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ANALYSIS OF STEAM PROPULSION PLANTS

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THE DEPARTMENT OF NAVAL ARCHITECTURE AND MARINE ENGINEERING

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MARINE ENGINEERING

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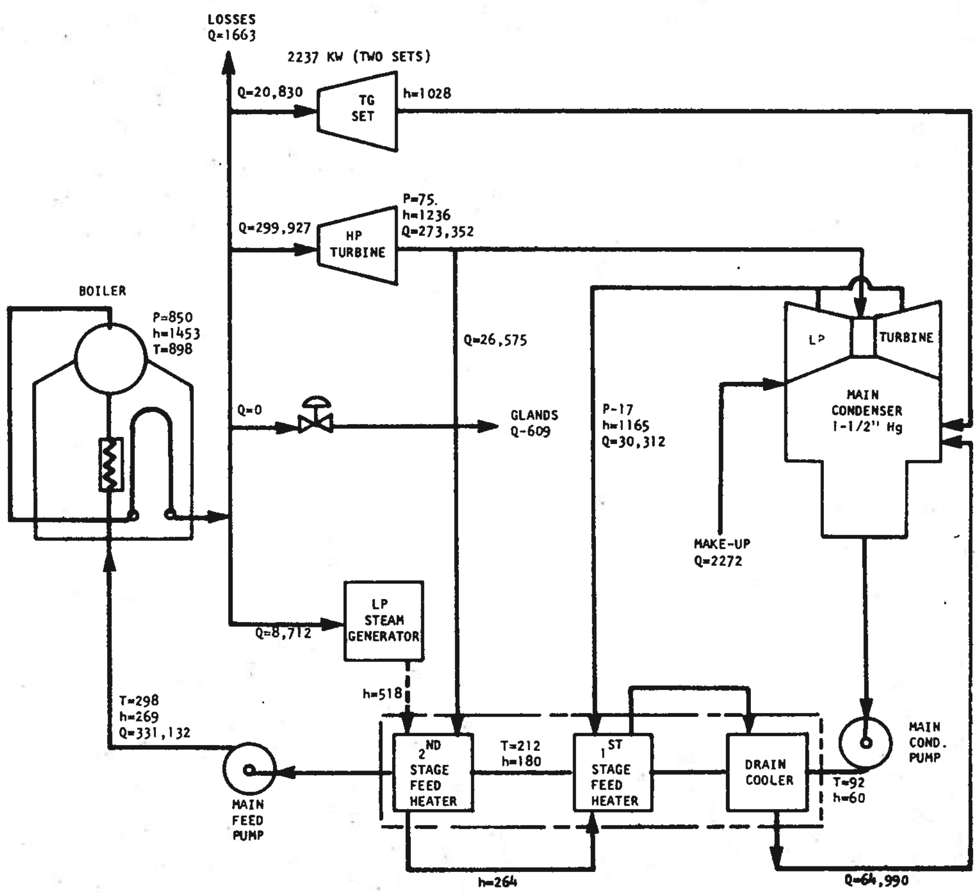
Dedicated to that unknown genious who made the first steam whistle

John B. Woodward



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PREFACE

These notes are intended to help University of Michigan students of marine engineering to learn how to do steam propulsion plant heat balances. They are not intended to compete with the SNAME heat balance bulletin, but rather to cover general principles and calculational techniques that that bulletin neglects because it is not an instructional tool for the uninitiated.

SI units are used. Necessary steam properties data are taken from Keenan, Keyes, Hill, and Moore: STEAM TABLES, Thermodynamic Properties of Water, including Vapor, Liquid, and Solid Phases, John Wiley & Sons, 1969.

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CHAPTER 1

FUNDAMENTALS OF THE STEAM PROPULSION PLANT

1.1 INTRODUCTION

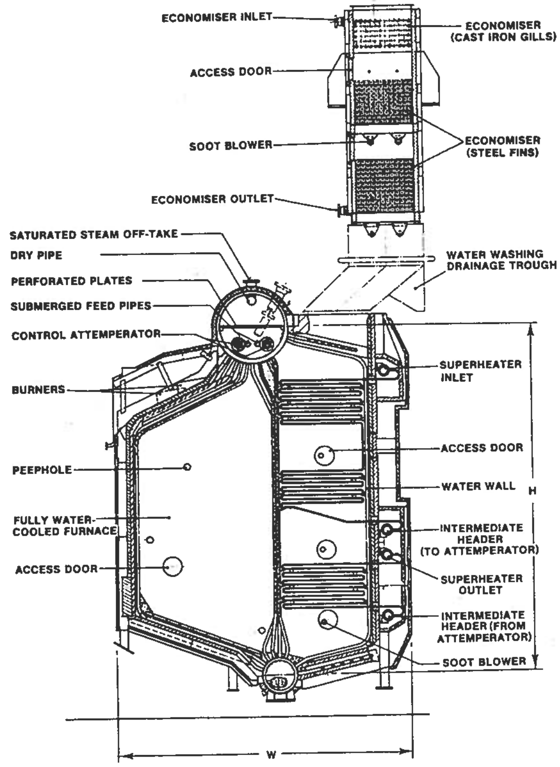
The main purpose of this work is to introduce the student to the process of analyzing the steam propulsion plant - to the complex process of the heat balance. However, it is obvious (even to the professor) that you can't very well analyze something you don't understand, so this chapter comes first with some description and other introductory material.

Now, it would be a happy thing if the problem of description could be disposed of with a picture or two of the steam propulsion plant. It can't, however, because no picture can grasp all of it; the plant is a complex array of many major and minor components, interconnected by a web of piping and wiring. These components are often spread through several distinct spaces (e.g. engine room, boiler room, auxiliary room) of the ship, so that no one picture can show more than a few components, and a collection of pictures of components cannot convey the relationships among them. This can be done by means of block diagrams showing the principal connections, often called "heat balance diagrams" because they show the results of the heat balance process, i.e. the flows of working fluid along the connections, and the fluid properties at significant points. Further understanding can be gotten from arrangement drawings, although they usually do not show connections, nor depict functioning. The two types taken together, however, give you about as much as you can hope to get from illustrations. Examples of each are included in this chapter.

Understanding of the components, being familiar with functions, sizes, appearances, performance parameters is also a vital part of understanding, although you need only rudimentary knowledge of these things to perform the heat balance analyses.

Exploring the components in depth is well beyond the scope of this modest work; it takes a whole fat book, so I advise you to consult the fat book (Reference 1). On the other hand, I can't resist putting in a few pictures of the two major items - the boilers and the propulsion turbines. Figures 1.1 and 1.2 show a typical marine boiler of recent design, the first being a sectional drawing and the second a pictorial cutaway view. Actually, it's not the same boiler in both pictures, but they are very similar. Figures 1.3 and 1.4 are pictorial cutaway views of propulsion turbines. Both include the reduction gear and show parts of the foundation. Figure 1.4 also includes the main condenser in its usual position below the low-pressure turbine.

I should emphasize that the complexity of the steam plant leads to a seemingly infinite variety in the details of the many actual examples. It is fruitless and wasteful to attempt a description of all the possibilities; this chapter must be limited to the most typical cases, plus a brief discussion of the many feasible alternatives. Although all of this makes the student's path to understanding more difficult, don't look upon it as a curse. I believe that the fact that steam propulsion calls upon the marine engineer to exercise his creative skills in blending up a competitive propulsion plant accounts in part for its survival against the powerful arguments in favor of the diesel. For instance, in the late 1960s steam enjoyed a revival, since it is superior in some ways to diesel in the high-power propulsion plants then being demanded. But the revival was possible only because steam survived through many lean years when it seemed that only the pig-headed prejudices of the North American marine engineers was resisting universal triumph of dieselization. The humorless Teutons of Sulzer, MAN, B & W, et al, had (and still have) a splendid product, but as Dr. Superheat observed in <u>The Diesel Engine: To Drive a Ship</u> (NA & ME Department Report 105) "You don't love a diesel engine - you respect it." Whether said pig-headed prejudice in favor of steam constitutes love is not easy to prove, but I believe it at



TYPICAL ARRANGEMENT OF E.S.D.III MARINE BOILER

E.S.D. III Double cased boilers—steam outlet 63Kgs/cm²—515C°

Approx. Evaporation tonnes per hour pounds per hour	Approx. weight tonnes (dry) excluding econ.	Approx. weight economiser tonnes (dry)	W mm ft	H mm ft	Length over casings
30—45 66 000—100 000	125	35	6 200 20′ 6″	7 300 24′ 0″	5 520 18′ 6″
45—65 100 000—143 000	140	45	6 700 22′ 0″	7 150 23′ 6″	6 100 20′ 0″
65—80 143 000—176 000	160	60	7 500 24′ 9″	7 700 25′ 4″	6 300 20′ 9″
80—100 176 000—220 500	200	80	7 700 25′ 4″	8 650 28′ 6″	6 650 21′ 10″
100—120 220 500—264 000	270	90	8 100 26′ 8″	8 700 28′ 8″	7 500 24′ 9″
120—145 264 000—320 000	310	110	8 600 28′ 4″	8 700 28′ 8″	8 300 27′ 4″
145—160 320 000—353 000	330	125	8 750 28′ 10″	9 400 31′ 0″	8 500 28′ 0″
160—180 353 000—397 000	360	150	8 850 29′ 2″	10 000 33′ 0″	8 700 28′ 8″
180—200 397 000—440 000	450	170	8 950 29′ 6″	10 950 36′ 0″	8 800 29′ 0″

N.B. This range of boiler is also available in Monowall construction.

FIGURE 1.1 Boiler

A modern Babcock & Wilcox marine radiant boiler

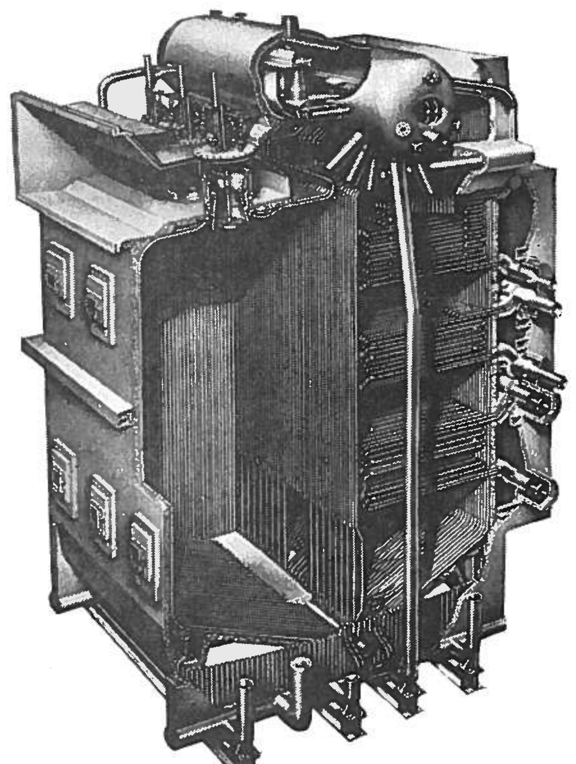


FIGURE 1.2 Boiler

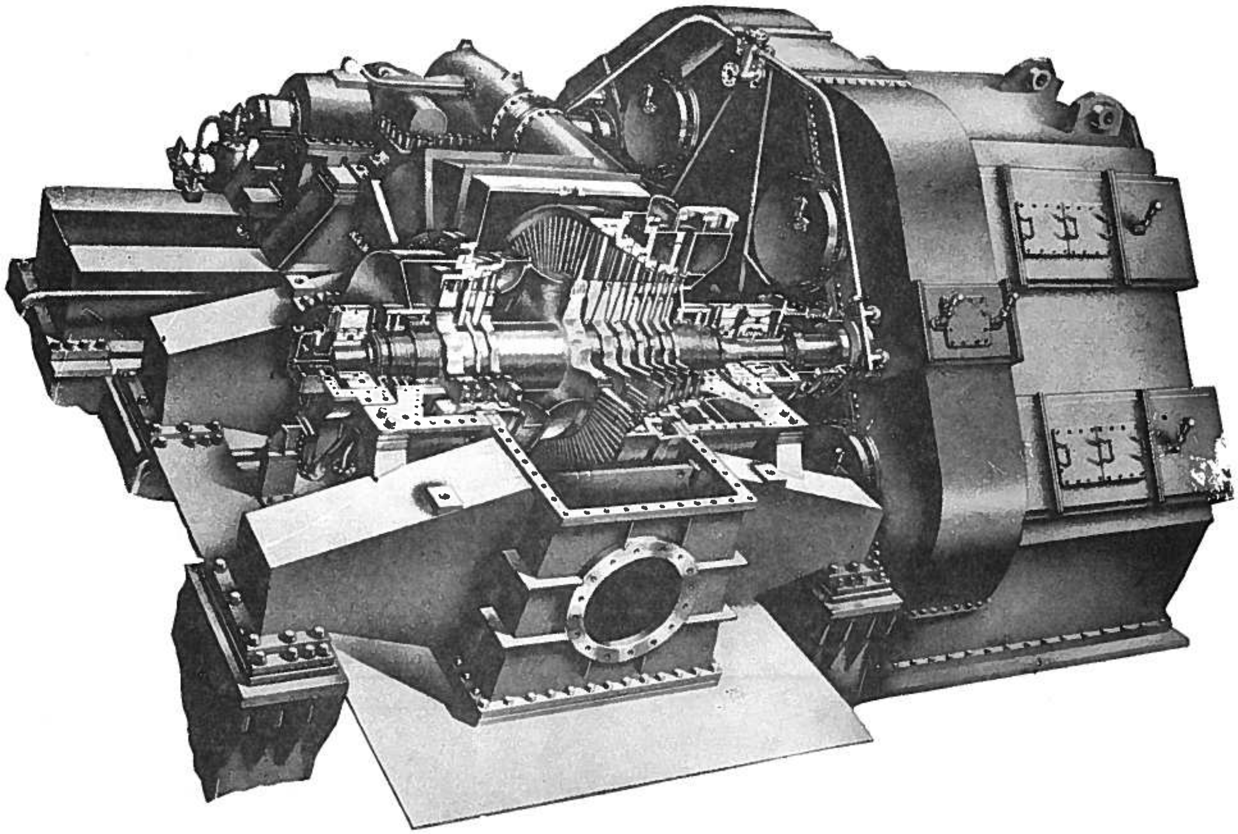


FIGURE 1.3 Turbine

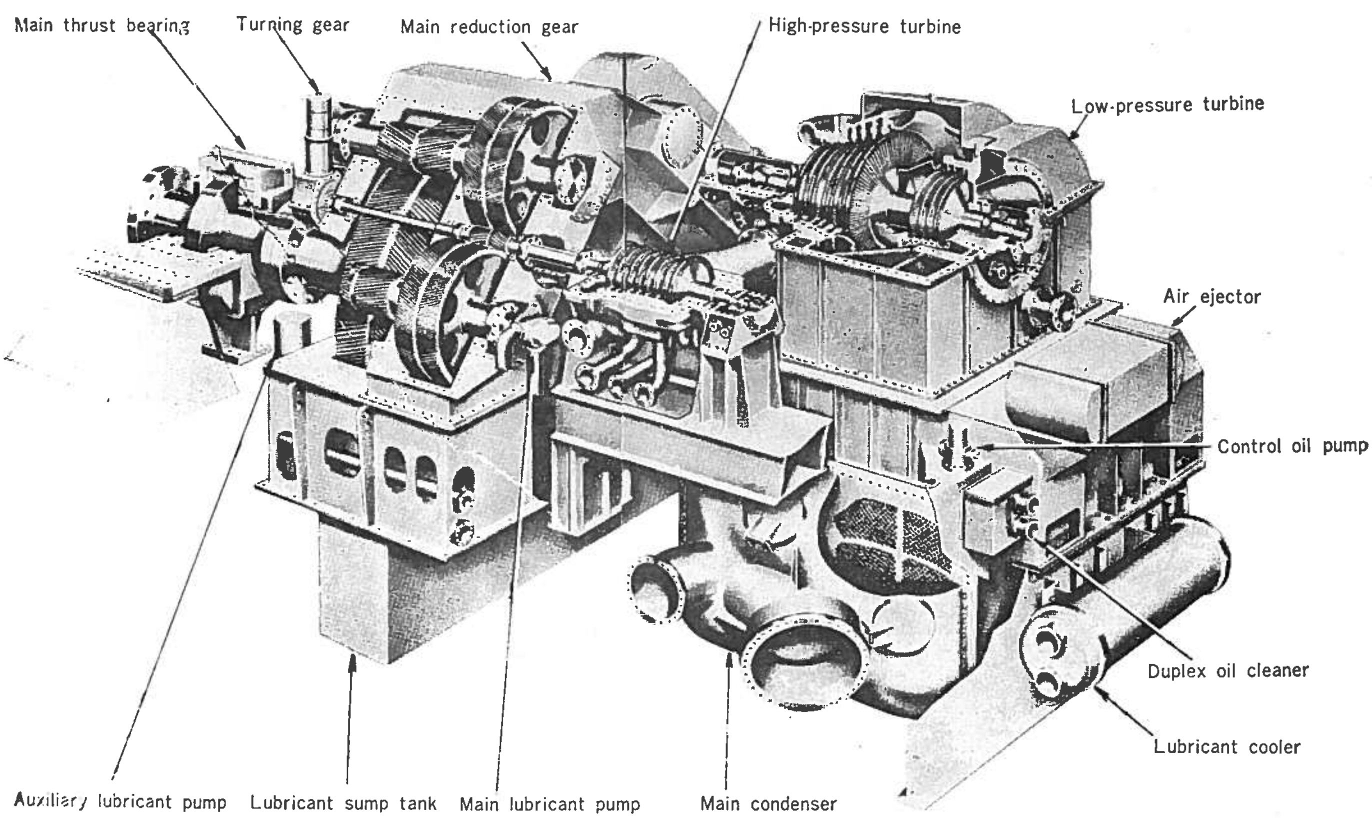


FIGURE 1.4 Turbine

least involves appreciation of what in recent years has been called "soul." Steam has it, diesel (and gas turbine, too) don't.

As this is written in 1980, steam is on the wane again because high fuel prices favor the more efficient diesel. However, steam has an inherent advantage over diesel in that it can be fueled by any combustible material. As petroleum fuels become more scarce, steam may rise once more, riding a wave of that ancient favorite, coal (wood, nuclear? unlikely, I'd say). Meanwhile, there is the possibility of using exhaust heat from a diesel, or even a gas turbine, to make steam to augment propulsion power. So don't go away!

1.2 THE TYPICAL STEAM PROPULSION PLANT

The major components of a steam propulsion plant are the boiler (one or two for each propulsion shaft), the engine (turbine, nearly always), a condenser, and a feed pump to return condensed liquid to the boiler. Usually the returning liquid is preheated in one or more feedwater heaters, this effecting the regeneration necessary for good efficiency. In addition, auxiliary devices are needed to support these devices in a practical working plant. The propulsion turbine requires a lube oil system including pumps, coolers, purifiers, and storage tanks; also a gland seal and leak-off system to prevent leakage of steam and air through the casing at the shaft penetrations. The boilers require a fuel system, with pumps and heaters; a draft system, with blowers and possibly preheaters to furnish combustion air; automatic (usually) controls for regulation of air, fuel, and feedwater; desuperheaters to furnish low-temperature steam to auxiliary trubines; and soot-blowers to clean tube surfaces. The condenser requires air ejectors or vacuum pumps to maintain its vacuum, condensate pumps to remove the condensed steam, and a circulating system to furnish cooling water. Further, there are many uses of electricity, both within the plant and throughout the ship, so that generators, usually driven by their own turbines, are needed. A source of highquality water is needed to replace that lost from the power plant, and usually a distilling plant powered by auxiliary exhaust steam or extraction steam is installed for this purpose, and also to furnish potable water. And further yet, there are items that have nothing to do with propulsion, but which are related to the power plant because they draw their energy from the boiler. Such might be turbine-driven cargo oil pumps (on tankers) or fire pumps, and domestic heating systems. Don't forget the whistle. And all electrical driven equipment is related to the power plant because it draws its power from the same generator that powers propulsion auxiliaries.

