Association.

CHAPTER 11

AUXILIARIES CONTINUED

11.1 INTRODUCTION

This chapter continues the consideration of auxiliary topics appropriate to the diesel and spark-ignition engines. Lube oil, foundations, and starting are considered.

11.2 LUBE OIL

Small craft lube oil systems are typically integral with the engine, and so not among the tasks of the vessel designer. Several points are to be noted, however.

Standby lube oil systems are required for ocean service by some classification societies*, these systems consisting mainly of a pump with strainer and check valve in an external pipe branch that parallels the engine lube oil pump. On some diesel engines, suction and discharge connections are provided so that the external system can easily be added. Pump flow and head must be as specified for the engine.

On vessels of long cruising range, a tank for carrying spare lube oil is sometimes installed as a convenience, although cans of oil can serve. Capacity of the tank can be estimated from the engine builder's estimate of rate of oil consumption, specified change intervals, and cruising range desired.

Oil must be changed periodically. Therefore easy access to drain plugs should be remembered in planning the foundation. However, if such access can't be provided, an alternative is to pump the oil out via the dip-stick hole.

11.3 FOUNDATIONS - GENERAL

The foundation supports the engine in its alignment with the shaft. It must obviously support the weight of the engine, but it also transmits propulsion thrust to the hull, and is often intended to stiffen the hull in way of the machinery to minimize hull deflections that might affect alignment. Thus stout fore-and-aft members are the main constituent of an engine foundation. No particular rules can be cited for dimensions, though some engine builders publish advice thereupon [2].

* American Bureau of Shipping requires this for ocean vessels over 300 gross tons. [5]

Figures 11.1 and 11.2 show typical diesel engines mounted on their foundations. In both the liners used to maintain the aligned position of the engine (as mentioned in Chapter 10) are visible beneath the engine feet. Figure 10.13 also gives a good view of foundations in a wooden boat. In this figure, observe the flat-bar cap on the wooden stringers, this to prevent misalignment caused by crushing of the wood fibers.

11.4 FOUNDATION LOADS

If a designer wishes to make an analytical design of a foundation, he must begin with knowledge of the loads imposed. This not being a text on structural analysis, he won't find much help here with such analysis, but it is appropriate to discuss what the loads may be, and to give estimates of their magnitudes.

The loads are both static and dynamic. Weight and propeller thrust are examples of the former, while engine vibration can produce the latter.

Propeller thrust load can be calculated from the relations among thrust horsepower, effective horsepower, and shaft horsepower, which give

$$T = \frac{325 \text{ SHP } \eta_P}{V(1 - w)} \text{ lbs} \tag{11.1}$$

where η_P = efficiency of propeller behind hull

SHP = shaft horsepower

V = speed, knots

w = wake fraction

If you don't know values for η_P and w, just using

$$T = 300 \frac{\text{SHP}}{V} \tag{11.2}$$

is a rule-of-thumb estimate that is likely to be good enough for the present purpose.

There is also the average torque reaction, given by

$$Q = \frac{33000 \text{ SHP}}{2\pi \text{ RPM}} \text{ ft-lb} \tag{11.3}$$

which is present as an upward force on one side of the foundation, and a downward force on the other.