Figure 1.5 pictures a workable steam propulsion plant in the format of a heat balance diagram, i.e. a diagram showing the components in block form, plus the water/steam flow paths. Brief explanations are given in a table that forms part of the figure. Items that would not be of direct concern in a heat balance are not shown, though the principal omitted ones are listed in the table.

Let us repeat and emphasize that great variety is found among steam plants, so that Figure 1.5 is only a sample. It is too gruesome a task to detail and explain all the possibilities, but below are listed a few of the major alternatives to be considered in laying out the cycle.

(1) What steam pressure and temperature? The higher the pressure and temperature, the higher is the efficiency, but these tend to be standardized by state-of-the-art and economic balance. Boiler pressures of 40 to 60 bar (4.0 to 6.0 MPa) have been typical recently. Steam temperatures are in the 500C range. Temperatures above this range are impractical with the boiler fuels currently used, because the superheater tubes slag badly if their outer surfaces exceed about 600C.

(2) How many stages of feed heating, and how should they be distributed over the available temperature range? Let's answer the second one first: the greatest benefit from

regenerative heating is obtained from N heaters if each heater heats the feedwater [1/(1 + N)][ΔT], where the last quantity is the difference between boiler saturation temperature and condenser hotwell temperature. As to what N should be, this is nominally established by a balance between the increasing efficiency with higher N, and the increasing first cost and complexity that more heaters introduce. A naval plant, where light weight and compactness are essential, usually has only one heater, and that provided more for deaeration than feed heating (see more discussion under (3)). A merchant plant usually has either two heaters with a resulting feed temperature of about 140 C, or four heaters with resulting feed temperature of about 220 C. The latter is more efficient, but the former allows part of the feed heating to be done by a boiler economizer, thus helping boiler efficiency but avoiding the corrosion problems of an air heater. The four-heater cycle usually employs a gas air-heater (rotary-regenerative type, most often) for boiler heat recovery, and generally results in a higher-efficiency plant than the other. Combustion air for the boiler is usually heated with extraction steam in the two-heater cycle, and this also is a regenerative feature.

(3) Direct-contact heaters or surface type? From the efficiency standpoint, direct-contact is preferred, because it does not require the temperature difference between steam and water found with the surface type. But a pump is required to move the feed on to the next heater, which inevitably must be at higher pressure. In the surface heater, there is no pressure relation between water and heating steam, so that water pressure in the first heater can be higher than boiler pressure, if desired. One direct-contact heater is nearly always used,* however, because it is vital

*In the few cases where this heater is not used, deaeration must be accomplished in the condenser.

that dissolved oxygen be removed from the feedwater to protect the boiler from corrosion, and this task is accomplished with relative ease by spraying the water through a steam chamber. By this process the water is provided with a large exposed surface, and is heated to saturation temperature, thus expediting liberation of gases. This is a direct-contact heating process, and so it is that deaeration needs demand a direct-contact heater. This heater always serves as a surge tank also for the feed flow. It is commonly known as the <u>deaerating</u> <u>feed</u> <u>tank</u> ("DFT") to express its deaerating and tankage functions.

(4) How to remove air from the condenser? Since the condenser operates under vacuum, continuous air removal (there are always leaks) must be provided. The traditional method is by steam-jet air ejector, with the jet steam plus any drawn from the condenser being condensed by the feedwater. An alternative is the motor-driven vacuum pump. The latter is preferred when remote starting and operation ("automation") is sought.

(5) Shall pumps be motor-driven or steam driven? Small turbines are poor in efficiency, whereas small motors are quite good in this respect. Hence motor drive is generally preferred from this standpoint, and also because remote control is simpler. Feed pumps are nearly always turbine driven, however, because of their need for speed adjustment, and because this liberates the vital boiler-feed function from dependence on the electrical generating and distribution systems. Naval steam plants usually apply turbine drive for all vital pumps, such as condenser circulating, condensate, lube oil, fuel oil, and forced draft blowers. A third alternative is to drive some of these from the propulsion turbine, which has a much higher efficiency than any auxiliary turbine. Separate drivers must be provided for periods when the propulsion turbine is not turning.

(6) How shall contaminated services be handled? "Contaminated" services are those, such as fuel oil heating, that may leak oil into the steam, and thereby cause fouling of boiler tubes. A popular method of avoiding boiler trouble from this source is to have a separate steam generator that is heated by the main steam; the "contaminated steam generator" is thus a tank fitted with a submerged steam heating coil. An alternative is simply to be careful: lead the condensate from possibly contaminating services back via an inspection tank so that the crew can divert oily water before it reaches the boiler.

(7) How about reheating: Reheating is a process of returning steam to the boiler for restoration of its temperature after it has passed part-way through the propulsion turbine. This is an alternative that is rarely used, but has the appeal of high efficiency, perhaps even rivalling diesel. Figure 1.7 illustrates with a heat balance diagram.

(8) What uses shall be made of extraction steam, besides feed heating? Miscellaneous steam services, such as fuel oil heating, steam atomization of fuel oil, cargo heating, and ship heating can all be served with extraction steam, with some small efficiency benefit from regenerative effect, or they can be served directly by a low-pressure steam supply, which may be the contaminated system, if one is installed.

(9) What to do with the drains (condensed heating steam) from feed heaters? Nominally, they can "cascade," i.e. flow to the next heater where the shell (steam side) pressure is lower. Fine, but the first heater then can drain only to the condenser (since only there is the pressure lower), but there it will give up its sensible heat to the sea. In order to avoid this waste, a pump is sometimes provided to send the first heater drains to the DFT. Or they may drain by gravity to a collecting tank, whence they are pumped to the DFT along with other collected drains.

═══	Superheated Steam
▬▬▬	Desuperheated steam
▨▨▨	Feed & condensate (main)
⁄⁄⁄⁄⁄	Aux Condensate or other water
———	Exhaust or extraction steam
- - - -	Mechanical

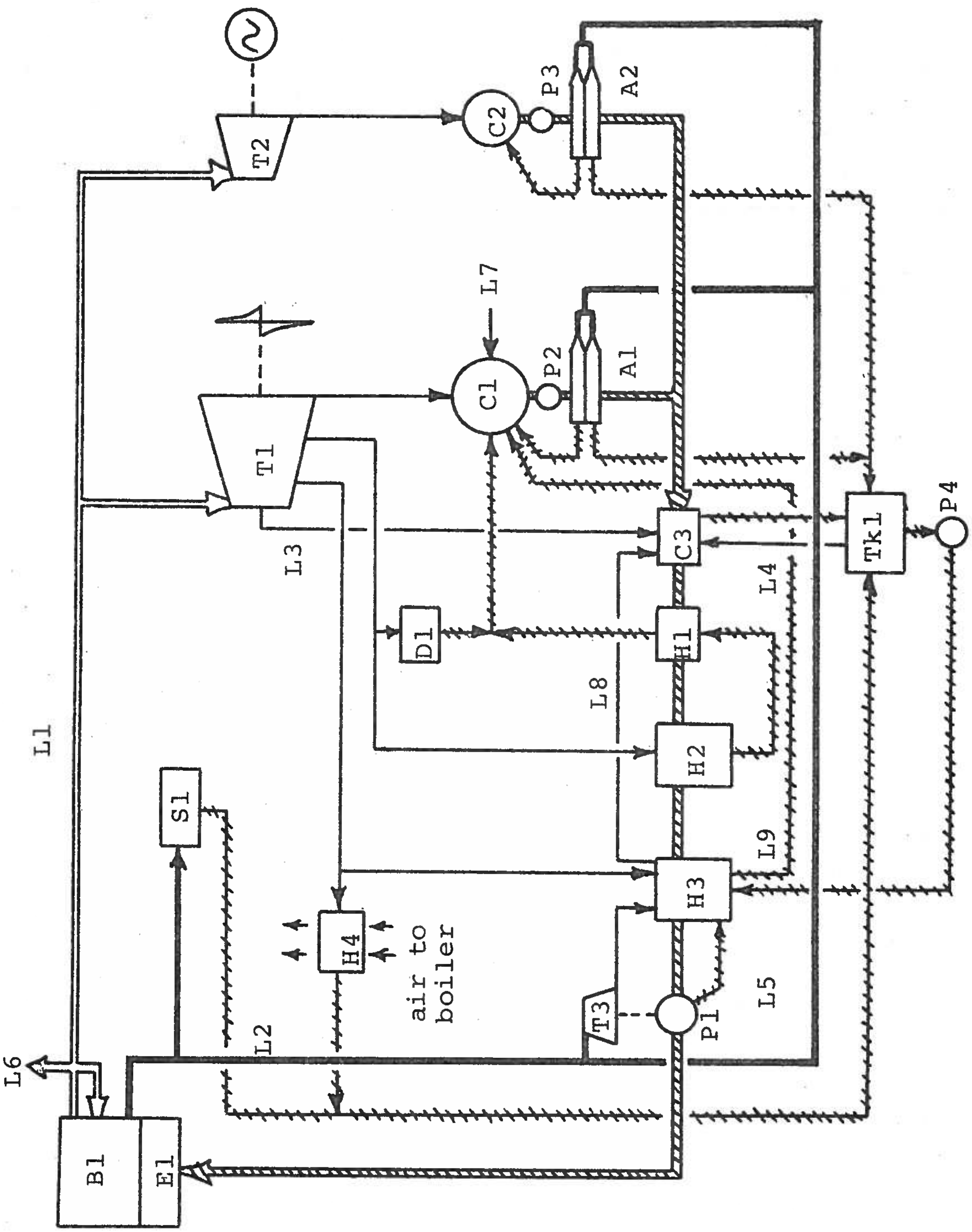


FIGURE 1.5 Typical Steam Propulsion Plant, Shown by Heat Balance Diagram

SYMBOL	EXPLANATION
A1, A2	Air ejectors for main and auxiliary condensers, and their condensers. Two stage. Motor-driven vacuum pumps sometimes used instead
B1	Boilers. Two per shaft most likely
C1, C2	Main and auxiliary condensers
C3	Gland leak-off condenser. Used to condense gland leakage from turbines and vents from H3 and Tk1. Sometimes combined in same shell with air ejector condensers
D1	Distilling plant. Makes fresh water for make-up feed, and domestic uses
E1	Economizer. Pre-heats incoming feedwater to boiler by cooling flue gas, thereby increasing boiler efficiency
H1	Drain cooler. Saves the thermal energy in drains coming from H2 - would otherwise be rejected to sea. May be in same shell with H2
H2	Feedwater heater, surface type
H3	Feedwater heater, direct contact type. Also serves as deaerator and surge tank
H4	Steam air heater. Preheats combustion air
L1	Superheated steam main
L2	Desuperheated steam main
L3	Turbine gland leakoff. Similar line from T2 is not shown
L4	Flashed vapor from Tk1 to C3
L5	Feed pump recirculation
L6	Represents losses and all non-return use of steam
L7	Make-up feed to replace losses, etc.
L8	Vent (air-steam mixture) from H3

FIGURE 1.5 continued --Key to the Components and Connections

L9	Recirculation from H3 to C1 and C2 (not shown). Operates at low loads so that A1, A2, C3 will have sufficient cooling water, and so that hotwell levels in C1 and C2 will be kept up
P1	Feed pump
P2, P3	Condensate pumps
P4	Drain pump for Tk1
S1	Represents all of the miscellaneous steam services, such as fuel heating, space heating, laundry, etc.
T1	Propulsion (main) turbine. Usually cross-compound, i.e. arranged in two separate (high pressure, low pressure) casings in series, but this detail not indicated here
T2	Generator turbine
T3	Feed pump turbine
Tk1	Drain tank
	The following important components are not shown on the diagram because they do not handle the working fluid
H5	Lube oil cooler, heat from main turbine and reduction gear bearings
P5	Main circulating pump. Sea water to C1 and H5
P6	Aux circulating pump. Sea water to C2
P7	Fuel oil pump
P8	Lube oil pump. L.O. to main turbine and reduction gears
P9	Forced draft blower. Combustion air to boiler

FIGURE 1.5 continued

(10) What condenser vacuum? This question parallels that concerning boiler pressure and temperature; higher the vacuum, the better will be the efficiency, but the bigger must be the condenser. Merchant practice is standardized to a 28-1/2" Hg vacuum. Navy is typically 25" Hg at full power. (These are about 0.005 MPa and 0.017 MPa, respectively.)

1.3 SAMPLE HEAT BALANCE DIAGRAMS

Figures 1.6 and 1.7 are sample heat balances, or at least the results thereof presented in graphical form (from General Electric Co.). Figure 1.7 includes a special feature that should be pointed out: reheating of the propulsion steam. You will notice that steam flows back to through the boiler after partially traversing the propulsion turbine. The practice is common in high-efficiency shore plants, but is relatively rare afloat.

1.4 SAMPLE ARRANGEMENT PLANS

Figure 1.8 is a set of arrangement drawings for a steam propulsion plant, taken from Reference 2. Since steam plants vary in their arrangements, this is offered only as a sample, though in many ways it is typical.

There are some principles to be followed in making an arrangement. A discussion of them is beyond the scope of this work, but I can give you a few references. For example, the chapter on piping in Reference 1 treats machinery arrangement. There is also a set of unpublished notes by Harold Johnson of Newport News Shipbuilding (Reference 3); I have a copy.

1.5 REFERENCES

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

2. T. B. Hutchinson, "30,000 shp Unitized Reheat Steam Turbine Propulsion," Transactions, Institute of Marine Engineers, Vol. 78, Nº 4, 1966.

3. Harold Johnson, "Machinery Arrangement Notes," unpublished.

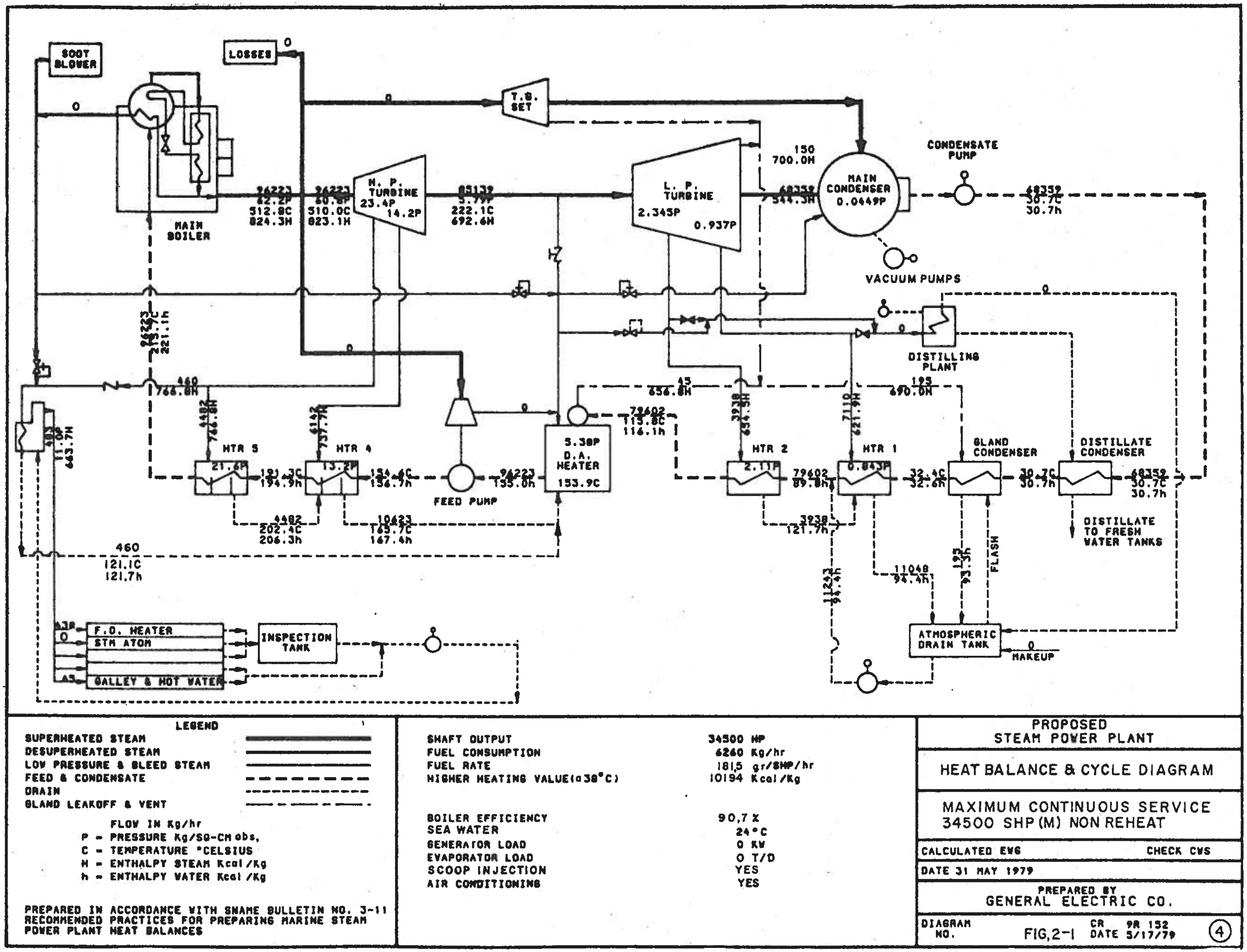


FIGURE 1.6 Typical Heat Balance Diagram (courtesy General Electric)

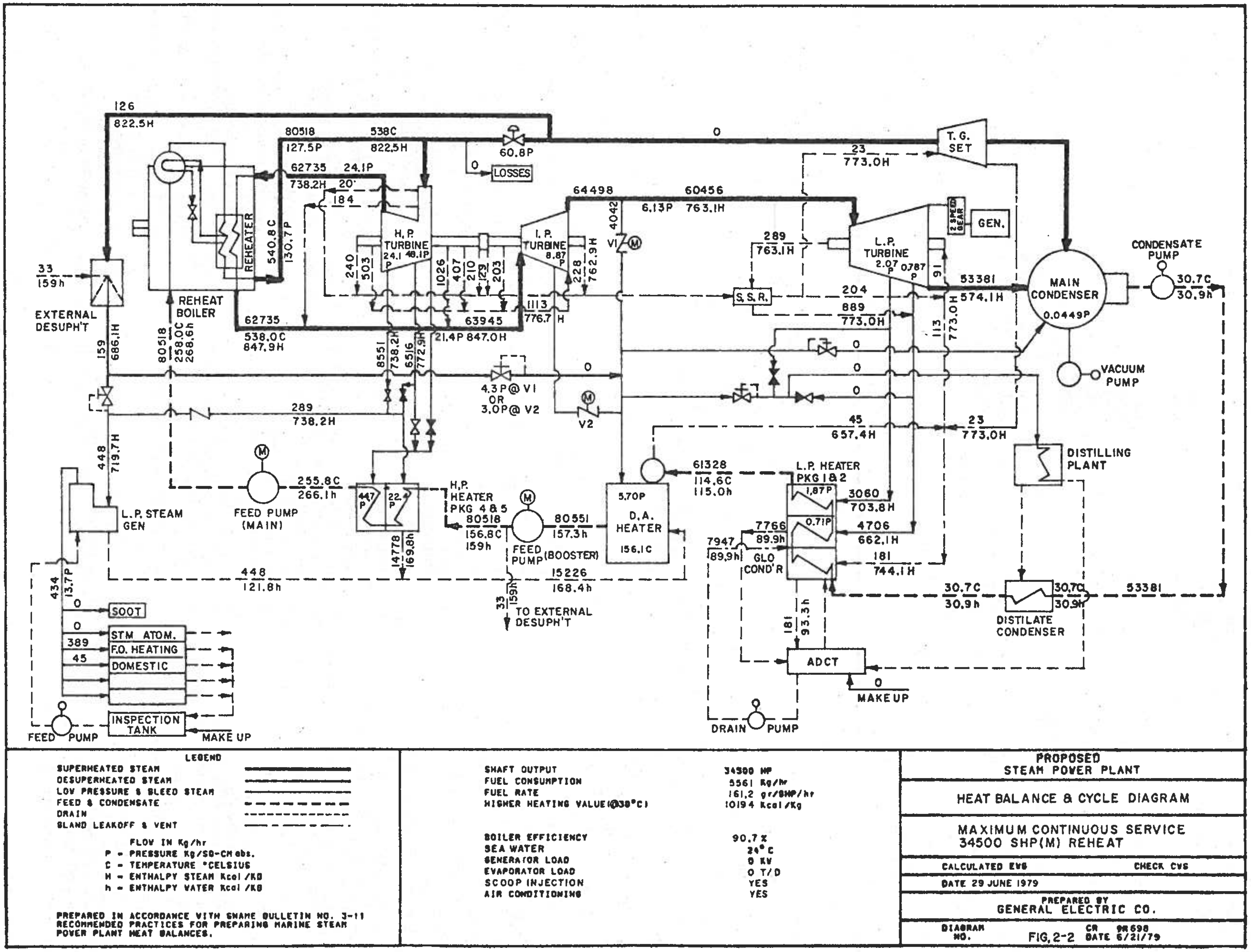
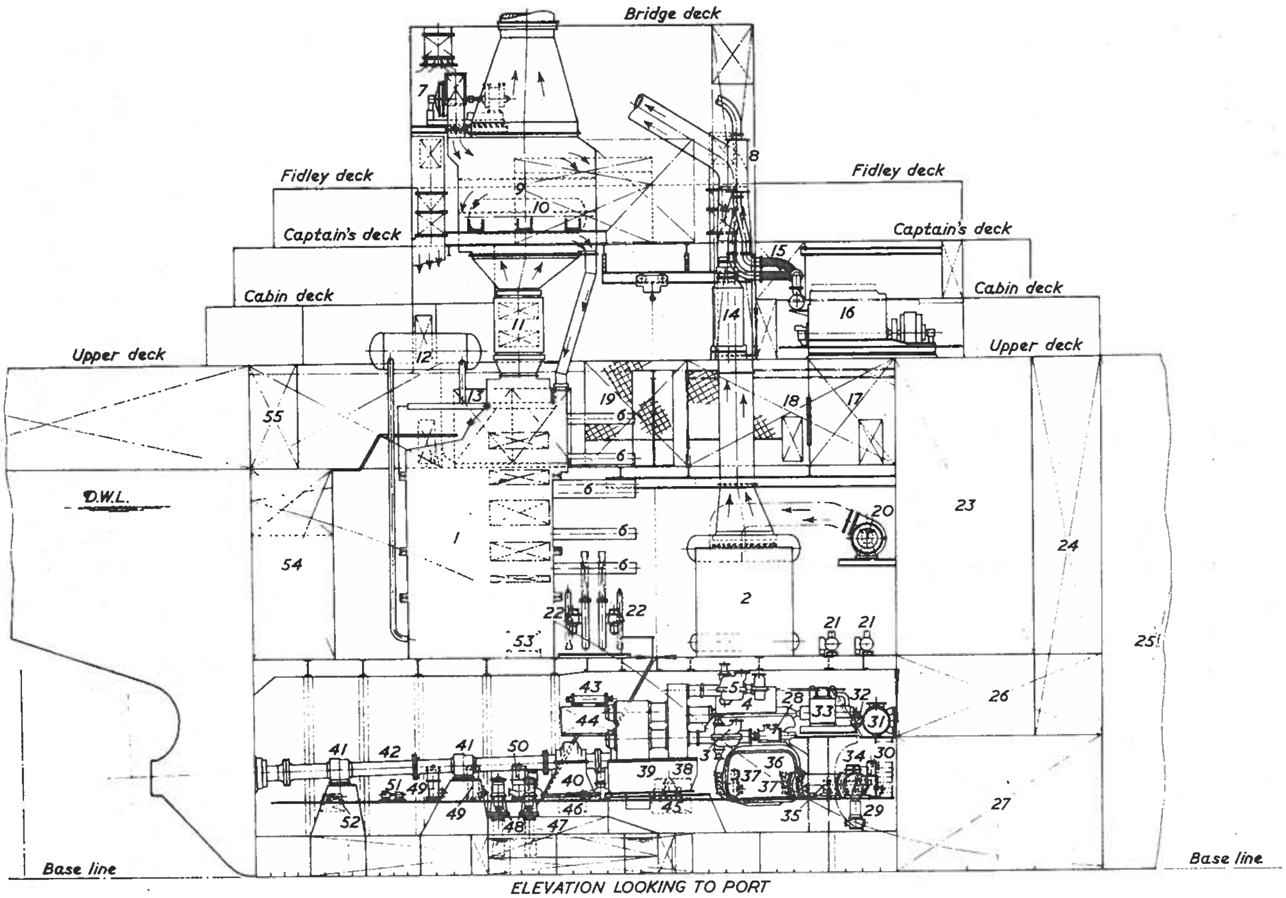


FIGURE 1.7 Typical Heat Balance Diagram for a Reheat Cycle (courtesy General Electric)



1) Main boiler
2) Auxiliary boiler
3) H.P. turbine
4) L.P. turbine
5) I.P. turbine
6) Retractable soot blower
7) Main boiler F.D. blower
8) Diesel generator exhaust silencer
9) Air-heater
10) Steam/steam generator
11) Economizer
12) Main boiler steam drum
13) Lubricating oil storage tank
14) Auxiliary de-aerator
15) Diesel generator lubricating oil storage tank
16) 750 kW Diesel generator
17) Machinery control room and switch-board room
18) Engineers' workshop
19) Electrical workshop
20) Auxiliary boiler F.D. blower
21) Main condenser vacuum pump
22) Fuel oil unit
23) Fuel oil settling tank
24) Pump room entrance trunk
25) Ballast tank
26) Cargo pump turbine flat
27) Cargo pump room
28) Main boiler feed pump
29) Main extraction pump
30) Auxiliary extraction pump
31) Auxiliary condenser
32) Auxiliary condenser air ejector
33) 750 kW engine-driven generator
34) Auxiliary condenser sea water circulation pump
35) Auxiliary feed transfer pump
36) Main condenser
37) Auxiliary boiler feed pump
38) Lubricating oil centrifuge
39) Main reduction gearing
40) Main thrust block bearing
41) Shaft bearing
42) Main line shaft
43) Drain cooler
44) Steam/steam generator feed tank
45) Lubricating oil sludge tank
46) Dirty lubricating oil tank
47) Lubricating oil drain tank
48) Lubricating oil pump
49) Sea water service pump
50) Bilge pump
51) Diesel oil transfer pump
52) Stern tube lubricating oil pump
53) Combined observation and filter tank
54) Fresh water tank
55) Distilled water storage tank
56) Main feed heater
57) Fire tower
58) Main de-aerator and storage tank
59) Lubricating oil gravity tank
60) Elevator
61) Ship's service and control air compressors
62) Ship's service air reservoir
63) Control air reservoir
64) Bunker tank
65) L.P. heater and gland vapour condenser
66) Evaporating and distilling plant
67) Demineralizing plant
68) L.P. heater drain pump
69) Fuel oil transfer pump
70) Feed transfer pump
71) Main circulating pump
72) Bilge injection
74) Steam/steam generator feed pump
75) General service pump
76) Fire pump
77) Lubricating oil coolers
78) Lubricating oil filters
79) Centrifuge equipment
80) Auxiliary distilled water storage tank
81) Air conditioning machinery compartment
82) Auxiliary lubricating oil centrifuge
83) Emergency Diesel-driven air compressor
84) Starting air reservoir
85) Diesel generator lubricating oil sump tank
86) Diesel generator heat exchanger
87) Diesel oil daily service tank
88) Auxiliary drain tank
89) Main drain tank
90) Lubricating oil heater

FIGURE 1.8 Arrangement of Steam Propulsion Plant, Elevation View (Reference 2)

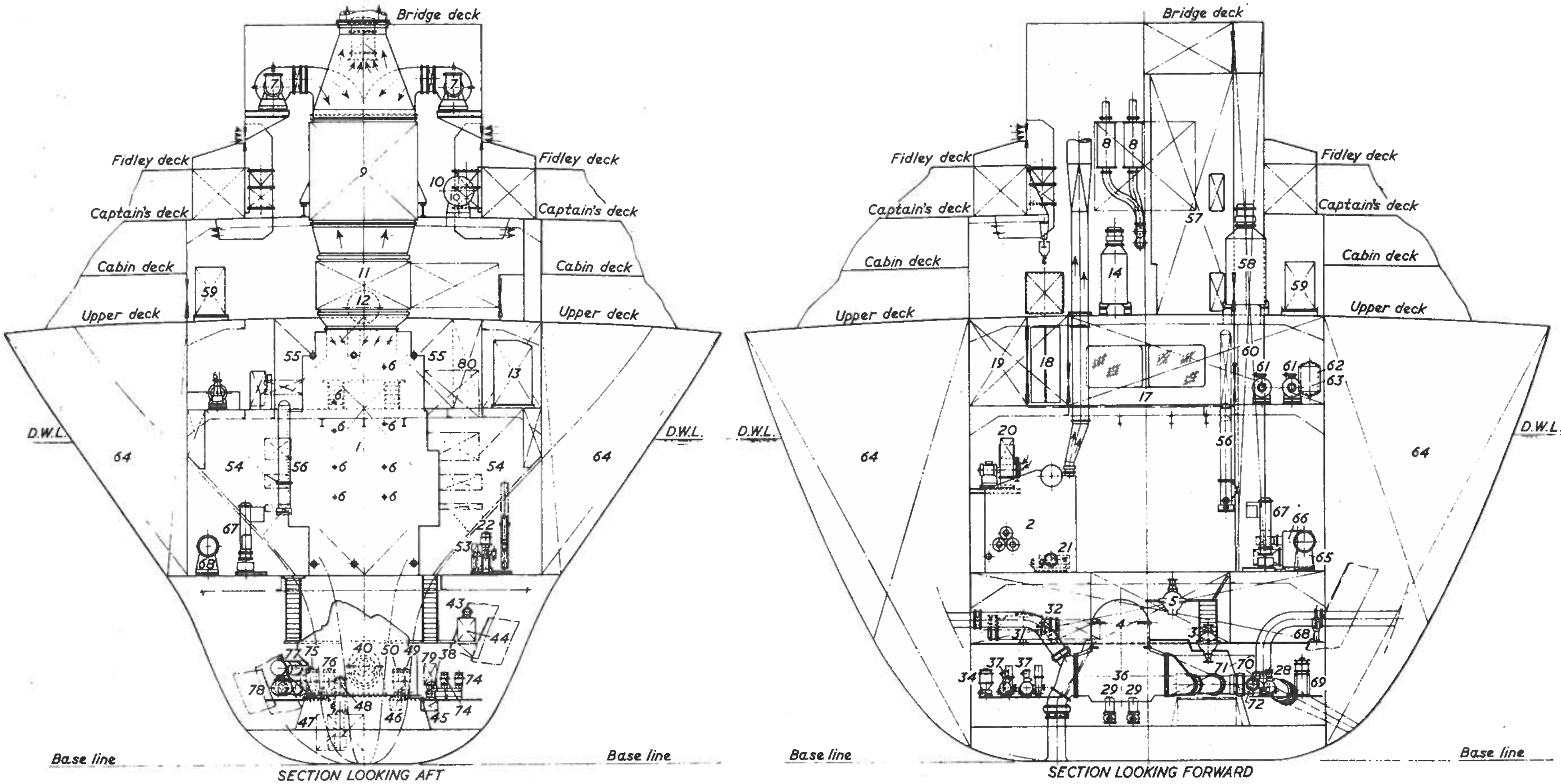
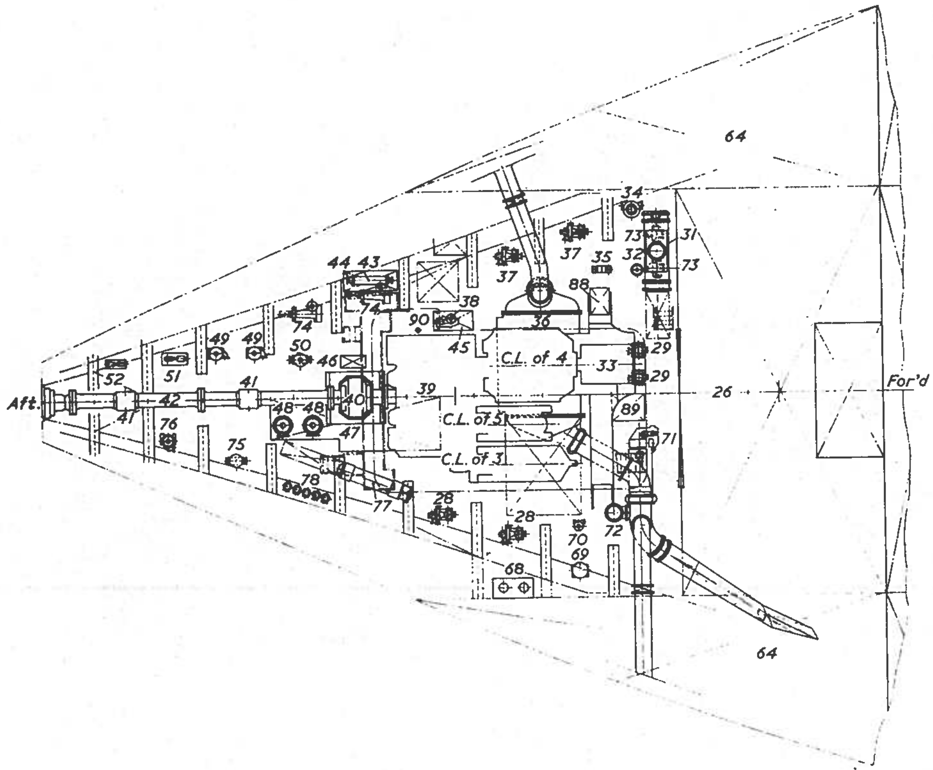
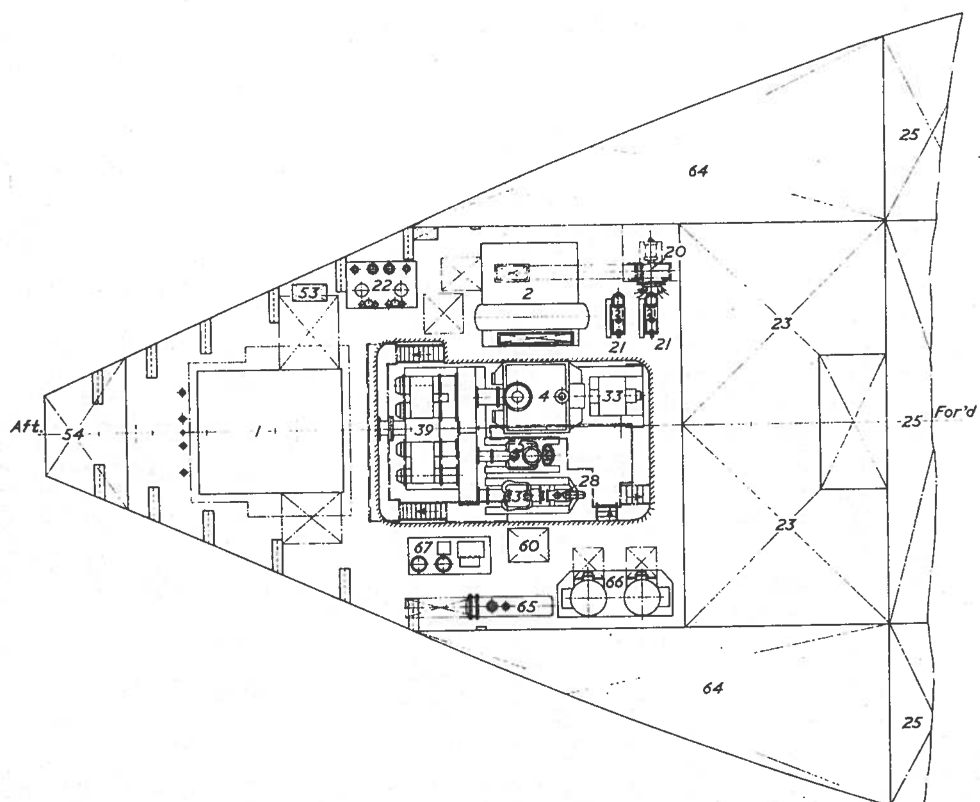


FIGURE 1.8 Continued, Sectional Views

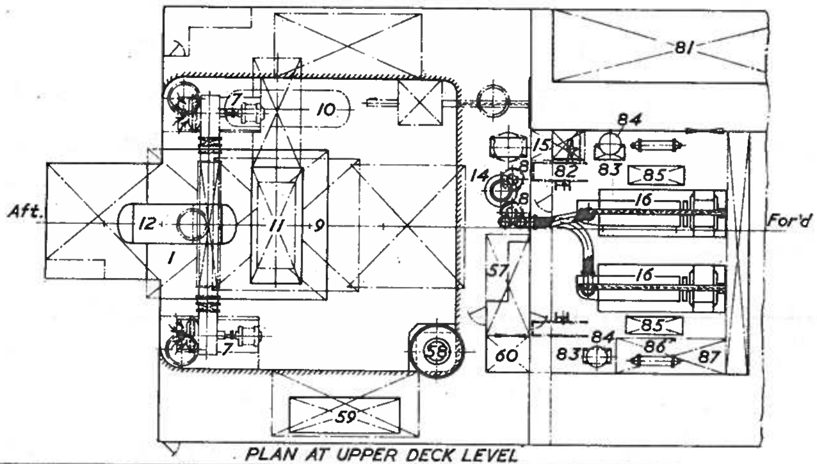


PLAN AT E.R. FLOOR LEVEL



PLAN AT 32'-0" FLAT ABOVE BASE

FIGURE 1.8 Continued, Plan Views



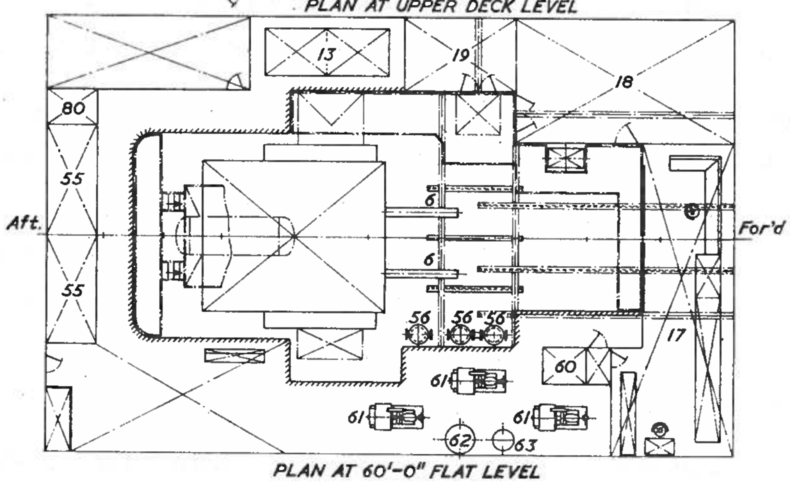


FIGURE 1.8 Continued, Plan Views

CHAPTER 2

CYCLE ANALYSIS

2.1 INTRODUCTION

It is here assumed that the student has taken at least a first course in thermodynamics, and as a result is familiar with conservation of mass and conservation of energy, and with the expression of the latter by the general energy equation. The reader is also expected to be familiar with the common properties, such as enthalpy, entropy, pressure, specific volume, and with their units in the SI system.