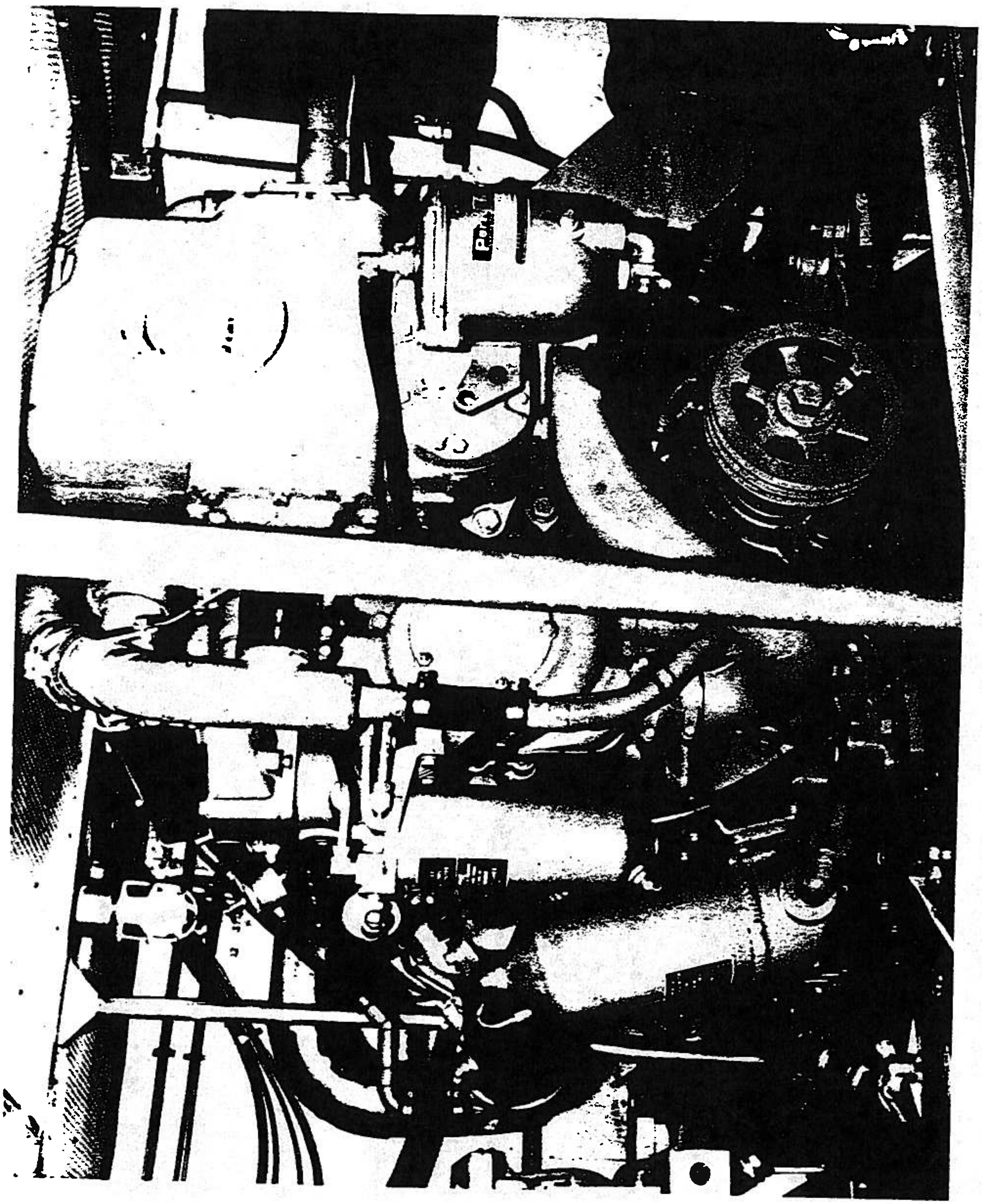


Figure 11.1 Engine Installation Showing Foundation (Courtesy Detroit Diesel)

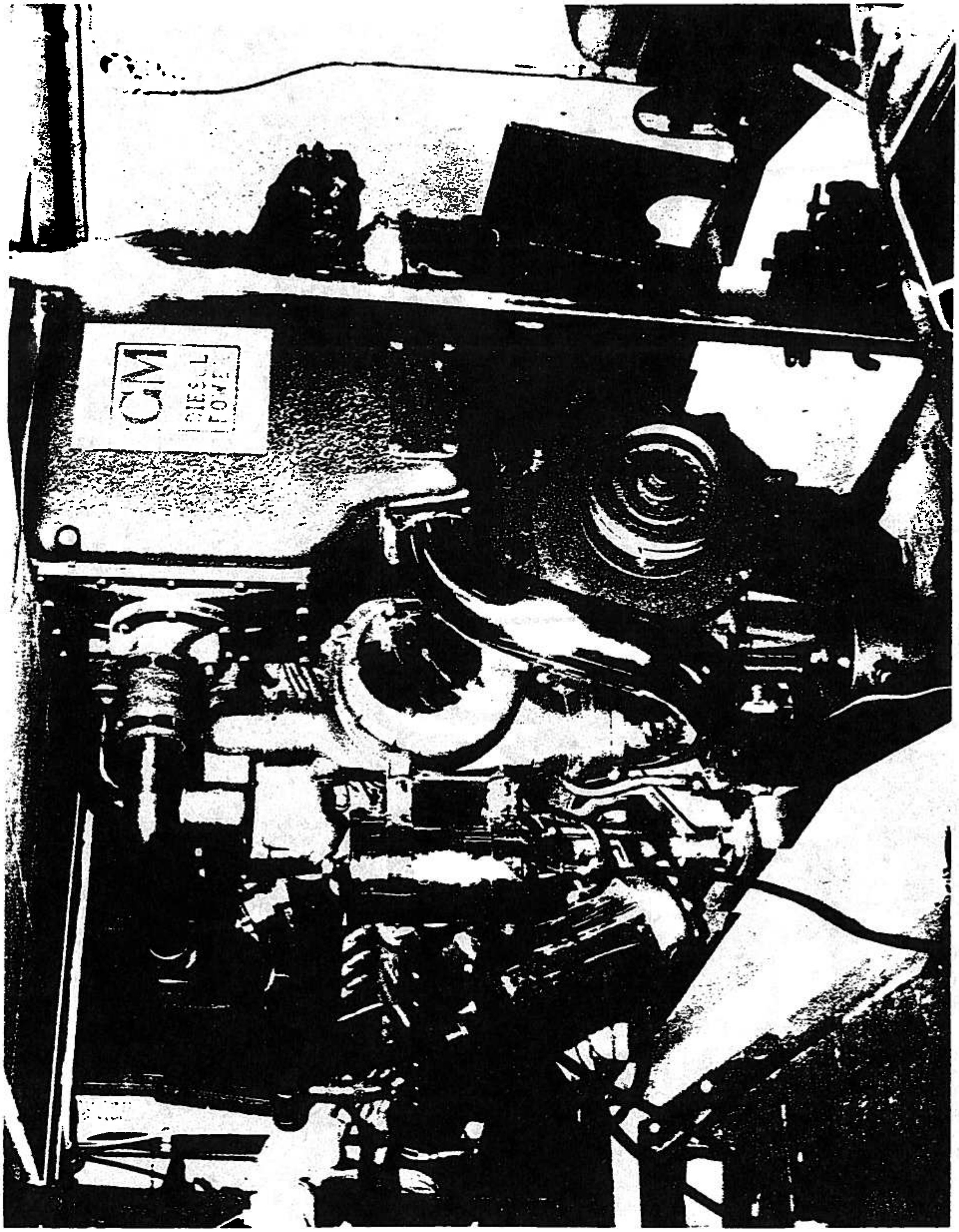


Figure 11.2 Engine Installation Showing Foundation (Courtesy Detroit Diesel)

Piping attached to the engine can apply significant thrust, especially piping subject to thermal expansion, such as the exhaust line. The remedy lies not in foundation design, however, but in providing flexibility in the piping to minimize this thrust.

The engine can also expand thermally. The principal sufferer is the engine itself, due to possible distrotion, rather than the foundation. Thus the remedy is to make the means of support flexible enough to allow the expansion with only a modicum of restraint. With the reciprocating IC engines, this is rarely necessary because the main structure remains relatively cool.

Rolling, pitching, and the other possible periodic motions of the vessel produce mass-times-acceleration forces. Formulas for the roll forces serve as an illustration. (Although easily derived if the roll is assumed to be a simple harmonic motion, the equations following are taken for convenience from Reference 3).

If the roll has a maximum roll angle (to one side of vertical) of θ_R degrees, and a roll period (complete cycle) of T_R, then maximum acceleration is

$$\frac{4\pi^3\theta_R}{180T_R^2} ,$$

and force is mass of the engine multiplied by this acceleration. If this force is resolved into horizontal and vertical components, and the constants evaluated, the equations are

$$H = \frac{0.0214\ W\ \theta_R\ y}{T_R^2} \quad \text{lb} \tag{11.4}$$

$$V = \frac{0.0214\ W\ \theta_R\ x}{T_R^2} \quad \text{lb} \tag{11.5}$$

where W = weight of engine, lbs.

x = horizontal distance, engine center of gravity to centerline of vessel, ft.

y = vertical distance, engine center of gravity to center of the roll motion, ft.

Roll periods are typically so long compared to any natural frequency of the foundation or any other structure, that they can be treated as static, i.e., simply added to weight and thrust loads.

Angular motions (rolling and pitching) also generate foundation forces due to the gyroscopic moment produced when rotating members are forced to change their planes of rotation. The moment acts in a plane perpendicular to the plane of the rotating member, and in the plane of the motion. For example, pitching causes the engine flywheel (rotating in an athwartship plane) to produce a moment in the vertical fore-and-aft plane, i.e., forces act up and down on the forward and aft ends of the foundation. Rolling would produce nothing from rotating elements in an athwartship plane.

The magnitude of the moment is given by [3].

$$M = \frac{WR^2}{9}\,\frac{2\pi N}{60}\,\dot{\theta} = 0.00325\ W\ R^2\ N\ \dot{\theta} \qquad (11.6)$$

where M = the moment, lb-ft

W = weight of the rotating element, lb

N = RPM of the rotating element

R = radius of gyration of the rotating element, ft

$\dot{\theta}$ = angular velocity of the inposed motion, rdn/sec

The moment causes forces on the extremes of the foundation, and these are directly calculable from its dimensions.

The period of these forces is that of the causing motion (e.g., pitching), and so this also can be treated as static forces.

Shock loads can occur, but only if somebody sets of a bomb under your boat, and therefore are not of general concern.

The U.S. Navy does concern itself with the effects of underwater explosions, so that shock loads are taken into account when designing for the Navy. Since it is of interest only in this special case, and since it is too big a topic to treat adequately here, it is passed over. Reference [3] can give you an introduction to the topic, and cites references for further reading. (Shock loads are, of course, dynamic rather than static.)

The reciprocating masses of the crank-connecting rod-piston assembly produce inertia forces and inertia moments about the crankshaft. In the usual multicylinder engine they are effectively cancelled by equal and opposite action of the different cylinders, so that inertia should cause the engine neither to hop up and down nor to rock from side. But the cylinder forces, acting oppositely at some distance apart, do cause internal moments that tend to bend the ends up and down. This tendency can be reduced by counterweighting of the cranks, and is opposed by the fore-and-aft stiffness of the engine structure. Since the engine cannot be infinitely stiff, there will be some tugging at the foundation bolts near the forward and aft ends.

Details of the engine balance analysis can be found in many textbooks. Reference 4 is a good example. However, calculation of external load magnitudes is not really feasible. Usually, in the case of dynamic loads, knowledge of the frequencies, or just the lowest frequency is sufficient (see further discussion in Section 11.5). The lowest frequency for the bending mentioned above is the RPM of the engine.

The engine instantaneous torque curve also contributes a dynamic load to the foundation via the torque reaction. The peak torque per cylinder cycle may be on the order of ten times the average, and this peak occurs once every revolution for a two-stroke engine, and once every two revolutions for a four-stroke engine. The lowest frequency is then found by multiplying this cylinder frequency by the number of cylinders.

11.5 FLEXIBLE FEET

This refers to those rubber supports such as typically used in mounting the automobile engines. Figure 11.3 shows a typical example.

To see the use and benefit of these devices, first look at the idealized picture of Figure 11.4: a mass m is supported by flexible members that are characterized by a spring constant $\underline{k}$. A periodic force $F_0 \cos \omega_f t$ acts on the mass. The first step is to determine the displacement of the mass under action of the force. The continuing periodic value of x is

$$x = \frac{\frac{F_0}{k}}{1 - (\frac{\omega_f}{\omega})^2} \cos \omega_f t \qquad (11.7)$$