A modest knowledge of the power cycle concept is also assumed, although a good deal of this topic — at least as it concerns steam plants — is reviewed here. The student should at least know what the Carnot cycle is, and understand its significance as a standard of comparison for other cycles.

The first objective of a cycle analysis is to find the <u>efficiency</u> obtained by passing a working fluid through a closed sequence of processes between various limits. The efficiency is the ratio of output (mechanical power) to input (heat added), i.e. the ratio of what you get to what you pay for. The second objective is to find how much working fluid must be cycled to achieve a unit output (e.g. kg water per kWh).

In design of the marine steam power plant, the fundamental tool of the marine engineer is the <u>heat balance</u>. It is a complex form of cycle analysis that accomplishes the objectives just mentioned: efficiency is found for the purpose of comparison with other designs and with other types of power plant, and for estimating fuel costs and necessary tankage volume; the amount of working fluid used, and its pressures and temperatures, are the basic information for estimating the rating (and hence size and cost) of the many plant components. Its complexity make it seem far removed from the simple examples of the thermodynamics text. Students are often awed by the complexity, and confused by the many details, and so are unable to make the jump from simple principle to complex practice. Here they are to be eased into it step-by-step. This chapter starts with the elementary example and builds up a few complications at a time. At the end a student should be prepared to face the actualities of Chapter 3.

2.2 THE RANKINE CYCLE

Figure 2.1 shows the Rankine cycle on the temperature-entropy plane. The four processes are

- 1-2 Reversible adiabatic pumping (condenser to boiler)
- 2-3 Constant-pressure heat addition (in boiler)
- 3-4 Reversible adiabatic expansion (in turbine)
- 4-1 Constant pressure heat rejection (in condenser)

The efficiency is given by

$$\text{Efficiency} = \frac{\text{net work}}{\text{heat added}} = \frac{\text{heat in - heat out}}{\text{heat added}} \tag{2.1}$$

The formula can be written in terms of the property <u>enthalpy</u>, since inspection of the general energy equation (see the discussion of the first law of thermodynamics in any thermodynamics text) shows that

(a) for a constant-pressure heat transfer process with no shaft work, the heat transferred is equal to the change in enthalpy, and

(b) for a shaft-work process, with no heat transferred, the work done is equal to the change in enthalpy.

Using these converts equation (2.1) to

$$\begin{aligned}\eta &= \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2} = \frac{(h_3 - h_4) - v(P_2 - P_1)}{h_3 - h_2} \\ &= \frac{(h_3 - h_2) - (h_4 - h_1)}{h_3 - h_2}\end{aligned} \tag{2.2}$$

where η = efficiency
h = enthalpy
P = pressure
v = specific volume

Note that in one of the versions of equation (2.2), an enthalpy change is expressed as $v(P_2 - P_1)$. This is the pumping process in which the only component of enthalpy change is the pressure rise at essentially constant v.

The next equation gives the flow rate per unit of net output; net output is $(h_3 - h_2) - (h_4 - h_1)$, and the heat rate equivalent of mechanical power is 1.00 kJ/kWs, so

$$W = \frac{3600 \text{ sec/hour}}{(h_3 - h_2) - (h_4 - h_1)} \text{ kg/kWh} \tag{2.3}$$

W is called the water rate or steam rate of the cycle.

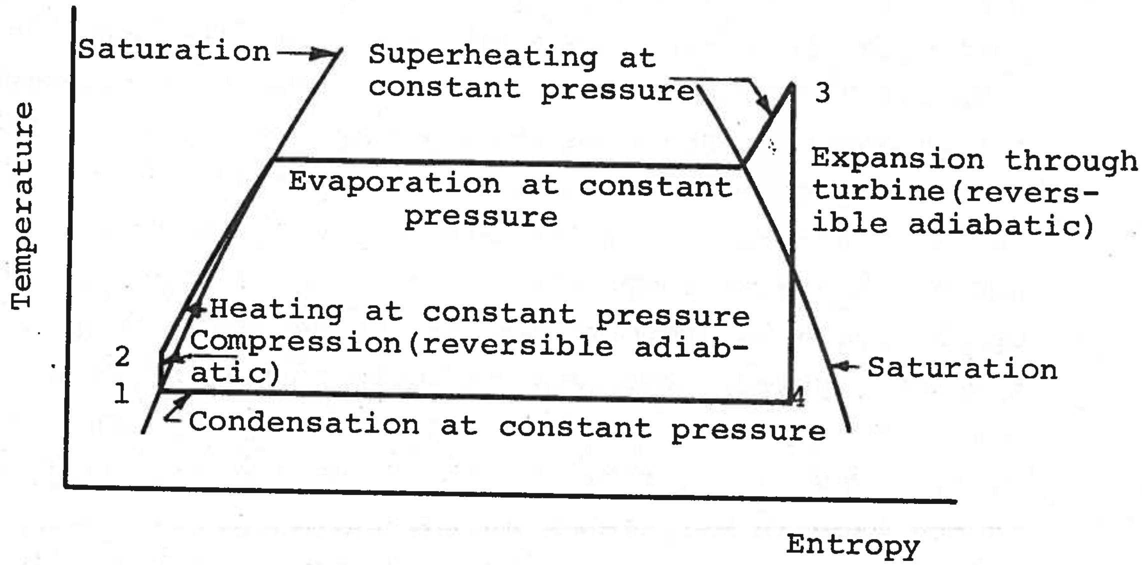
The heat rate is rate of heat expenditure (input) for a unit output, and is

$$H = \frac{3600}{\eta} \text{ kJ/kWh} \tag{2.4}$$

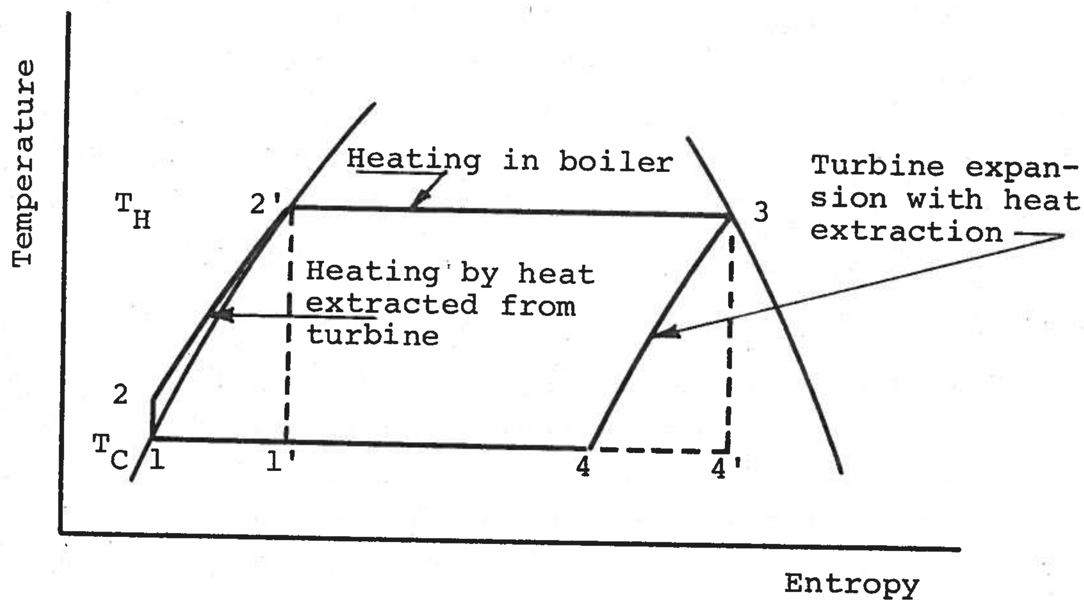
2.3 THE REGENERATIVE VAPOR CYCLE

2.3.1 General

A second sketch in Figure 2.1 shows a modified cycle. There is no superheat, but this is merely a convenience to make following discussions easier. The important modification is the shift of process 3-4 to the left. This is accomplished by extraction of heat from the fluid as it flows through the turbine; the extracted heat is added to the fluid along the line 2-2', i.e. part of the boiler's job is assumed by this internal shuffling of heat. On the T-S plane as used here, heat transferred in a curve representing the process, so that the heat



Rankine Cycle



Regenerative Cycle

FIGURE 2.1 Power Cycles on Temperature Entropy Plane borrowed from the turbine is proportional to the area 3-4-4' (using 4-4' as the datum line). Since heat added along 2-2' must equal that given up along 3-4, the area 3-4-4' equals the area 1-2-2'-1' (the jog 1-2 is so slight that it doesn't spoil the argument). The area enclosed by the cycle 1-2-2'-3-4 can be seen to be equal in area to that enclosed by the rectangular cycle 1'-2'-3-4'. The rectangular cycle is the Carnot cycle between the same temperature limits T_H and T_C. The area enclosed by the cycle is proportional to the net work, and is the same for both cycles. The heat rejected is proportional to the length 1-4 in one cycle, or 1'-4' in the other, and these lengths are equal. Thus it is that the heat-juggling <u>regenerative</u> cycle is the same in efficiency as the Carnot cycle. It is more efficient than the simple Rankine cycle, and as efficient as it possibly can be, as shown by its equality with the Carnot cycle.

In practice it is not feasible to effect direct transfer of heat between the turbine and the boiler. Regeneration is, however, accomplished in heat exchangers (feed heaters) which use steam <u>extracted</u> or <u>bled</u> from the turbine after it has flowed part way through, to heat the water (feed) on its way to the boiler. The efficiency does not approach that of the cycle described above, but still is distinctly superior to Rankine cycle efficiency. Also, superheated steam is typically used, even though the theoretical development is based on saturated.

It is not practicable to propound a simple efficiency formula for a regenerative cycle efficiency, beyond the word-statement of equation (2.1). Efficiency can be found, however, by working out the terms of equation (2.1) to suit a particular situation. The following section illustrates.

2.3.2 <u>An Example</u>

A power cycle is specified with the following principal pressures and temperatures (subscripts correspond to cycle points in top sketch, Figure 2.1):

P_3 = 40 bar (4.0 MPa)

T_3 = 500 C

$T_1 = T_4$ = 32.9 C (corresponding to a pressure of 0.05 bar (0.005 MPa) in condenser)

The corresponding enthalpies are

h_3 = 3445.5 kJ/kg (from STEAM TABLES)

h_4 = 2162 kJ/hg (found by striking a constant-entropy line down from point 3 to its intersection with the P=0.005 mPa line, using Mollier chart)

h_1 = 137.8 kJ/kg (saturated liquid at 0.005 MPa)

h_2 = 141.8 kJ/kg (=137.8 + 1.0053 m^3/kg x 3.995 MPa)

v = 1.0053 is specific volume of water @ 32.9 C, m^3/kg

It is further specified that there be two feedwater heaters between pump and boiler. The first is to heat the water to 100C, and the second to 140C (saturation temperature in the boiler is 250.4C) using steam extracted from corresponding points in the turbine. Heating is accomplished by mixing steam with water, so that pressure in the heaters must be saturation pressures for the specified temperatures, i.e. 0.10135 MPa for 100C and 0.3613 mPa for 140C. Since no piping pressure drop is assumed in an idealized analysis such as this, these are the pressures at which the steam is extracted from the turbine. Hence the term "corresponding points" a few sentences above: the points at which the pressures are 0.10135 mPa and 0.3613 mPa in the turbine. For the calculation to follow, the enthalpies of the heating steam are needed. They are found by laying out the turbine <u>state line</u>, which is in this simple example the straight line from 3 to 4, on a Mollier chart (an enthalpy-entropy chart), and noting the enthalpies where this line intersects the contours P=0.10135 and P=0.3613. Their values are included in the summary following:

$T_{H1} = 100$ C | $T_{H2} = 140$ C

$P_{H1} = 0.10135$ MPa | $P_{H2} = 0.3613$ MPa

$h_{EX1} = 2575$ kJ/kg | $h_{EX2} = 2800$ kJ/kg

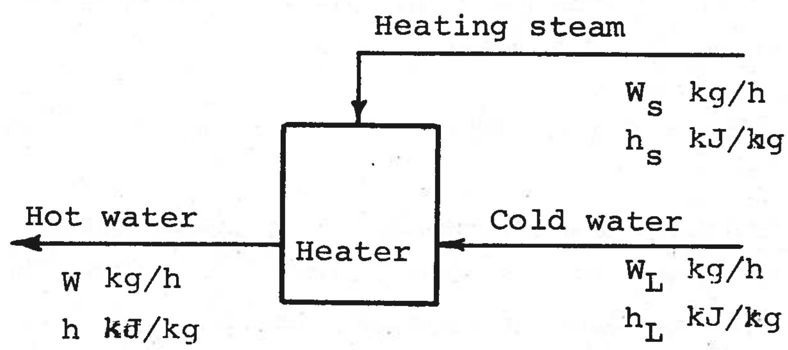
$h_{H1} = 419.0$ kJ/kg | $h_{H2} = 589.1$ kJ/kg

Meanings of subscripts should be apparent from the preceding discussion; the last enthalpy in each column is that of the saturated liquid leaving the heater.

Now, the problem is the analysis defined earlier, i.e. finding efficiency and water rate. The requisite steps in finding efficiency are calculation of net work and of heat input, also as before. Heat input comes easy; it is the heat required to heat the water leaving the second heater (saturated liquid at 140C) to the conditions at point 3. Net work is more involved; the extraction process robs a fraction of the steam flow from the turbine before it completes the transfer of energy to the turbine, and this must be accounted for in the net-work calculation, as will be illustrated. Meanwhile, the vital bit of information is how much steam is extracted by the heaters. The steam mixes with the water being heated, and is condensed to a liquid in the process. The amount of steam extracted is thus that amount which the water can condense in being heated through the specified range (later, when more practical details are added, this rule has to be expanded, but it suffices for this elementary example). The calculation of this amount is based on conservation of mass, and conservation of energy. Look at Figure 2.2 to get an idea of the situation with heaters in which water and steam mix (direct contact heaters). The equations that express the conservation principles, using the symbols of Figure 2.2, are

$$W = W_S + W_L \tag{2.5}$$

$$W_S h_S + W_L h_L = W\ h \tag{2.6}$$



Flows to and from a direct-contact feedwater heater

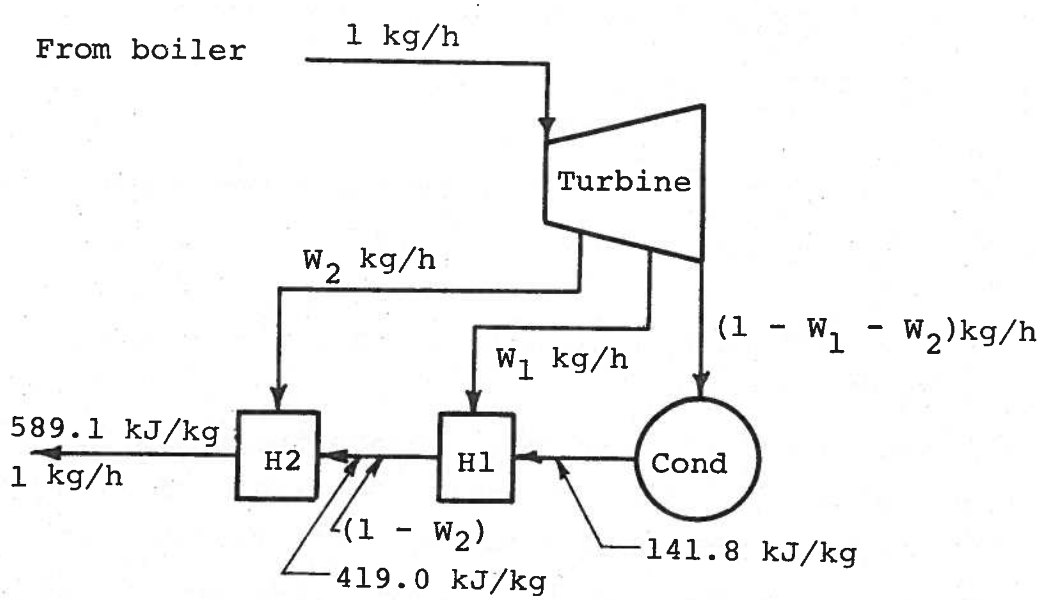


Illustration for the numerical example, finding extraction flows to feedwater heaters

FIGURE 2.2 Illustrations for Examples in Text

From which $W_1 = 0.1058$ kg/h $\qquad W_2 = 0.0714$ kg/h

Since the problem is being worked with 1.0 kg/h total flow, these values are numerically equal to the fractions of the total flow extracted. Subsequently, you will note w being used to denote a fraction of a flow, i.e.

$$w_1 = 0.1058 \text{ kg/h} \qquad w_2 = 0.0714 \text{ kg/h}$$

Now, to find the net work. The gross work is the sum of work done by 1 kg flow to the first extraction, by $1-w_2$ kg flowing to the second extraction, and by $1-w_2-w_1$ flowing thence to the condenser. Net work is this less (h_2-h_1). Thus,

$$\begin{aligned}\text{net work} = {} & (3445 - 2800) + (1 - w_2)(2800 - 2575) + \\ & (1 - w_2 - w_1)\ (2575 - 2162) - (141.8 - 137.8) \\ = {} & 1189.8 \text{ kJ/kg (per original 1 kg)}\end{aligned}$$

And heat added = 3445 - 589.1 = 2856 kJ/kg

And efficiency $= \dfrac{1193.8}{2856} = 0.418$

By contrast, if there were no regenerative feed heating, efficiency would be found by equation (2.2) as

$$\eta = \frac{(3445 - 2162) - (141.8 - 137.8)}{3445 - 141.8} = 0.387$$

Now, let's proceed to calculate the flows for a specified power output, and also several other items of interest. To introduce an element of realism, let's say that the net work rate is 10,000 kw. The total flow rate, or flow to the turbine throttle, is

$$W_T = \frac{(3600)(10000)}{1189.8} = 30{,}257 \text{ kg/h}$$

The extracted flows are

$$W_1 = (0.1058)(30{,}257) = 3201 \text{ kg/h}$$

$$W_2 = (0.0714)(30{,}257) = 2160 \text{ kg/h}$$

The heat transfer rates involved in the cycle are

$$\begin{aligned} \text{in boiler:} \quad & Q = (30{,}257)(3445 - 589.1) = 86.4 \times 10^6 \text{ kJ/h} \\ \text{in heater 1:} \quad & Q = (3201)(2575 - 419.0) = 6.90 \times 10^6 \\ \text{in heater 2:} \quad & Q = (2160)(2800 - 589.1) = 4.78 \times 10^6 \end{aligned}$$

If the heating value of the fuel being burned to supply the boiler heat is assumed, then the fuel rate of the cycle — the fuel consumption rate per kW — can be calculated. Take the value to be 43,000 kJ/kg.

$$FR = \frac{3600}{(0.418)(43{,}000)} = 0.200 \text{ kg/kWh}$$

2.3.3 Discussion of Above Example

The example in 2.3.2 illustrates the analysis of a simple regenerative power cycle, using two direct-contact feed heaters. It was calculated simply and directly, and so scarcely hints at the lugubrious complexities of a "heat balance" of a practical power plant, but yet it does serve as an introduction to some aspects of practical heat balance. So far as techniques go, it introduces

(a) Finding enthalpy at which steam is extracted from a turbine.