The maximum value of x per cycle is therefore

$$x = \frac{\frac{F_0}{k}}{1 - (\frac{\omega_f}{\omega})^2} \qquad (11.8)$$

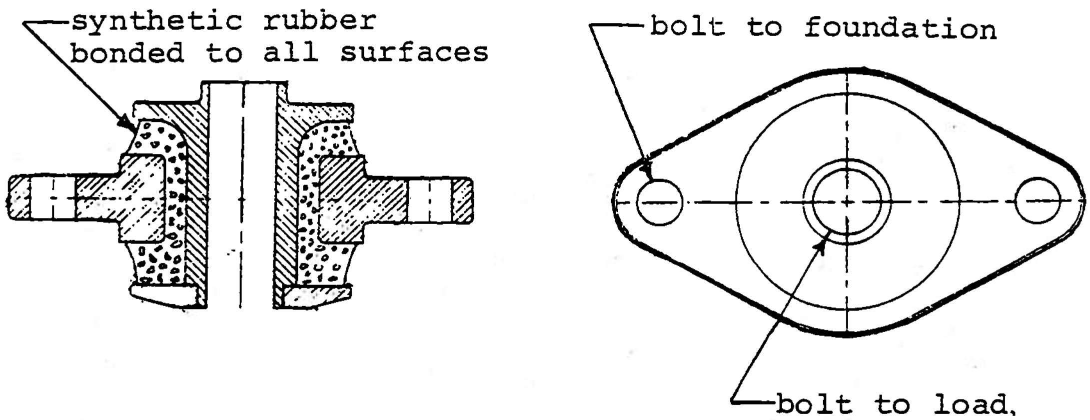


FIGURE 11.3 Flexible Support - Bonded Spool Type

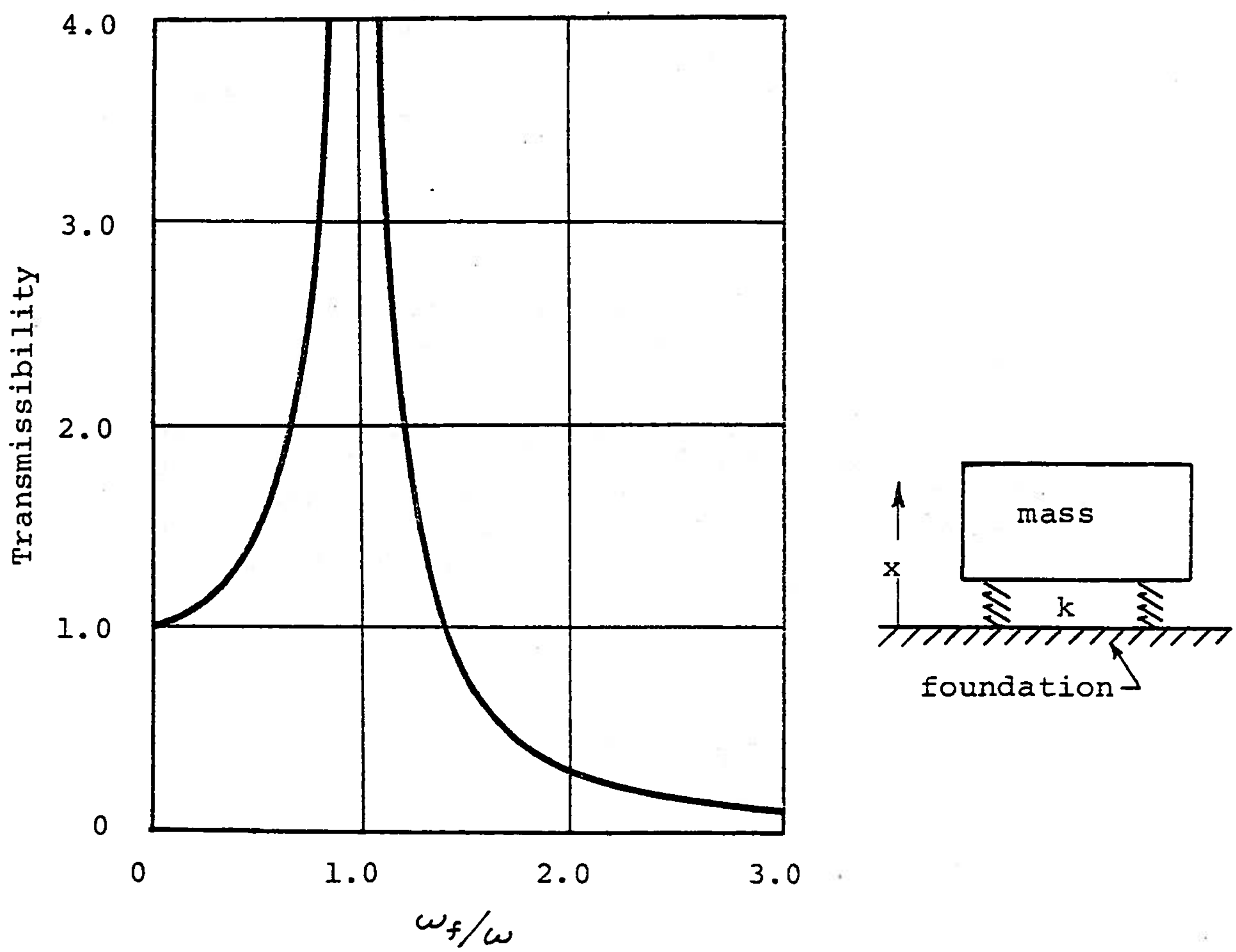


FIGURE 11.4 Transmissibility of Elastic Support with no Damping

where

$$\omega = [\frac{k}{m}]^{1/2} ,$$

the natural frequency of the spring-mass combination. The force f that the springs exert on the foundation is their deflection times their spring constant. For the maximum f, the ratio of f to F_0 is

$$\frac{f}{F_0} = \frac{1}{1 - (\frac{\omega_f}{\omega})^2} \tag{11.9}$$

This ratio is the <u>transmissibility</u> of the springs. Note that if $\omega_f = 0$, i.e., F is a static force, then all of it is transmitted to the foundation. Likewise if the springs are stiff enough that $\omega >> \omega_f$. On the other hand, if the springs are soft, so that $\omega << \omega_f$, then very little of F is transmitted. If $\omega = \omega_f$, then resonance occurs, and unless there is a lot of damping, or some additional restraint on the motion (as there always is in practice), the mass bounces wildly and exceedingly large forces are transmitted to the foundation. Figure 11.4 shows transmissibility as a function of ω_f/ω for zero damping.

A cure for a dynamic force is thus to place the engine on mountings having a spring constant related to engine mass in such ratio that $(k/m)^{1/2} << \omega_f$, where ω_f is the radians/second frequency of the dynamic force.

It is now appropriate to discuss this question: when is a force a dynamic force? Previous sections have discussed periodic forces (e.g., forces due to rolling), yet advised that they could be treated statically. The answer is this: a force must be considered to be dynamic if its frequency is near that of a natural frequency for hull vibration. The forces originating with vessel motion are of such low frequency that when transmitted to the hull via the foundation, they do not excite a hull response.

The actual magnitude of the dynamic force is usually of less interest than its frequency. Note from Figure 11.4 that at resonance, large responses can come from small forces, so that if a harmonic of the engine torque curve were to coincide with a hull harmonic, the flexible feet between engine and foundation might be needed. "Might be" because experience has shown that resonance between higher order excitations and higher vibration modes are usually pretty weak. Also, other remedies, such as a vibration damper in the engine might make the external impulses too weak to be troublesome.

Doubts aside, if flexible supports are to be used, they must be "soft," i.e., have low enough spring constant that their transmissibility is small for the dynamic force of interest. Naturally, they must transmit the dead weight of the engine, and being soft, will deflect significantly under this weight. This fact must be accounted for in planning the shaft system alignment.

11.6 STARTING

The internal combustion engine cannot simply be "turned on" because of its inability to run self-sustained below a certain minimum speed. For the reciprocating engines, leakages of several kinds are significant at low RPM. Leakage of cylinder charge past the piston rings reduces compression, and leakage of heat to the cylinder walls reduces the compression temperature that is essential to diesel operation. For the gas turbine, a basic defect is the inability of the turbine to develop power sufficient to drive the compressor when speed is low. In sum, all engines of the internal combustion engine type must be spun up to a self-sustaining speed by external means. Provision of the correct fuel-air ratio is also a necessary part of the starting process, as exemplified by the familiar choke on the spark-ignition engine. The major item so far as auxiliary systems is concerned, however, is the means of cranking the engine up to firing speed.

Small craft engines are typically started by a small cranking motor that engages a ring gear on the flywheel (or equilvalent, in the case of the gas turbine) quite in the fashion of an automotive starter. But the motor is not necessarily electric; pneumatic and hydraulic motors are also used.

The electric system, with storage battery as the source of energy, is common with all types of engines because electric power is nearly always required aboard for other purposes. The storage batteries are compact, though heavy. For propulsion plants above a few hundred horsepower, the weight and cost of the batteries becomes excessive compared to the alternatives. It also has a list of disadvantages that appear when conditions are unusual, the prominent ones being

1. Cold weather lowers the discharge capacity of the battery.

2. Bilge water and other sources of moisture are hazards to the electrical parts.

3. If the battery becomes discharged, recharging takes several hours, and at best is a great nuisance. In an emergency situation, it might be disastrous. If

the vessel relies on an engine-attached generator, then recharging is impossible save at dockside. Of course, if the vessel is twin-screw, or an independent battery is used for non-engine loads, then complete loss of stored energy is unlikely.

A pneumatic motor and its related system are much less likely to be affected by water, temperature, dust, stalling, and overloads than the electric starter, and higher cranking speeds are usually possible. On the other hand, the compressed air energy storage is bulky, and requires that there be an air compressor to charge the air flasks. Since pneumatic controls are frequently used with the propulsion plant, however, this component may be required anyhow. An advantage relative to emergency situations is the usual provision of a small hand pump by which the operator can pump up enough pressure by hand to get going.

Hydraulic starting has all of the advantages of pneumatic starting, and is somewhat more compact in its energy storage. Although some vessels will have a central hydraulic system (fishing vessels, in particular), hydraulic starting is usually accomplished by an independent system attached to the engine, and furnished with it. A pump is attached to the engine for charging accumulators from an oil reservoir, and an emergency hand pump is provided as in the case of pneumatic.

Gas turbines have requirements that do not trouble the reciprocating engines. They require a purge before starting to remove lurking vapors, and may have to be spun for even cool-down awhile after stopping. These actions are easily accomplished by running on the starting motor, but add significantly to the amount of energy that must be stored to operate the starter.

The designer's major decision in the starting area is usually which of the three common methods to employ, but he may also need to consider how much energy storage to provide. The answer depends on how much per start is required, and how many starts the system should be expected to provide without recharging. The former question can be answered only by data on the individual engine as stated by the engine builder. The number of starts will depend on the type of vessel, and the service intended, and hence falls to the designer's judgment. However, if the vessel is built to the rules of a classification society, e.g., American Bureau of Shipping, [5] the number of starts to be stored will be found in those rules. They can be used for guidance, even though the rules are not being officially used.

11.7 REFERENCES

1. Caterpillar Tractor Company, Cat Marine Engine Application and Installation Guide.

2. Detroit Diesel Division, General Motors, Marine Engine Installation Facts.

3. R. M. Cashman, "Design of Marine Machinery Foundations," Transactions, Society of Naval Architects and Marine Engineers, 1962.

4. J. P. den Hartog, Mechanical Vibrations, Fourth Edition, McGraw-Hill, 1956.

5. American Bureau of Shipping, Rules for Building and Classing Steel Vessels, 1970.

CHAPTER 12

GAS TURBINE AUXILIARY CONSIDERATIONS

12.1 INTRODUCTION

The gas turbine requires much the same auxiliary services as the other internal combustion engines: fuel, lube oil, intake of air, exhaust, etc. The only notable item missing is the engine jacket cooling water, although the lube oil from engine and reduction gear must be cooled. But with each system, points unique to the gas turbine should be noted.