(b) Finding the amount of steam extracted by applying a conservation calculation to a heater using this steam.

And note the design input information contained in the results, such as

(a) The fuel consumption rate is found, so that fuel tankage can be assigned, fuel costs estimated, and fuel system components sized.

(b) Flow rates throughout the cycle are established, so that all pipe and pump sizes can be found.

(c) Principal specifications for system components are found, so that their design can proceed. For example for the second heater, one can specify thusly: a direct contact heater to heat 28,096 kg/h of water from 100C to 140C, using steam at 0.3613 MPa, 2800 kJ/kg. Although a complete specification would doubtless contain more details, this is the essence of what is needed for a heat-exchange specialist to proceed with design of the heater.

2.3.4 Introduction of the Replacement Factor

If you do the calculation of steam flow without extraction in the 2.3.2 example, using equation (2.3) to get flow per kWh, the result is 28,147 kg/h, showing the 2,110 kh/h extra flow is needed to compensate for the work lost when steam is extracted from the turbine before it reaches the condenser. In an actual heat balance, it is necessary to calculate the individual extraction flows, and calculate how much the flow to the turbine (the throttle flow) must be changed so that turbine output stays the same. The calculation is effected readily by means of a replacement factor. The replacement factor for each extraction opening is the fraction of the extraction flow that must be replaced, i.e. added to the throttle flow, to avoid disturbing the mechanical output of the turbine. It is a function of enthalpies, as developed and illustrated next.

Repeat the calculation of throttle flow made in 2.3.2, but with the denominator expanded, and subsequently rearranged, to show up the replacement factors.

$$W_T = \frac{(10,000)(3600)}{[(3445-2800) + (2800-2575)(1-w_2) + (2575-2162)(1-w_2-w_1)]}$$

$$= \frac{(10,000)(3600)}{[(3445-2162) - w_2(2800-2162) - w_1(2575-2162)]}$$

$$= \frac{(10,000)(3600)}{(3445-2162)\left[1 - w_2\,\frac{2800-2162}{3445-2162} - w_1\,\frac{2575-2162}{3445-2162}\right]}$$

$$= \frac{(10,000)(3600)}{(3445-2162)(1 - w_2R_2 - w_1R_1)}$$

$= 30,257$ kg/h when numerical values are used.

The factors R_1, R_2 are the replacement factors. You can readily see what the general formula for these factors is. It should be noted here that the w are fractions of the total flow, not of the original non-extraction flow.

The following formulas relate the several flows and the replacement factors:

$$W_T = \frac{3600\times\text{power}}{\Delta h_w\left[1 - \sum_i w_iR_i\right]} \tag{2.7}$$

$$W_T = W_{NE} + \sum_i W_iR_i \tag{2.8}$$

$$W_{NE} = \frac{3600\times P}{\Delta h_w} \tag{2.9}$$

$$W_i = w_iW_T \tag{2.10}$$

where P = power output

Δh_w = enthalpy difference, throttle to condenser

W_{NE} = non-extraction flow rate

W_T = throttle flow

2.3.5 Cycle Analysis with Component Inefficiencies

To add another item of realism, let's say that the boiler and turbine are to a certain extent inefficient, i.e. they cannot make use of all of the energy supplied to them, and let's again illustrate by example. Assume that boiler is 90% efficient, and turbine is 80% efficient.

The steam is still supplied to the turbine at the same condition as before, and it still exhausts to the same pressure, but in passing from throttle to exhaust, only 80% of the available enthalpy difference is converted to work. Thus the enthalpy at which the steam exhausts is 3445 - 0.80(3445-2162) = 2419 kJ/kg. Look at Figure 2.3, which shows the pertinent turbine information on the enthalpy-entropy plane. The two end points are spotted (P=4.0 MPa, h=3445 kJ/kg and P=0.005 MPa, h=2419), and then a straight line drawn to represent the intermediate state points through the turbine (straight line is good approximation to what actually happens). As before, the enthalpies at extraction points are found by reading their values where the specified pressure contours cross the turbine state line.

The boiler inefficiency has less impact on the calculation. Boiler losses represent the heat that does not enter the working fluid, e.g. heat that goes up the stack rather than to boiling water and superheating steam. Therefore, calculations of efficiency and flows <u>within the working-fluid cycle</u> are not affected; boiler efficiency enters when the fuel rate is calculated.

The calculations of 2.3.2 are repeated below with the numbers appropriate to this section.

Extraction flows: $2785\, w_1 + 141.8(1 - w_1 - w_2) = 419.0(1 - w_2)$

$$2985\, w_2 + 419.0(1 - w_2) = 589.1$$

giving $w_1 = 0.0979 \qquad w_2 = 0.0663$

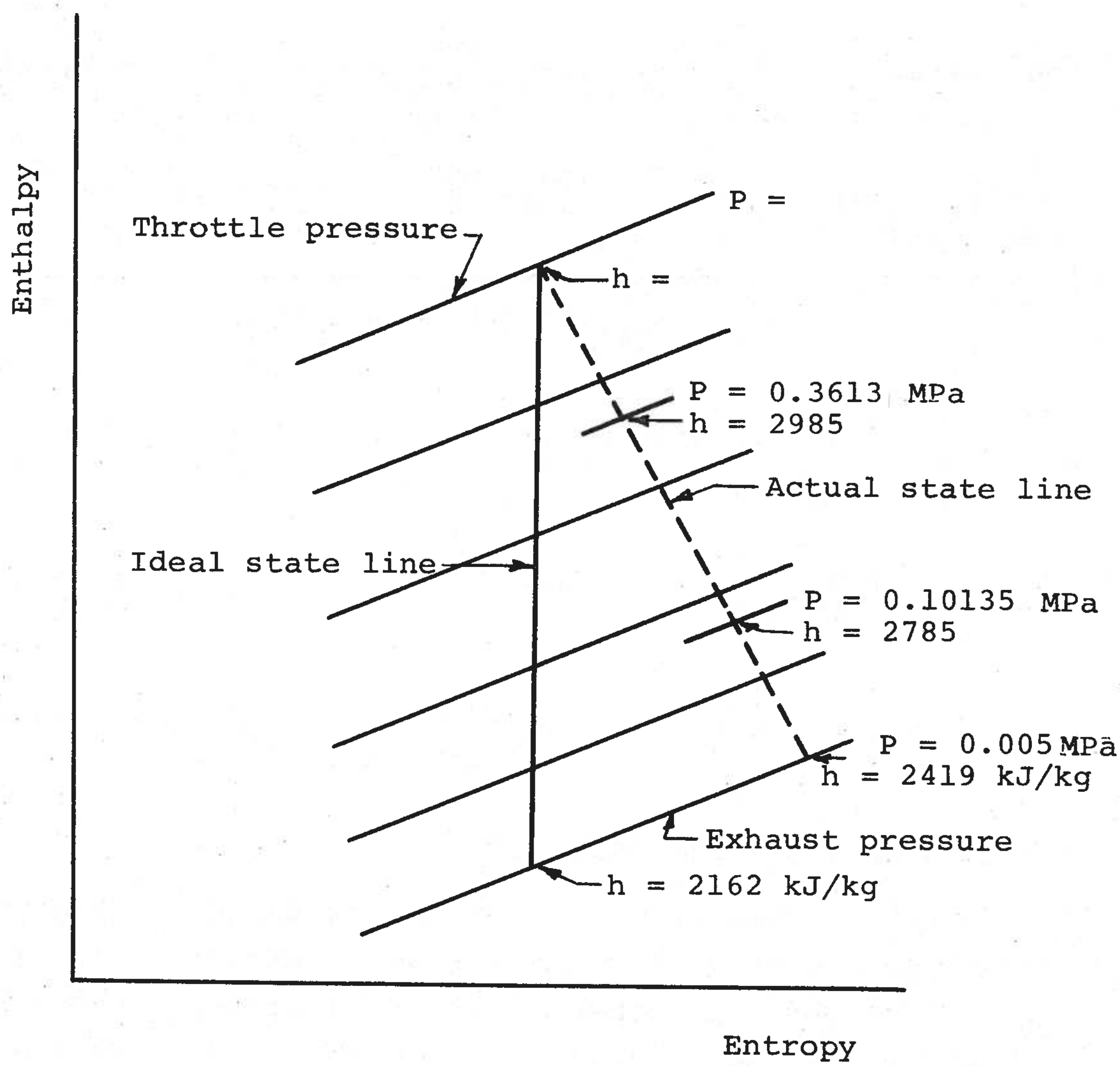


FIGURE 2.3 Turbine State Line for Example of Cycle Analysis with Component Inefficiencies

Net work: $\text{work} = (3445 - 2985) + (1 - 0.0663)(2985 - 2785) + (1 - 0.0663 - 0.0979)(2785 - 2419) - (141.8 - 137.8)$

$= 948.6$ kJ/kg

Efficiency: $\eta = \dfrac{948.6}{2856} = 0.332$

Fuel rate: $F_R = \dfrac{3600}{(0.332)(0.90)(43{,}000)} = 280.0$ kg/kWh

Throttle flow: $W_T = \dfrac{(10{,}000)(3600)}{948.6} = 37{,}950$ kg/h
(for 10,000 kW)

$$W_{NE} = \frac{(10{,}000)(3600)}{(3445 - 2419)} = 35{,}088 \text{ kg/h}$$

Repeat finding of throttle flow, based on W_{NE} and replacement factors:

$$R_1 = \frac{2785 - 2419}{3445 - 2419} = 0.357 \qquad R_2 = \frac{2985 - 2419}{3445 - 2419} = 0.552$$

$$W_T = \frac{37{,}950}{[1 - (0.357)(0.0979) - (0.552)(0.0663)]} = 40{,}874 \text{ kg/h}$$

2.4 A NOTE ABOUT THE FEED PUMP

We've diligently calculated the feed pump work whenever getting cycle net work in this chapter. Numerically it isn't a big thing, but at higher pressures it becomes increasingly significant, so it shouldn't be neglected. But then, a nettling question comes up when you look at a cycle diagram, such as Figure 2.2: where is the feed pump? Where indeed — that figure doesn't show any feed pump at all! Well, your professor was embarrassed by not knowing where to put it, and left it out in hopes you wouldn't notice. Thing is, the pressure rises successively from condenser to heater H1 to heater H2 to the boiler, hence there should be a pump after the condenser, one after heater H1, and another after heater H2. These collectively would form the "pump" of the cycle analysis.

Now, if you improved (?) the analysis by calculating feed pump work for each of these three individual units, the calculations would be much more complex, but with a negligible improvement in accuracy. To spare you this punishment, the calculations are made as if there were only one pump, and it's assumed to raise the pressure from that of the condenser to that of the boiler.

There's no one place this equivalent pump can be put in Figure 2.2, so he escaped the difficulty by leaving it out of the diagram. When one gets to the fine points of reality, all pumps must be shown and accounted for, of course.

2.5 REFERENCES

No references have been cited in this chapter, but for completeness, one thermodynamics text should be listed. Also, numerical values for the steam properties must be looked up somewhere. Thus the following brief list.

1. G. J. van Wylen and Richard E. Sonntag, Fundamentals of Classical Thermodynamics, John Wiley, 1965.

2. Steam Tables, Thermodynamic Properties of Water, including Vapor, Liquid and Solid Phases, John Wiley & Sons, 1969.

CHAPTER 3

TECHNIQUES OF PRELIMINARY HEAT BALANCE

Chapter 2 introduced the concept of a heat balance (the "cycle analysis" of practical design), but it didn't take you very far into the techniques of doing it. The example problems illustrated the principles, and introduced some of the concepts (e.g. replacement factor) that are needed in this chapter, but didn't get into the complexities of an actual propulsion plant. This chapter introduces some of those complexities. At the end of the chapter you will still not have seen them all, but should have gotten enough understanding to be able to withstand the further buffets of reality on your own. In particular, you should understand the technique of the overall balance.

The best-known authority in the marine heat balance field is the SNAME Technical & Research Bulletin 3-11, Marine Steam Power Plant Heat Balance Practices, (Reference 1). It's packed with the information you need, especially for preliminary heat balancing, but it's not a textbook. It's more of a handbook; you have to know from other sources what to do with the information it presents. The major purpose of this chapter is to ease you up to the point of knowing.

Before getting to work, let's look at that word "preliminary." It implies that plant components are to be designed to suit the result of the heat balance. You are free to specify temperatures and pressures, for example, in the expectation that someone will design the components to produce them. This is quite different from being handed an assortment of existing turbines, boilers, pumps, heat exchangers, etc, and then being asked to calculate what they will do. That's discussed in a later chapter, so forget it for now.

3.2 GENERAL TECHNIQUE

You typically start with a "heat balance diagram," a block diagram of propulsion plant components on which all significant flows of steam and water are shown. The general approach is the following:

(1) Identify all of the flows, and note how each is to be found in terms of the others, i.e. write an equation for each flow.

(2) Calculate the values of the coefficients of the relationships found in 1.

(3) You now have a set of equations for the fundamental unknowns of the problem, i.e. the flows. If not linear, then linearize, then solve. (See Section 3.5 for linearization technique.)

(4) Calculate values of other parameters (not flows) that depend on the flows.

3.3 EXAMPLE OF A SIMPLE PRELIMINARY HEAT BALANCE

3.3.1 Preliminary Discussion

Look at Figure 3.1, a block diagram or "heat balance diagram" of a steam propulsion plant. It doesn't contain all of the complexity of an actual plant, but has sufficient complexity to make necessary the techniques just listed. This example is similar to the last one in Chapter 2 in that it also has two regenerative heaters, and pressures and temperatures are the same. However, several changes and additions appear, and we had better discuss them before getting into the calculations.

First, the feed pump is shown in its correct place, and driven by its own turbine, as is common practice. Exhaust steam from the turbine is condensed in the second heater, thus contributing to the heating done by extraction steam. To find how much exhaust steam is flowing, and its enthalpy, you should make the same calculation done for the propulsion turbine (using the

Mollier chart, and all that). For this example, however, a rule-of-thumb formula is stated for the steam flow, and the exhaust enthalpy is given.

Second, the first heater is as a shell-and-tube (or "surface") type, meaning that water and heating steam do not mix, but are separated by keeping the water inside tubes passing through the steam space. The steam condenses on the outer surface of the tubes, and the condensed liquid (the "drains") leave the heater separately (note WDR in the figure), and at a temperature higher than that of the water being heated. This difference in temperature is the "terminal temperature difference" (TTD). The following summarizes the other significant points:

(1) Temperature of drains is the sum of TTD and the temperature of the water leaving the heater.

(2) Temperature of the steam as it condenses is the same as that of the drains.

(3) Pressure of the steam in the heater is saturation pressure corresponding to steam temperature.

(4) Enthalpy of steam entering the heater is found at the intersection of propulsion turbine state line and heater steam pressure.

(5) Pressure mentioned in (4) is the sum of the pressure mentioned in (3), plus any pressure drop in the pipe connecting heater to extraction point on the turbine. (In the upcoming example, that pressure drop is zero.)

(6) The flow of extraction steam to the heater is found by a heat balance formula of the form "heat in equals heat out" as in Chapter 2. For the example here, that formula appears as

$$\mathrm{WEX1} = \mathrm{WCOND}\,\frac{(\mathrm{HH1} - \mathrm{HCOND})}{(\mathrm{HEX1} - \mathrm{HDR})} \qquad (3.1)$$

Symbols used here are taken from Figure 3.1. HH1, HCOND, and HDR are saturated liquid enthalpies.

Third, a condensate pump is shown (the circle just to the left of the condenser). This pump removes water from the condenser and pumps it through the tubes of the first heater into the second heater. The feedpump does the pumping job from there to the boiler. The energy added by the condensate pump is small, hence the liquid is regarded as being saturated from condenser until it passes the feed pump. The pump is driven by an electric motor (not shown in the figure). The power is small, hence its accurate determination is not important, and is considered to be included in the electric load.

Fourth, a turbine driving the ship service electric generator is included. Its steam consumption (WGT) should be found by a turbine calculation identical to that used for the propulsion turbine, after an analysis of the electric load has shown what power is required. For this example, however, a rule-of-thumb formula is stated for the steam consumption. It is based, in part, on total flow of water to the boiler (WBO), hence is related to propulsion plant auxilliary loads such as power required for boiler draft fans and that condensate pump.

3.3.2 The Flow Equations

There are 8 flows: WNE, WPRT, WGT, WEX1, WEX2, WFPT, WCOND, WBO. All except the first are marked on Figure 3.1, and you can determine what they are by looking at that figure. WNE is the steam flow to the propulsion turbine when no extraction is occurring; its value is 105,263 kg/h, gotten by putting 30,000 kW and the difference between 3415 and 2419 kJ/kg into Equation (2.9). It's a constant in the problem solution, so we need say no more about it.

WFPT is the flow of steam to the propulsion turbine when extraction is occurring. Its formula, which you should recognize from Chapter 2, is

$$WPRT = WNE + R1 \cdot WEX1 + R2 \cdot WEX2 \qquad (3.2)$$

where

$$R1 = \frac{HEX1 - HEXH}{HSUP - HEXH} = 0.407 \qquad R2 = \frac{HEX2 - HEXH}{HSUP - HEXH} = 0.552 \tag{3.3a,b}$$

with the numerical values applying to the example. You should recognize the formulas for R1 and R2 from Chapter 2.

WGT is the steam flow to the generator turbine. For the example, the rule-of-thumb for electric power is 800 + 0.02 WBO, and the steam flow is 4.5 times this, or

$$WGT = 3600 + 0.09\ WBO \tag{3.4}$$

WEX1 is the extraction flow to the first (lower pressure) heater. The formula has been discussed earlier, and appears as Equation (3.1). Numerical values entered for the enthalpies reduce it to WEX1 = 0.1184 WCOND. Again, you should recognize the formula from Chapter 2, although the shell-and-tube construction has forced one change (i.e. HDR $\neq$ HH1).

WEX2 is the extraction flow to the second (higher pressure) heater. This heater is of the direct-contact type as introduced in Chapter 2, but the calculation of WEX2 is complicated by the contribution of WFPT to the heating. The principle of "heat in = heat out" nonetheless does the job, giving

$$WEX2 = \frac{HH2 - HH1}{HEX2 - HH2}\ WCOND - \frac{HFPT - HH2}{HEX2 - HH2}\ WFPT = 0.0710\ WCOND - 1.0897\ WFPT \tag{3.5}$$

WFPT is the steam flow to the feed pump turbine, and subsequently the exhaust flow therefrom to the second heater. For this example, we're using a simple rule-of-thumb, it being

$$WFPT = 0.06\ (WCOND + WFPT + WEX2) \tag{3.6}$$

WCOND is the sum of flows leaving the condenser, hence is sum of flows entering. Look at Figure 3.1.