The intake and exhaust systems, being so integral a part of the gas turbine installation, have been discussed in Chapter 5.

Special mention should be made of a handy reference,1, the Navy's Installation Design Criteria for Gas Turbine Applications in Naval Vessels. This document offers much detailed information on all aspects of marine gas turbine plants. Although specifically navy, and emphasizing ship-size machinery, it can be of great service to any designer working with marine gas turbine propulsion.

12.2 FOUNDATIONS AND SUPPORT

The foundation can be similar to that of the reciprocating engines. Although the gas turbine is a much lighter engine, torque and thrust loads must still be borne, and local stiffening of the hull by the foundation is still valuable. Nonetheless, the lightness of the engine and the absence of fluctuating engine torque may enable the designer to accomplish a generally lighter foundation. But as in Chapter 11, an exposition on structural design is considered beyond the scope of this text.

Support of the engine by the foundation may be quite different from that of the reciprocating engines. Because of the aircraft ancestry of some gas turbines, and because of the freedom from the high peak pressures of the other engines, the engine casing is typically of comparatively light construction, and consequently is unable to accept distorting and forces and moments. Coupled with this is a significant thermal expansion when heating up, found with this type of engine because it does not use jacket cooling; this is the very thing that would produce distortion if restrained. Unique suspension schemes have thus been developed. One of these is pictured in Figure 12.1. Three suspension points provide restraint each in only one or two directions. Generally the forward support allows axial sliding, while