WBO is the sum of flows entering the boiler, or leaving the boiler. Look at Figure 3.1.

We have now completed steps 1 and 2, i.e. written the flow equations, and calculated the coefficients for every term in these equations. To summarize, these equations are now

$$
\begin{aligned}
WPRT &= 105{,}263 + 0.407\ WEX1 + 0.552\ WEX2 \\
WGT &= 3600 + 0.09\ WBO \\
WEX1 &= 0.1184\ WCOND \\
WEX2 &= 0.0710\ WCOND - 1.0897\ WFPT \\
WFPT &= 0.06\ (WCOND + WFPT + WEX2) \\
WCOND &= WPRT + WGT - WEX2 \\
WBO &= WEX2 + WFPT + WCOND
\end{aligned}
\tag{3.7}
$$

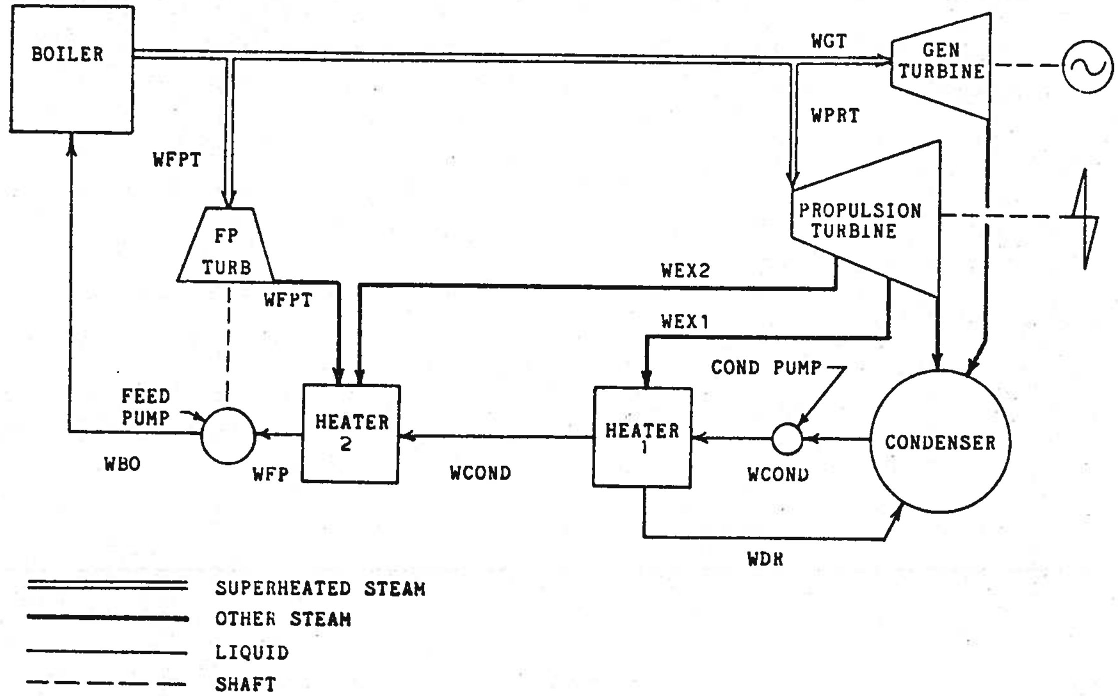
Seven linear equations, seven unknowns. In the recommended step 3 you solve them! Easy, eh? Or maybe not, depending on how you do it. By hand (i.e. without computer) it will be tough; with computer -- which will likely have a library routine for solving sets of equations -- easy. I'll leave the choice to you. Nonetheless, the following sub-section outlines an iterative scheme that represents the traditional way of hand calculation, a way that you can use with a programmable calculator that doesn't have the capacity to accept a set of 7 equations simultaneously, and a way that you might want to use if one or more of the equation coefficients are non-linear. (See Section 3.5 for the non-linear part.)

3.3.3 Iterative Solution Technique

It's simple in principle: (1) make an estimate of each unknown, (2) enter these estimates into the right sides of Equations (3.7), (3) solve, (4) replace the estimates with the solution values, (5) repeat, repeat, repeat, until the changes in the unknowns become insignificant. The table below contains the numbers that actually occurred in my solution by this method.

WPRT	105,263	105,263	110,250	111,765	....	111,482
WGT	14,000	15,300	15,056	15,572		15,785
WEX1	0	12,463	14,293	14,689		15,048
WEX2	0	-154	1,241	395		172

(continued)



SPECIFIED CONDITIONS

Propulsive power	30,000 kW
Electric power	800 + 0.02 WBO kW
Boiler pressure	4.0 MPa
Turbine exhaust pressure	0.005 MPa
First heater temperature, out (TH1)	100 C
Second heater temperature, out (TH2)	140 C
First heater terminal dT (TTD)	10 C

DERIVED CONSTANTS

Superheated steam enthalpy (HSUP)	3445 kJ/kg
Turbine exhaust enthalpy (HEXH)	2419
Extraction enthalpy (HEX1)	2837
Extraction enthalpy (HEX2)	2985
Enthalpy at TH1 (HH1)	419.0
Enthalpy at TH2 (HH2)	589.1
Enthalpy from condenser (HCOND)	137.8
Enthalpy of heater 1 drain (HDR)	461.3
Enthalpy from feed pump turbine (HFPT)	3200

FIGURE 3.1 Diagram and Data for First Example

WFPT	7,000	6,727	7,721	7,931	....	8,124
WCOND	105,263	120,717	124,065	126,943		127,095
WBO	130,000	127,290	133,028	135,269		135,391

The 8 dots represent 8 columns omitted, so there were 12 iterations following the initial estimates. A lot, surely, but the major work was setting the equations up on a programmable calculator. After that, it was just a few minutes of punching the keys. If I had been less particular about getting an "exact" answer (no change to the left of the decimal in going from the n^{th} iteration to the $n+1^{th}$), I could have been satisfied with far fewer than 12 iterations. Note that except for WEX2, all of the variables are close to their final values after 3 iterations. The difficulty with WEX2 is caused by its being the small difference between two larger numbers. You won't always run into this situation, but sometimes, as here.

How did I make the initial estimates? I started with no idea of what WEX1 and WEX2 could be, so simply set them at zero, confident that the iterative process would take care of me. This assumption made WPRT equal to WNE, then by looking at the flow diagram and the equations (e.g. for WFPT) I got rough values for the other flows. Your estimates might have been different, but no matter, for the final result does not depend on these estimates. What does depend on them is the number of iterations required. The better the guess, the fewer the iterations. If you were doing this without a programmed machine, then it indeed might be important to make the best possible estimates to begin. An experienced "heat balancer" can usually guess quite close to the answer, and would probably solve this problem with two or three iterations.

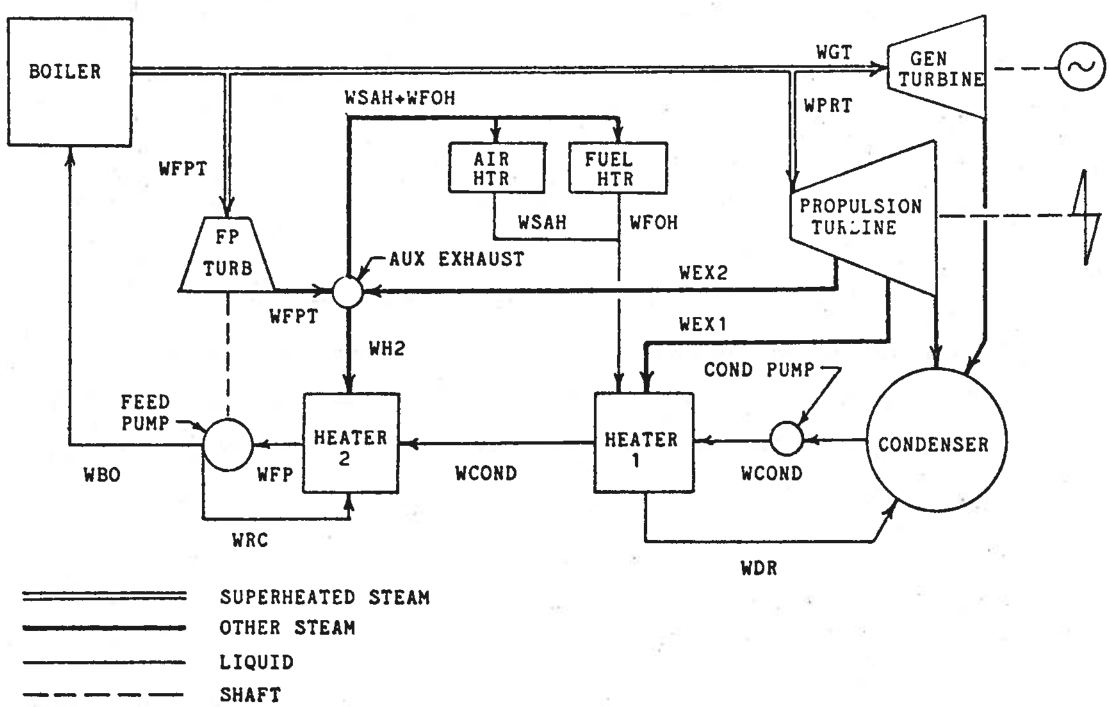
3.3.4 Finishing the Example

The remaining step is to calculate the other values that may be required in the problem. The examples in Chapter 2 have given some hint of these, e.g. magnitudes of heat transferred. Here

WBO has been found to be 135,391 kg/h; the enthalpy change in producing this steam is 2856 kJ/kg (from HSUP - HH2); heat transferred in the boiler is therefore 386.7×10^6 kJ/h. If this heat is produced in a boiler of 0.83 efficiency, burning coal of 30,000 kJ/kg heating value, then 15,530 kg/h of coal is burned, and the specific fuel consumption (fuel rate) is 518 gr/kWh.

3.4 ANOTHER EXAMPLE

The second example is much like the first, but has been complicated by the addition of several real-practice elements to the propulsion plant. Look at Figure 3.2, and compare to Figure 3.1. The additional elements that are apparent in the new figure are an air heater, a fuel heater, a recirculation line (flow WRC), and the circle marked AUX EXHAUST. An added feature that is not visible, but which appears in the calculations, is an accounting for the change in enthalpy of feedwater caused by the feed pump.



The air heater is a preheater for the combustion air, and the fuel heater is the heater always essential for preheating residual oil fuel to reduce its viscosity. Both are supplied with steam from the auxiliary exhaust, and their condensate flows to the first heater, where it contributes to the heating. Both are assumed to operate at the same pressure as heater 2, hence the enthalpy of this condensate is taken to be HH2.

The formula for the steam required (WEX1) by heater 1 is changed by the addition of WSAH and WFOH, but it is still nothing more than a simple heat in = heat out expression. It is

$$\begin{aligned} \mathrm{WEX1} &= \frac{\mathrm{HH1 - HCOND}}{\mathrm{HEX1 - HDR}}\,\mathrm{WCOND} + \frac{\mathrm{HDR - HH2}}{\mathrm{HEX1 - HDR}}\,(\mathrm{WSAH + WFOH}) \qquad (3.8) \\ &= 0.1184\ \mathrm{WCOND} - 0.248\ (\mathrm{WSAH + WFOH}) \end{aligned}$$

The auxiliary exhaust is just a pipe into which exhaust steam flows from all sources (though there's only one exhaust source here), and into which the extraction steam (WEX2) flows to make up any deficiency in heating demand. Of course these two steam contributions mix, producing an enthalpy that is different from that of either source. This complicates the calculations, but we can avoid most of the trouble by remembering from the previous example that heater 2 used up <u>all</u> of the exhaust steam, then required a further contribution from the extraction steam. No different here, so the formula for WEX2 is in principle the same as before. Since no part of WFPT is available to supply the two new heaters, we'll have to say that their steam comes from the propulsion turbine and therefore has enthalpy HEX2. Here, though, we'll simply say

$$\mathrm{WSAH} = 0.04\ \mathrm{WBO} \qquad (3.9)$$

$$\mathrm{WFOH} = 0.005\ \mathrm{WBO} \qquad (3.10)$$

And where did these formulas come from? They are rules-of-thumb devised for <u>this</u> example. In an example going back closer to fundamentals, you would have to make a heat balance calculation for each of these heaters, balancing the heat required by air or oil against the heat supplied by the unknown steam flows.

The recirculation line is commonly used to ensure that water always flows through the feed pump. The amount recirculated (WRC) is usually treated as a constant (5000 kg/h in this example). Not much trouble in that, but a further complication is the energy (enthalpy change) that the water picks up in being pumped, the "regain" in the terminology of the SNAME heat balance bulletin. It contributes to the heating in heater H2, and so must be included in the heat balance calculations appropriate to that heater. This regain is calculable by a method outlined in the heat balance bulletin, but for this example we'll take it to be equal (approximately correct) to the feed pump work that has been calculated previously, namely 4.0 kJ/kg.

The general procedure is the same as in the previous example--just more of it. Now I count 11 unknown flows to be found by solving 11 simultaneous equations. The equations are

$$\mathrm{WPRT} = \mathrm{WNE} + \mathrm{R1}\cdot\mathrm{WEX1} + \mathrm{R2}\cdot\mathrm{WEX2} \tag{3.11}$$

$$\mathrm{WGT} = 3600 + 0.09\ \mathrm{WBO} \tag{3.12}$$

$$\mathrm{WEX1} = \frac{\mathrm{HH1} - \mathrm{HCOND}}{\mathrm{HEX1} - \mathrm{HDR}}\ \mathrm{WCOND} + \frac{\mathrm{HDR} - \mathrm{HH2}}{\mathrm{HEX1} - \mathrm{HDR}}\ (\mathrm{WSAH}+\mathrm{WFOH}) \tag{3.13}$$

$$\mathrm{WEX2} = \frac{\mathrm{HH2} - \mathrm{HFPT}}{\mathrm{HEX2} - \mathrm{HH2}}\ \mathrm{WFPT} + \frac{\mathrm{HH2} - \mathrm{HH1}}{\mathrm{HEX2} - \mathrm{HH2}}\ \mathrm{WCOND} + \frac{\mathrm{HH2} - \mathrm{HBO}}{\mathrm{HEX2} - \mathrm{HH2}}\ \mathrm{WRC} + \mathrm{WSAH} + \mathrm{WFOH} \tag{3.14}$$

$$\mathrm{WFPT} = 0.06\ (\mathrm{WFP}) \tag{3.15}$$

$$\mathrm{WCOND} = \mathrm{WPRT} + \mathrm{WGT} - \mathrm{WEX1} - \mathrm{WEX2} + \mathrm{WDR} \tag{3.16}$$

$$\mathrm{WBO} = \mathrm{WFP} - \mathrm{WRC} \tag{3.17}$$

$$\mathrm{WFP} = \mathrm{WEX2} + \mathrm{WFPT} + \mathrm{WCOND} + \mathrm{WRC} - \mathrm{WSAH} - \mathrm{WFOH} \tag{3.18}$$

$$\mathrm{WSAH} = 0.04\ \mathrm{WBO} \tag{3.19}$$

$$\mathrm{WFOH} = 0.005\ \mathrm{WBO} \tag{3.20}$$

$$\mathrm{WDR} = \mathrm{WEX1} + \mathrm{WFOH} + \mathrm{WSAH} \tag{3.21}$$

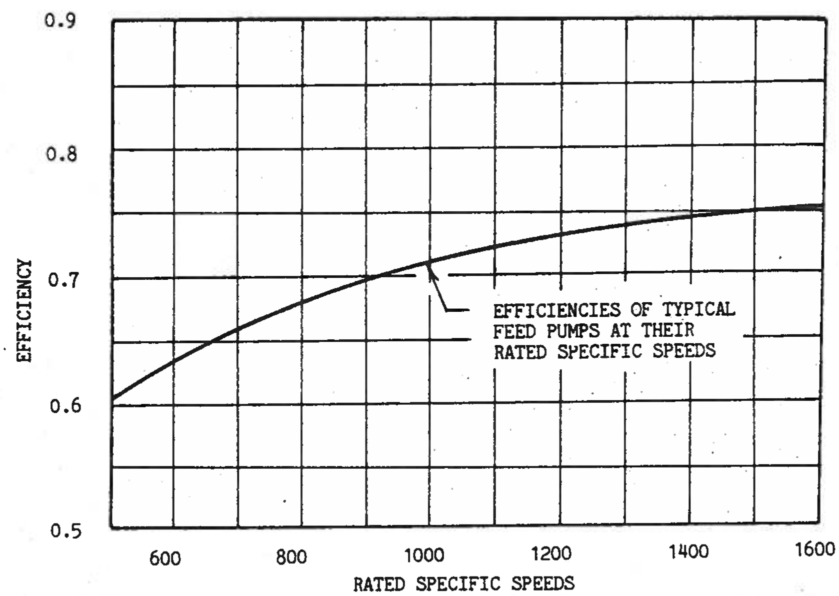
Using the same iterative technique explained in the previous example, I find the following values:

WPRT = 114,295	WFPT = 8,643	WRC = 5,000
WGT = 16,115	WCOND = 130,558	WSAH = 5,562
WEX1 = 13,906	WBO = 139,053	WFOH = 695
WEX2 = 6,109	WFP = 144,053	WDR = 20,164

3.5 USING THE SNAME HEAT BALANCE BULLETIN (Reference 1)

The principal value of the SNAME bulletin is as a source of data and of standard formulas for the component flows. Therefore, if you are to do a heat balance, that step 2 of Section 3.2 will depend largely on what you find in the bulletin. In some cases, the formulas given are simple rules-of-thumb similar to those used in this chapter, but generally they are more complex and you will meet some complications that I have not included. My expectation is that you can successfully cross the gap to the bulletin formulas, armed with an understanding of the message to this point, and with a decent reading ability.

But a word about non-linearities before you drop this report to plunge into the bodacious bulletin. These do occur, and can cause extra work in the solution phase (step (3) of Section 3.2). Take, for example, the calculation of steam flow to the feed pump turbine. This is proportional to the power required by the feed pump, which is a product of pump flow and pump head divided by pump efficiency. Pump head is a constant in a preliminary heat balance, so amend that to pump power proportional to flow divided by efficiency. Efficiency is a function of <u>specific speed</u>, a parameter proportional to the square root of the flow. The efficiency vs specific speed curve looks like Figure 3.3 so you can see that pump power, hence pump turbine steam flow (WFPT in the examples), is indeed a non-linear function of flow through the pump (WFP). This spoils everything, it would seem, for your computer won't solve a set of non-linear equations. However, some extra work fixes things, as the following outlines:



1. Start with an estimated value of feed pump flow, and use this value to calculate an estimated turbine steam flow, following the procedure outlined in the bulletin.

2. Using those two estimated flows, solve for the coefficient c in WFPT = c•WFP.

3. Use the value of c as a constant coefficient in solving the set of flow equations, which of course are now linear.

4. Go back to step 2 and solve for c again with the new values of WFPT and WFP. It is likely to be different, and unless you decide that the difference is trivial, repeat step 3 with the new value.

5. Do step 4 again, and around you go, repeating until the change in c does become insignificant.

At the same time, you should be looking at all other coefficients for non-linearities of this nature, and eliminating them at the same time in the same way.

3.6 REFERENCES

1. Society of Naval Architects and Marine Engineers Technical and Research Bulletin 3-11, <u>Marine Steam Power Plant Heat Balance Practices</u>.