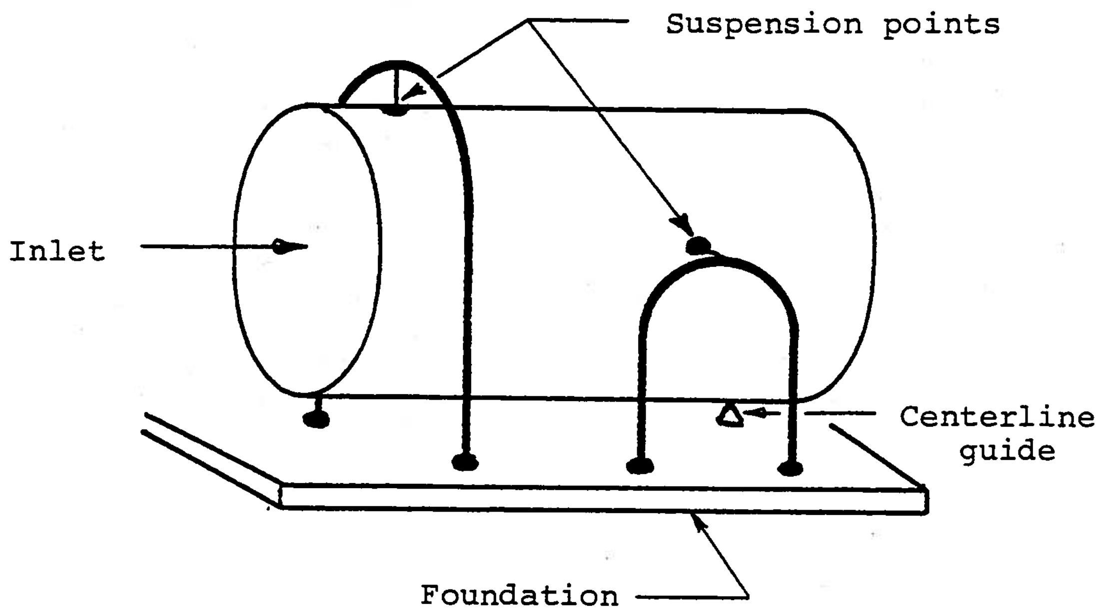


FIGURE 12.1 Gas Turbine Three-Point Suspension System

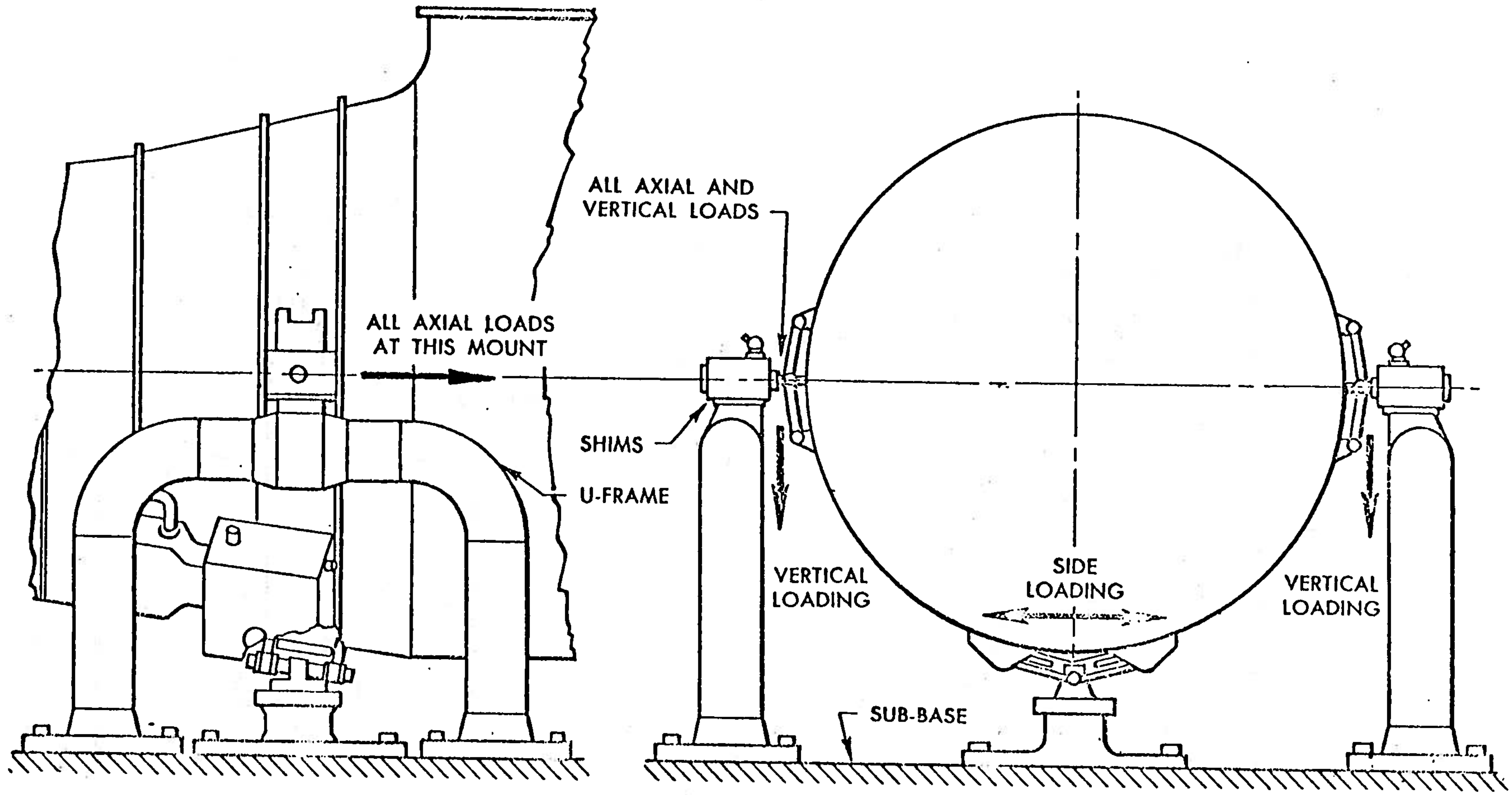


FIGURE 12.2 Example of Propulsion Gas Turbine Mounting (Aft) (Reference[1])

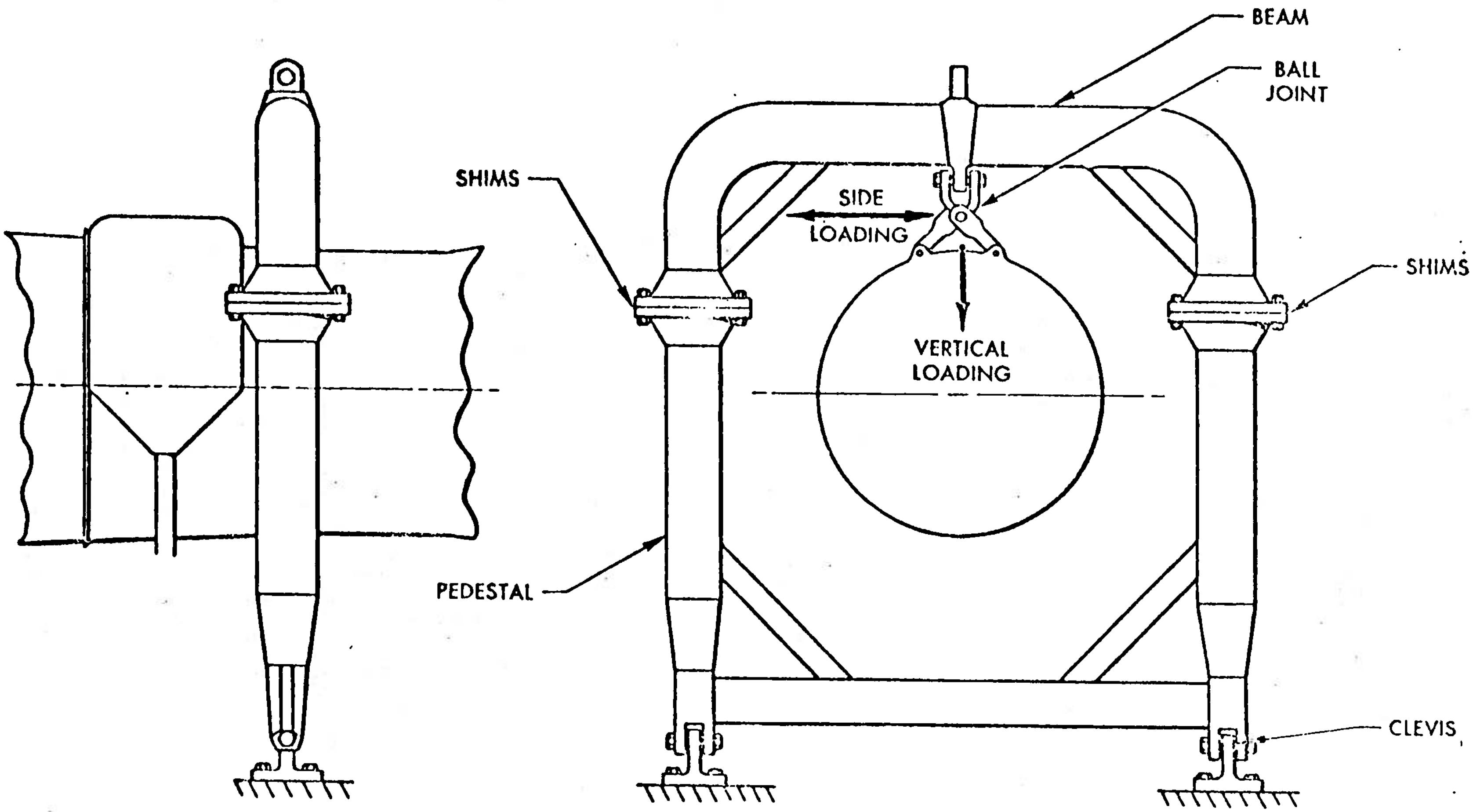


FIGURE 12.2 Continued, Showing Forward Mounting

the aft support allows sideways sliding. Thermal growth is also accommodated by linkages and ball joints connecting the engine to these supports. Figure 12.2 (from [1]) gives details of these connections.

At the lowest end of the power scale, say an automotive (e.g., Ford) turbine of several hundred horsepower that might be used in a pleasure boat, such elaborate suspension should not be necessary if the casing is comparatively rugged, and the support points on the engine are close together. It is reported (personal communication) that the Ford turbine is fastened rigidly to its foundation at its two aft mounting pads (look back at Figure 5.2: these are item C), and that the centerline forward pad (B in Figure 5.2) is attached to a plate that can slide axially, or to a rubber pad that can deform in shear to accommodate expansion.

12.3 FUEL SYSTEMS

It is essential that the fuel be delivered to the engine clean and water-free. Dirt of any kind is bad because by clogging fuel passages it can cause a maldistribution of fuel flow to combustors (if more than one is used), and thus a maldistribution of temperature in the turbine. The consequent hot spots can cause early blade failure. Water is bad because it is likely to be salt water, with the salt promoting hot corrosion. Also, water in fuel tanks promotes bacterial growth, with resultant slimes and gooey gunk that can clog filters.

Protection for the engine in the form of a filter in the fuel line is therefore always required. The "coalescent" type is being used in Coast Guard vessels and merchant ships since it effectively removes water as well as solid particles. It uses a filtering element such as glass fibers that cause water droplets in the passing oil to coalesce into larger drops on its fibers; when they become large enough, they drop off, collect in the bottom of the filter casing, and can be drained off. A second set of filter elements is usually in series with the first, it being of a material (e.g., like botting paper) that absorbs the remaining water, while allowing the oil to pass.

The materials used depend on the fuel to be used. If diesel oil is the fuel, ungalvanized steel is appropriate. If "jet fuel" (kerosene type distillate) is used, steel is not suitable because of this fuel's tendency to loosen rust and scale. Copper-nickel (70-30 preferred) is recommended, [1] it also being acceptable for diesel oil. Tanks to contain jet fuel are said [1] to require internal coating with epoxy or urethane.

Fuel oil is used as the lube oil cooling medium in aircraft practice, and this scheme can also be used in marine practice, but not with diesel oil. The rather high (> 300F, perhaps) temperature of the lube oil may cause coking of diesel fuel, with subsequent distress to the fine passages of the combustors.

12.4 LUBE OIL

Like the other internal-combustion engines, the lube oil system is usually an integral part of the engine, but a cooler may need to be supplied externally with sea water or other cooling medium. If the oil is a synthetic one, as it may have to be because of the high temperatures encountered, cooling is said [1] to cause problems because of material incompatibility. The synthetic oils react with copper, so that the usual copper alloys for sea service can't be used for tubes in the cooler, though monel is possibly satisfactory. These are alternatives, however. Fresh water can be used as the coolant, with subsequent transfer to the sea, though this means the complication of an extra fluid system with its pump, etc. Another alternative is to transfer the heat to the reduction gear lube oil, with it then being cooled in a sea-water heat exchanger. Because it is likely to be a mineral oil, it will not have the materials problem in its cooler. Use of the fuel as the cooling medium has been mentioned in the preceding section. Air cooling is also feasible, though the heat exchanger is bulky compared to a liquid-cooled unit.

12.5 ENGINE WASHING

An auxiliary system unique to marine gas turbines is the water-washing arrangement that has been installed on many vessels. This is a simple fresh water supply ending in nozzles that spray directly into the compressor suction opening. It is used while the engine is being spun on its starter to wash out salt deposits. Some vessels have apparently found this accessory to be quite necessary; recall that Chapter 5 has cited a published report of severe loss of performance of a small-craft turbine, essentially all due to this easily removed deposit. However, the washing system has not been universally used with small craft, some apparently relying on well-protected intakes with demisting filters to keep the salt out. But since most small gas turbine craft must be regarded at this time (1971) as experimental, I would judge that too little experience is on hand to say whether washing systems can be safely omitted. Of course, portable washing schemes (garden hose?) may be a feasible compromise if some washing is found to be necessary.

12.6 ENGINE COMPARTMENT COOLING

All of the internal-combustion engines require a flow of air in excess of actual engine intake in order to keep the machinery space at a reasonable temperature. The gas turbine differs only in that a greater flow is likely to be required because of the hotter surfaces, it not having the benefit of cooling water passages like the other engines. Application data sheets on several ship-size gas turbines show temperatures near 1200F on their surfaces near the combustors. Insulation of the engine itself is frowned on because it would make the metal surface even hotter, but insulation on compartment boundaries is doubtless essential for both temperature control and sound reduction. Even though the intervening space may be kept fairly cool by a good circulation of air, the components within the space may be severely heated by direct radiation from the engine. Heat shielding or placement outside may be necessary for heat-sensitive items.

It is difficult to say how much air should be circulated for cooling, since it depends on things that are unique to the particular installation. A rule-of-thumb which I picked up from application literature on a ship engine says about 10% of the engine air flow should be added for compartment cooling. The flow can be effected by a blower, or by an eductor such as pictured in Chapter 5. In the latter case, a small blower to remove residual heat after shutdown may be needed, though with a small vessel merely opening the engine hatches should be sufficient for this.

12.7 REFERENCES

1. U.S. Navy technical manual, Installation Design Criteria for Gas Turbine Application in Naval Vessels, NAVSHIPS 0941-038-7010.

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