CHAPTER 4

LATER HEAT BALANCE PROCEDURE

4.1 INTRODUCTION

As noted in Chapter 3, a preliminary heat balance is made with the assumption that components will be chosen or designed to suit the outcome of the heat balance. The most significant impact of this assumption occurs with the surface feedwater heaters. With them, you recall, a terminal temperature difference is chosen for each heat exchanger, which then directly gives the temperature of the product water from heater shell pressure. But once the heat balance is completed, the heaters are by implication designed — their surface areas and heat transfer coefficients are determined. If you should then do a heat balance at another condition (say one-half power) for the same plant, you no longer have the freedom to choose a terminal temperature difference, nor can you use the same one. You must calculate what that difference will be for the particular heater parameters fixed by the first balance. The tables are turned: the balance must suit the components, rather than the components suiting the balance.

The same complication exists whenever the plant components exist, even if only in the design stage. For example, even if a preliminary heat balance is to be revised after components have been specified, you will have to work with the characteristics that have been established (unless the whole thing is being thrown out). And, of course, if the plant actually is an existing one, the component characteristics must be respected.

The SNAME heat balance bulletin (Reference 1) doesn't make any distinction between the later and preliminary procedures. In fact, it is inconsistent in that it instructs you to use procedures that are strictly "later" in some instances; for example, for auxiliary turbines you are to decide a <u>rating</u>, and then calculate steam consumption for the heat balance power with the aid of a load factor supplied. Fine. When you come to do (say) a half-power heat balance, the rating is the same; you will automatically take care of the difference in performance by reading from the bulletin's curves an appropriate value of load factor. The bulletin eliminates the pain of transition in this way for several components, but not for all. The main purpose of this chapter is to fill in this deficiency, and also point out where "later" techniques are already used in bulletin-style calculations.

Incidentally, the term "later heat balance" is my invention; you won't find it generally used.

4.2 DIFFERENCES INTRODUCED BY THE LATER PROCESS

4.2.1 Propulsion Turbine

The turbine state line shifts as the load changes. Thus, for example, a half-power heat balance cannot use a full-power state line. The SNAME bulletin (Reference 1) doesn't help in estimating the change in the state line. The best resort is to the turbine manufacturer for a half-power state line. If this is impracticable, Section 4.5 of Reference 2 gives an outline of how the off-design conditions might be estimated.

The pressure at the extraction openings is a function of flow through the turbine, and therefore must be adjusted for off-design conditions. Reference 1 does help with this; see also the example in Section 4.3 here.

4.2.2 Auxiliary Turbines

The SNAME bulletin assumes that a rating will be set for auxiliary turbines, and provides load factors to take care of the off-design conditions. Thus, if you follow its instructions, you will be following "later" techniques in all heat balances

— preliminary or not — and so there is no new problem.

4.2.3 Electric Load

The handy formula in Reference 1 is but a rule-of-thumb, and gives an estimate only for the design condition. Even in a preliminary heat balance, an analyst may prefer to make a detailed compilation of the electric load, provided sufficient data is on hand. If so, it is relatively easy to adjust the load to suit a different power-plant condition. Also, there is an interaction between generator and heat balance that is not accounted for in the rule-of-thumb. For example, the power drawn by a condensate pump motor is a function of the flow through the pump. Consequently, generator load, and hence the steam consumption of its turbine, is affected by the balancing process. If precision demands it, the analyst should adjust the electric load to suit the conditions of load actually calculated.

4.2.4 Boilers

Boiler efficiency, superheated steam temperature, desuperheated steam temperature and pressure, and perhaps superheated steam pressure (it may be kept constant by automatic combustion control) are functions of load on the boiler, and all of these will affect the heat balance. The best thing is to have a set of characteristic curves for the boiler. See Figure 4.1 for an example. Or such a set for a boiler assumed to be similar can be used to estimate fractional changes from design conditions.

4.2.5 Surface-Type Heat Exchangers

The fundamental heat exchanger relation is

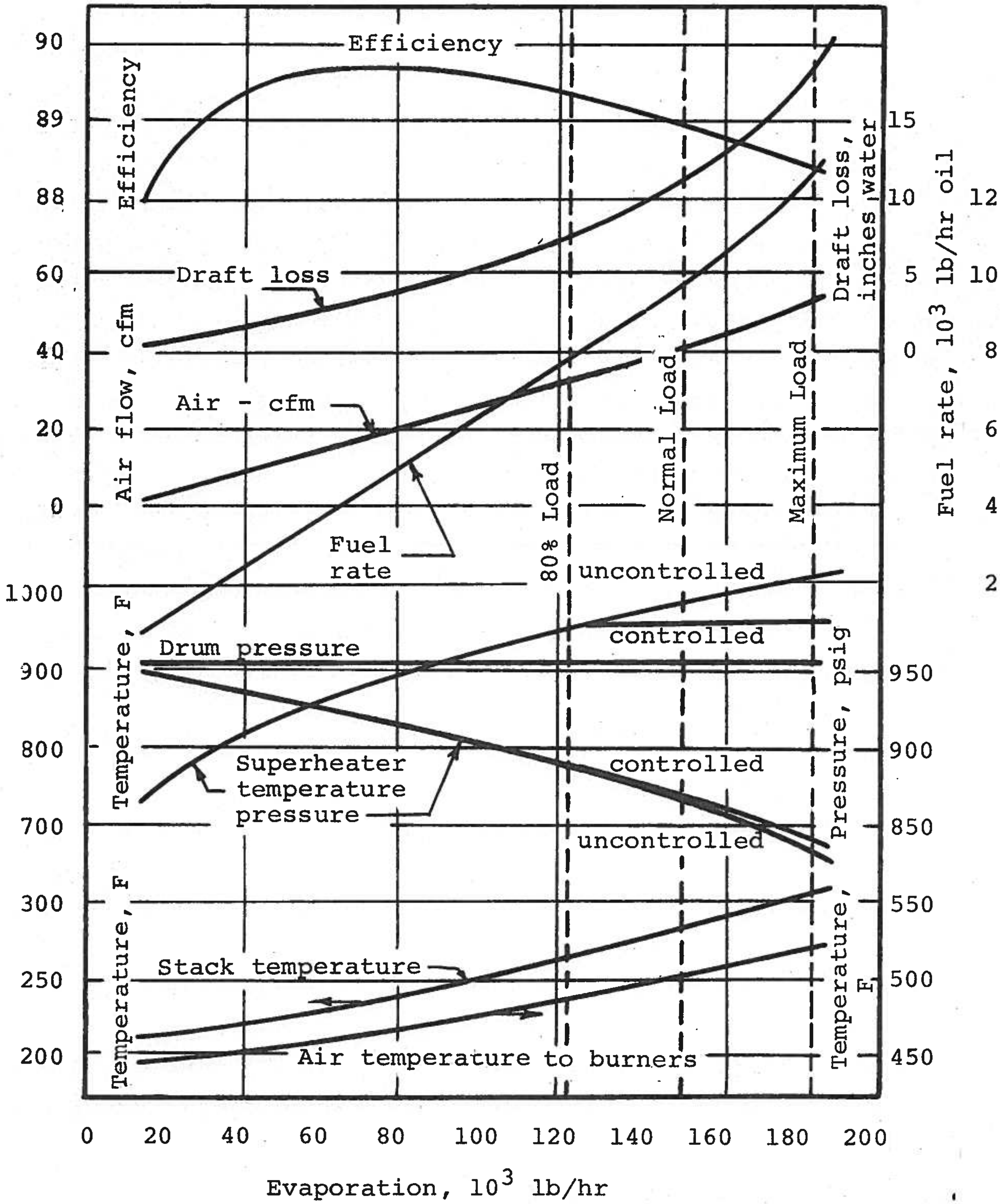
$$Q = U\ A\ \Delta T_m \tag{4.1}$$

where Q = rate of heat transfer, kJ/h

A = heat transfer area, m^2

U = overall heat transfer coefficient, kJ/h m^2 °C

ΔT_m = log mean temperature difference, °C



From instruction book for Foster Wheeler DSD type boiler

FIGURE 4.1 Example of Boiler Performance Curves

The meaning of ΔT_m is illustrated by Figure 4.2 which shows temperature distribution in a steam-to-water heater. Formula-wise

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\log_e \frac{\Delta T_1}{\Delta T_2}} \qquad (4.2)$$

the ΔT_1 and ΔT_2 are defined by the figure. Note that ΔT_2 is the terminal temperature difference. In a preliminary heat balance, ΔT_2 is specified, and ΔT and Q are found; ΔT_1 and ΔT_2 fix ΔT_m . The heat exchanger designer thus works to give the product $U A$ the necessary value. Once it is set, you are stuck with it. A is, of course, an invariable characteristic of the heater; U varies with load. For a given load (i.e. Q), ΔT_m must therefore be adjusted to suit the dictates of equation (4.1). The greater be Q , the greater must be ΔT_m and hence the greater must be terminal ΔT.

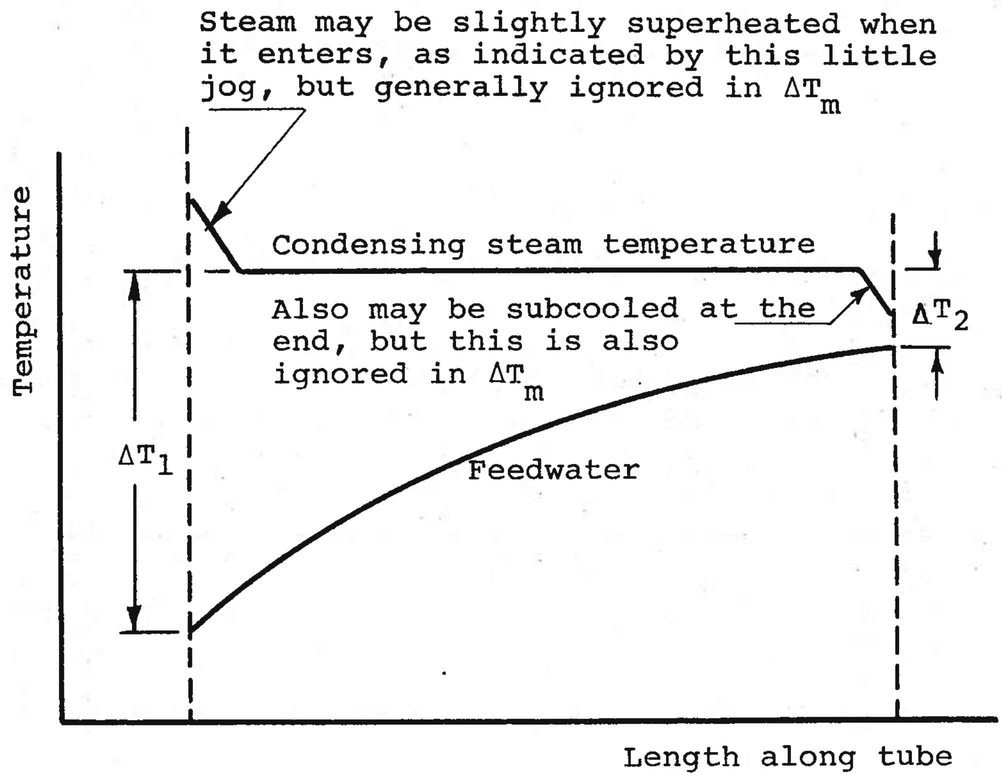
Q is the rate of heat transfer. It must equal the rate at which heat is given up by the steam, and must equal the rate at which it is absorbed by the feedwater. Put in equation form,

$$W_{STM} (h_{STM} - h_{DRAIN}) = W_{FEED} (h_{OUT} - h_{IN}) \qquad (4.3)$$

$$U A \Delta T_m = W_{FEED} (h_{OUT} - h_{IN}) \qquad (4.4)$$

The second can be expanded in terms of temperatures, using (4.2), and by expressing the liquid enthalpies as products of temperatures and the specific heat, C , of the liquid.

$$\frac{U A (T_{OUT} - T_{IN})}{\log_e \frac{T_{STM} - T_{IN}}{T_{STM} - T_{OUT}}} = W_{FEED} C (T_{OUT} - T_{IN}) \qquad (4.5)$$



ΔT_2 = terminal temperature difference

ΔT_1 = inlet temperature difference

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\log_e \frac{\Delta T_1}{\Delta T_2}}$$

This formula is strictly correct only for single-pass heaters. Can be used for multi-pass with a correction factor (see any text on heat exchanger design). Correction for two-pass is close to 1.0, and so is often ignored.

FIGURE 4.2 Significant Temperatures and Temperature Differences in Surface-Type Heater from which

$$\frac{T_{STM} - T_{IN}}{T_{STM} - T_{OUT}} = e^{\frac{U\ A}{W_{FEED}\ C}} \qquad (4.6)$$

In the later heat balance, you must satisfy (4.3), just as in the preliminary, and in addition have the temperatures satisfy equation (4.5) or its equivalent, (4.6).

U is a weak function of W_{FEED} and the temperatures. The relationships are given graphically in Reference 3, and a condensed version of the reference is given on the sheet STANDARD FACTORS that accompanies these notes.

Examples later in this chapter illustrate the use of these concepts.

4.2.6 Condensers

The condenser behaves fundamentally in the same way as the feedwater heater. The preceding discussion applies to it, indicating that the vacuum (i.e. saturation pressure corresponding to T_{STM} in equation (4.6)) may have to be calculated in a later heat balance.

4.2.7 Direct-Contact Feed Heaters

Temperature difference is no worry in a direct-contact heater, but you may have to calculate the shell pressure. Such a calculation is illustrated by example later. If a reducing valve is installed to regulate steam pressure to the heater, the problem is eliminated as long as pressure upstream of the valve is equal or greater than its set pressure. If it falls below, then the valve is wide open, and can do nothing to hold up the pressure. You must calculate.

4.2.8 Piping between Components

Piping pressure drops are closely proportional to square of the flow. The SNAME bulletin says to take pressure drop as a percentage of the initial pressure; this is an adequate beginning for the design condition, but in going to an off-design condition, variation of pressure drop according to the sqaure of the flow is called for. Later examples illustrate.

4.3 AN ILLUSTRATION OF CONTRASTS IN TECHNIQUES

Figure 4.3 shows semi-pictorially part of a simplified steam propulsion plant. A heat balance problem is to be worked that serves as a vehicle for contrasting some of the later and preliminary techniques. Let the figure serve to define the symbols. Suppose W_T is known and fixed, and that P_o and h_o are relatively invariant compared to other things, and so can be considered constant (to make the problem simpler). Find the temperature (or enthalpy h_1) of the feedwater leaving the heater. Since six other things are also unknown, namely P_2 , h_2 , h_3 , h_4 , W_E , and P_3 , seven relations are needed to solve. The table below given these relations as they would be for the two techniques.

	EQUATION	PRELIMINARY	LATER HEAT BALANCE
1	Heat balance at heater	Same as later	$W_E h_3 + (W_T - W_E) h_o = (W_T - W_E) h_1 + W_E h_4$
2	Extraction pressure	P_2 = constant given, or worked back from a given value of h_1	$P_2 = P_2 (W_T - W_E)$ i.e. P_2 is a function of flow to condenser. See text following this table
3	Shell pressure	$P_3 = 0.9\ P_2$ i.e. 10% loss	$P_3 = P_3 (P_2, W_E)$ Usually $P_3 = P_2 - k \cdot W_E^2$ where k is constant determined by pipe size and length
4	Extraction enthalpy	Same as later	$h_2 = h_2 (P_2)$ i.e. determined by intersection of turbine state line with P_2 contour on chart

5	Shell enthalpy	Same as later	$h_3 = h_2$ i.e. assume no loss of enthalpy in flow process
6	Drain enthalpy	Same as later	$h_4 = h_f(P_3)$ i.e. assume drain enthalpy is sat. liquid h at shell pressure
7	Heat transfer capability of heater	$h_1 = h_4 - \Delta T_{TERM}$ assuming $\Delta h = \Delta T$ in range of interest	$h_1 = h_1(h_3, h_4, h_O, W_T - W_E)$ outlet T or h depends capability of heater to transfer heat. See 4.2.5

Variation of extraction pressure with flow is depicted by Reference 1, and Reference 2 gives an explanation. A qualitative explanation is also given here with the aid of Figure 4.4. Fluid flows via two nozzles in series from a high fixed pressure to a low fixed pressure. Sum of pressure drops across the two must sum to the total ΔP, no matter what the flow. The upstream nozzle has a variable area for control of flow; propulsion turbines are typically controlled this way — by adjusting first stage nozzle area. The sketch is thus a crude model of a two-stage propulsion turbine. Suppose the control area is reduced. This shifts the nozzle characteristic to the dashed line, since for reduced area there must be a higher ΔP for any flow. But remember that the <u>total</u> ΔP must be the same, making it obvious that the pressure distribution shifts as flow is changed, and thus also obvious that extraction pressure is a function of flow.

4.4 NUMERICAL ILLUSTRATIONS OF LATER TECHNIQUES

4.4.1 <u>A Direct-Contact Heater Problem</u>

This illustrates the problem that might occur in calculating the shell pressure in a direct-contact heater. The pressure is established by the rate of condensation in the heater — the "heat balance" process — but it must also equal the pressure

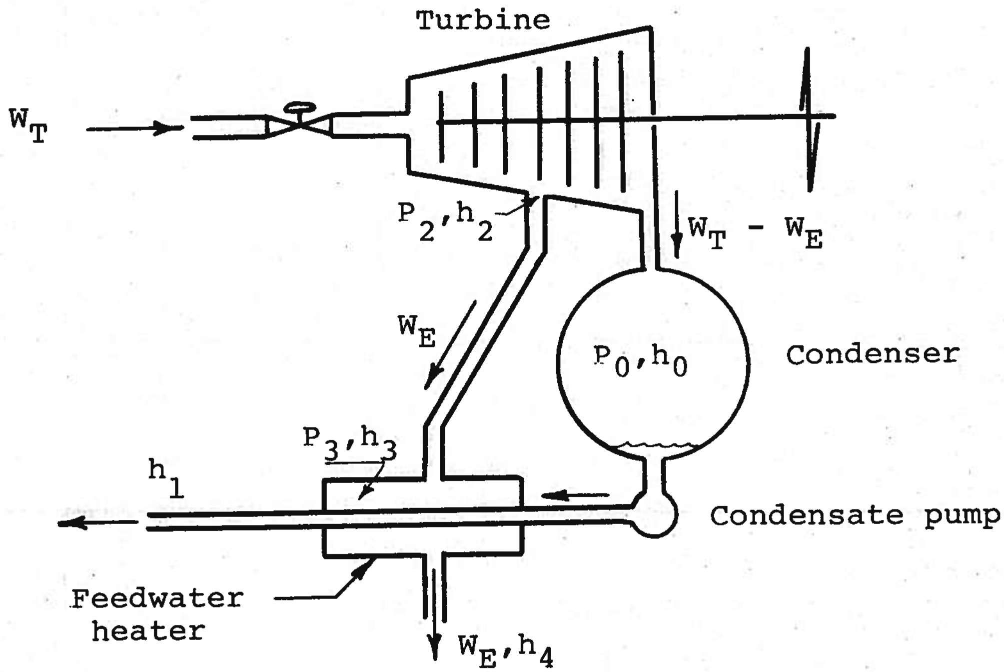


FIGURE 4.3 To Illustrate Example Problem in Section 4.3

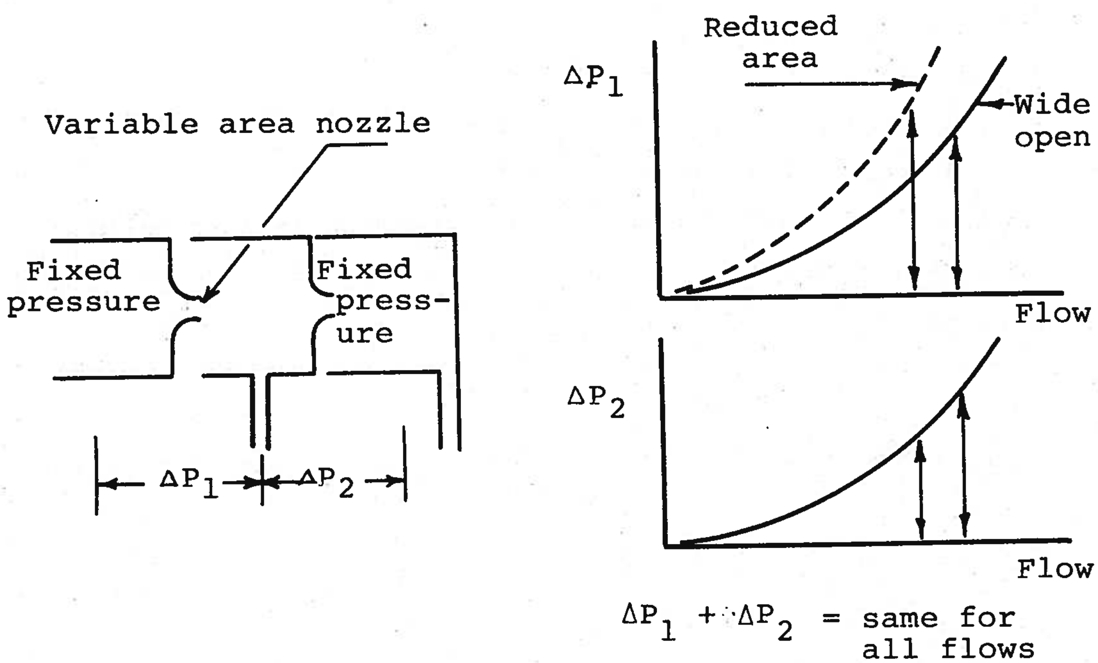


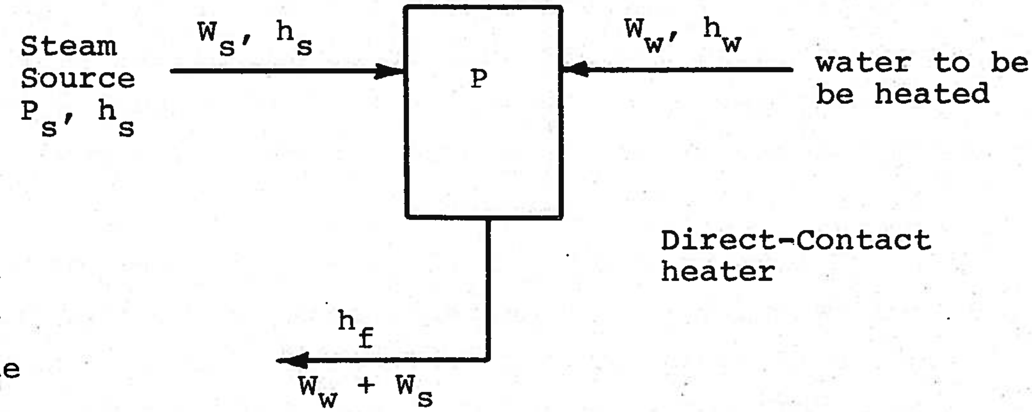
FIGURE 4.4 Nozzles in Series

that the steam source can provide. Here the source pressure is taken to be constant, but unlike the preliminary procedure, the pressure drop between source and heater is a function of flow, assumed to be proportional to square of the flow, as in an actual pipe. The problem is to find temperature of water leaving, and the amount of steam used. Look at the top sketch in Figure 4.5

The technique is trial-and-error: choose a shell pressure, then calculate the resultant steam consumption according to the heat balance relations (i.e. conservation of mass and energy), and according to the pipe flow characteristic. A pressure that gives the same consumption from both is the correct one. This is done systematically in the table below.

						SOURCE
1	P	0.250	0.270	0.268	0.2685	Assumed values
2	ΔP	0.050	0.030	0.032	0.0315	0.30- 1
3	W_s	7071	5477	5657	5612	[2 /K]$^{1/2}$
4	h_f	535.4	546.3	545.2	545.5	Sat liquid h at P = 1
5	Δh_w	116.4	127.3	126.2	126.5	4 - 419
6	$\times 10^{-6}$ Q_w	11.64	12.73	12.62	12.65	Heat absorbed by W_w = $W_w \times \Delta h_w = W_w \times$ 5
7	Δh_s	2264.6	2253.7	2254.8	2254.5	2800 - 4
8	W_s	5140	5648	5597	5611	6 / 7

Lines 3 and 8 in the 0.2685 column are very close, so accept 0.2685 as the correct pressure; water temperature leaving is thus 129.8 C, the saturation temperature for 0.2685.



P = Pressure

h = enthalpy

W = flow rate

W_w = 100,000 kg/h

h_w = 419 kJ/kg

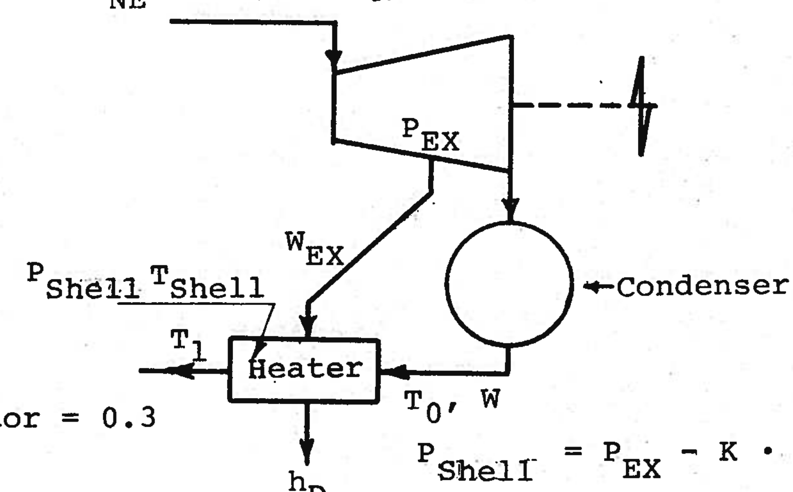
P_s = 0.30 MPa

h_s = 2800 kJ/kg

$P = P_s - K \cdot W_s^2$ (pressure drop in pipe)

$K = 10^{-3}$ Pa/(kg/h)^2

W_{NE} = 40,000 kg/h



P_{EX} = 0.17 MPa
W_{NE} = 40,000
and W_{EX} = 0

h_{EX} = 2750 kJ/kg

Replacement factor = 0.3

$\Delta T_w = T_1 - T_o$

T_o = 40C

$\Delta h_s = h_{EX} - h_o$

$P_{Shell} = P_{EX} - K \cdot W_{EX}^2$

$K = 1.5\times10^{-3}$Pa/(kg/h)^2

$\Delta T_{TERM} = T_{Shell} - T_1$

P_{COND} = 0.0075 MPa

FIGURE 4.5 Sketches for Numerical Examples

4.4.2 A Surface-Type Heater Problem

Look at Figure 4.5, bottom sketch. A surface type feedwater heater uses extraction steam to heat 110F water. The heater is designed to heat 10^5 lb/hr water from 160F to 240F with shell pressure of 30 psia (T_S = 250F), and with water velocity in tubes of 6.0 ft/sec. Its surface area is 293 ft^2, and U at design condition is 750 Btu/hr ft^2 °F (gotten from STANDARD FACTORS sheet, based on ΔT_m of 36.4°F, F_{ft} of 221, and multiplier of 1.0. (Now you see that I've had to relapse into English units, this because the data comes that way.) The 293 ft^2 area is 27.22 m^2, and the 750 Btu/hr ft^2 U is 15,332 kJ/h m^2 °C. These numbers are used in the subsequent calculation, with U assumed proportional to the square root of water flow, with base flow being 40,000 kg/h. This last is a reasonable approximation to the STANDARD FACTORS curve.

Turbine flow and extraction-point conditions are given at zero extraction flow, and the pipe pressure-flow characteristic is given. Condenser pressure is estimated to be 0.0075 MPa at all conditions encountered in the problem (a simplification). See the figure for the given numerical values, and the definitions of symbols.

The problem is to find how this heater performs under conditions specified in the sketch (which differ from design conditions), i.e. how hot it heats the water, and how much steam it takes. The procedure can be the same as in 4.4.1, causing a second level of iteration. The procedure to be followed is this:

1. Estimate a value of P_{EX} from the knowledge that it will be lower than the 0.017 MPa shown in Figure 4.5. Try 5% lower than the 0.017 MPa given.
2. Estimate a value of the extraction flow W_{EX}. Try something on the order of 10% of W_{NE}.
3. Following these two estimates, proceed in straight-forward fashion to calculate all pressures, temperatures, flows, rates of heat transferred, etc. The step-by-step details

of this procedure can be seen by tracing through the calculation block; each calculation is explained in the SOURCE column and its associated footnotes.

4. Note that heat given up by the steam (line 9) and the heat transferred by the heater (line 19) are both calculated. These must be equal, hence inequality sends you back to the top of the column to make changes and begin again.

5. Note that after the first iteration, the value of P_{EX} is left unchanged, and that it is left unchanged until the two Qs are approximately equal. P_{EX} could be changed every time, but I'm illustrating here the technique of making the big adjustment before making the small adjustments. If you try to adjust everything at once, you may become confused, bemused, and bewildered, all of which are early symptoms of dismay. Hence the "second level of iteration" mentioned earlier. Take care of the big ones first, then the minor changes.

6. When it's time to adjust P_{EX}, use the "flow-to-the-following-stage" proportionality. This approximation is the best that can be had here in the absence of exact information (as might be given by a steam turbine manufacturer) on how extraction pressure behaves. It assumes that the pressure difference between P_{EX} and the condenser pressure is proportional to the flow that goes past the extraction opening (i.e. that is not extracted). Details for the illustrative calculation are found in the first footnote following the calculation block.

7. After readjusting P_{EX}, the complete balancing procedure is repeated until the two Qs are once more equal, then P_{EX} is checked again, Qs balanced again, etc. etc. But it shouldn't be this bad. Note in the following calculations that I did not have to change P_{EX} again after the first adjustment.

						SOURCE
1	P_{EX}	0.1600	0.1600	0.1586	0.1581	Note
2	W_{EX}	4000	4200	4300	4220	Estimated
3	K	1.5×10^{-3}				Given constant
4	ΔP	0.0240	0.0265	0.0277	0.0267	$(3)(2)^2$
5	P_{SHELL}	0.1360	0.1335	0.1309	0.1314	$(1)-(4)$
6	T_{SHELL}	109.1	107.9	107.3	107.4	T_{SAT} for (5)
7	h_D	457.5	452.4	450.0	450.5	h_f for (5)
8	h_{EX}	2750				Given, note
9	$Q(10^3)$	9,170	9,650	9,890	9,704	$(2)((8)-(7))$
10	R	0.30				Given, note
11	W	37,200	37,060	36,990	37,046	$W_{NE}\cdot(1.0-(10))(2)$
12	T_0	40				Given
13	T_1	98.4	101.7	103.4	102.1	$(12)+(9)/(11)$
14	ΔT_{TERM}	10.7	6.2	3.9	5.3	$(6)-(13)$
15	ΔT_0	69.1	67.9	67.3	67.4	$(6)-(12)$
16	ΔT_{MEAN}	31.3	25.8	22.3	24.4	Note
17	U	14,786	14,758	14,744	14,755	Note
18	A	27.22				Given constant
19	$Q\ (10^3)$	13,066	10,356	8,933	9,808	$(16)(17)(18)$

Notes:

Line (1) First value of P_{EX} (0.1600) is estimated. The values used after the second calculation is found by ((11) $/W_{NE}$) (0.17 - 0.0075) + 0.0075.

Line 8 Extraction enthalpy is a function of P_{EX}, and so should undergo slight changes as P_{EX} goes from 0.16 to 0.1586 to 0.1581, etc. Kept constant here to simplify the demonstration. Would need the turbine state line.

Line 10 Replacement factor is a function of h_{EX}, but since the latter is kept constant in this demonstration, R is kept constant also.

Line 13 Specific heat of water near 100 C is 4.22.

Line 16 From the formula for log mean temperature difference, which here is (15 - 14)/ ln (15/14)

Line 17 Velocity of water in the heater tubes is proportional to the flow (line 11) rate through them. U is taken to be proportional to square root of flow, with U=15,332 when W=40,000.

Line 19 Note that the demonstration is not carried out to an exact equality of lines 9 and line 21. But given the uncertainties in U and h_{EX}, and in the scheme for estimating P_{EX}, an acceptable balance is obtained.

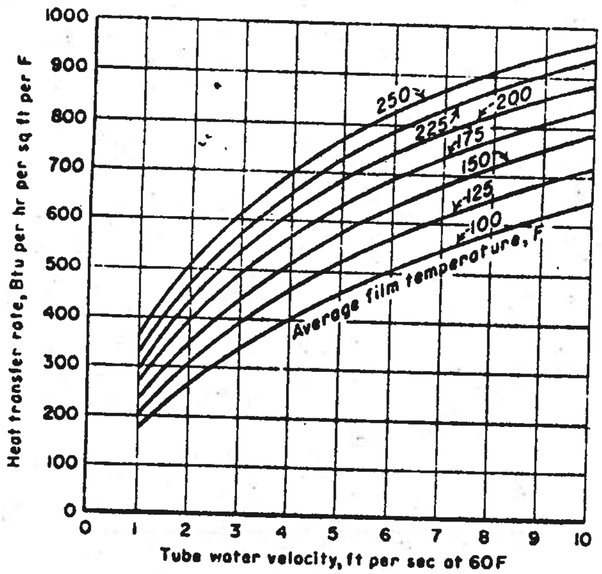
4.5 REFERENCES

1. Recommended Practices for Preparing Marine Steam Power Plant Heat Balances, Technical and Research Bulletin 3-11, Society of Naval Architects and Marine Engineers, 1971.

2. Jens T. Holm and J. B. Woodward III, "Steam Power Plant Thermodynamics," Chapter 2 of Marine Engineering, Society of Naval Architects and Marine Engineers, 1971.

3. Stnadards of Bleeder Heater Manufacturers Association.

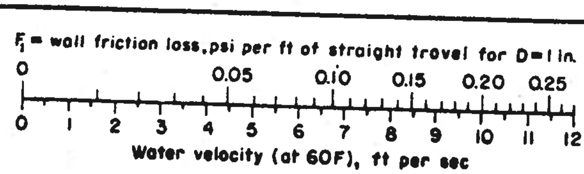
Standard factors . . . *they're basis for sizing heaters*

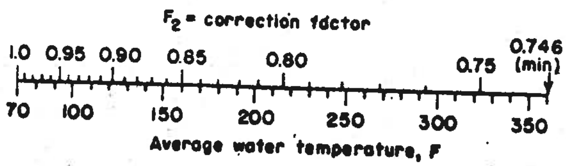


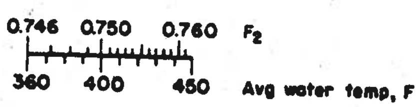
I—Multipliers For Base Heat-Transfer Rates

(For tube OD 5/8 to 1 in. incl)

BWG	TUBE MATERIAL: Ars-cu	Adm	90/10 Cu-ni	80/20 Cu-ni	70/30 Cu-ni	Monel
18	1.00	1.00	0.97	0.95	0.92	0.89
17	1.00	1.00	0.94	0.91	0.87	0.85
16	1.00	1.00	0.91	0.88	0.84	0.82
15	1.00	0.99	0.89	0.86	0.82	0.79
14	1.00	0.96	0.85	0.82	0.77	0.75
13	0.98	0.93	0.81	0.78	0.73	0.70
12	0.95	0.90	0.77	0.73	0.68	0.65
11	0.92	0.87	0.74	0.70	0.65	0.62
10	0.89	0.83	0.69	0.66	0.60	0.58
9	0.85	0.80	0.65	0.62	0.56	0.54







II—Feedwater-Heater Tube Constants

Tube BWG	5/8-in. OD D	5/8-in. OD $D^{1.24}$	3/4-in. OD D	3/4-in. OD $D^{1.24}$	7/8-in. OD D	7/8-in. OD $D^{1.24}$
18	0.527	0.452	0.652	0.589	0.777	0.731
17	0.509	0.433	0.634	0.567	0.759	0.710
16	0.495	0.418	0.620	0.554	0.745	0.695
15	0.481	0.404	0.606	0.538	0.731	0.678
14	0.458	0.380	0.584	0.514	0.709	0.652
13	—	—	0.560	0.488	0.685	0.625
12	—	—	0.532	0.467	0.657	0.594
11	—	—	—	—	0.635	0.570
10	—	—	—	—	0.607	0.538

Basis for designing feedwater heaters is outlined in the *Standards of Bleeder Heater Manufacturers Assn, Inc.* Data abstracted from these standards are given in the left-hand column. The graph gives the heat-transfer rates to be used for given tube water velocities and average film temperatures. The latter is calculated as:

$$F_{ft} = t_s - (0.8 \times t_{lmtd})$$

where F_{ft} = average film temperature, F
t_s = saturation temperature of steam, F
t_{lmtd} = logarithmic mean temperature difference, F

Heat-transfer rates in excess of those for 250-F film temperature should not be used. These rates apply when steam is supplied from: turbine exhaust, turbine bleed point, evaporator vapor outlet, boiler. Water flowing through the tubes should be feedwater as would be found normally in a closed cycle.

The *Standards* specify that condensing zones in heaters and simple condensing heaters should not be designed for a closer approach to saturated steam temperature than 2 F. The heat-transfer rates of the graph are subject to several corrections. If either steam or water differs from the standard specified above, multiply the rates by 0.85. If both differ, multiply by 0.72.

Finally, refer to Table I and find a multiplier corresponding to the tube thickness and tube material, to use on the rate from the graph. The multipliers grow smaller with decreasing tube size and decreasing copper content. Tube material must be chosen to cope with the corrosion conditions that will be encountered.

Subcooling zones in heaters shouldn't be designed for a terminal difference of less than 10 F between the shell drain and the tube inlet.

Pressure drop of feedwater flowing through seamless drawn tubing in the heaters is figured as:

$$\Delta p = F_1 F_2 (L + 5.5D) N / D^{1.24}$$

where Δp = pressure drop, psi
F_1 = correction factor from scale, *left column*
F_2 = correction factor from scale, *left column*
L = total linear feet of tubing divided by number of tube holes in one tube sheet
D = tube ID, in., from Table II
N = number of passes

In figuring F_2 the average water temperature is equal to $t_s - t_{lmtd}$.

Pressure drop through heaters is important in determining condensate- and feed-pump pressures needed.

Heat exchangers for general process and refinery work are covered in the *Standards of Tubular Exchangers Manufacturers Association*. Because of the wide variety of fluids covered, determinations of heat-transfer rates are more detailed than those for feedwater heaters.

The method involves selecting film coefficients on each side of the tubing, fouling resistances on each side, resistance of the tube wall corrected for thickness, and ratio of internal to external tube surfaces. These factors vary with tube material and the fluids used.

Friction factors based on Reynolds numbers, which in turn involve fluid viscosity and density, must be determined in advance of calculating the pressure drop through the heater tubes.

Thermal conductivities and specific heats of the gases, vapors and liquids must also be considered.