OPTIMIZATION OF NAVAL PROPULSION MACHINERY

by

DOUGLASS F. HAYMAN, JR.

B.S., United States Naval Academy

(1956)

HAROLD H. OTTO

B.E., Nova Scotia Technical College

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SUBMITTED IN PARTIAL FULFILLMENT

OF THE REQUIREMENTS FOR THE

DEGREE OF NAVAL ENGINEER

AND THE DEGREE OF

MASTER OF SCIENCE IN NAVAL ARCHITECTURE

AND MARINE ENGINEERING

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

May 1962

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Department of Naval Architecture and Marine Engineering, 19 May 1962

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Thesis Supervisor

Accepted by ______________________________
Chairman, Departmental Committee on Graduate Students

OPTIMIZATION OF NAVAL PROPULSION MACHINERY by DOUGLASS F. HAYMAN, JR., USN and HAROLD H. OTTO, RCN. Submitted to the Department of Naval Architecture and Marine Engineering on 19 May, 1962 in partial fulfillment of the requirements for the Master of Science degree in Naval Architecture and Marine Engineering and the Professional degree, Naval Engineer.

ABSTRACT

The optimum naval propulsion plant is considered to be the one with the least total weight of machinery plus fuel. Perturbations of a modern destroyer propulsion cycle, with standard equipment components, are considered. Boiler pressure, condenser pressure, low pressure turbine exhaust annulus area, condenser surface, and leaving loss are considered variable. Equations are derived which express the variations in weight of important components. Availability balance methods are applied in order to relate component efficiencies to fuel weight. Theoretical and numerical proof is given that leaving loss can be optimized on the basis of minimum turbine and condenser weight, independent of the rest of the cycle. This reduces the computations necessary in "brute force" analysis by an order of magnitude. As an example of the method, an availability balance is made for DLG-6 at cruising condition. Using 1050°F steam, boiler pressures from 800 psia to 1600 psia, and a broad range of condenser-L.P. turbine combinations, best parameters are found for ranges of 3,000, 5,000, 7,000, and 10,000 miles. Optimum condenser pressure is found to be fairly constant at 1.35″ Hg. Abs., for the cruising condition and 75°F cooling water. The example studied indicates that standardization of naval propulsion plants at 1200 psia is on the high side of the optimum.

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NOMENCLATURE

A_a	=	Last high pressure annulus area in main engine. This is based on an arbitrary 310 f.p.s. axial velocity.
BTU	=	British thermal unit
b	=	availability
C	=	constant
D	=	diameter
e_m	=	mechanical efficiency of the turbine and reduction gearing
F	=	Farenheit
f	=	friction factor
G	=	mass rate of flow
g	=	acceleration of gravity
HHV	=	higher heating value of fuel
I	=	irreversibility
J	=	heat-work conversion factor
K	=	a constant
L	=	a characteristic dimension
lb	=	pound mass
n	=	number of weight variable machinery components considered
P	=	pump work
p	=	pressure
p'	=	pressure after main feed pump in psig.
p_o	=	1200 psia
psia	=	absolute pressure, pounds per square inch
psig	=	gage pressure, pounds per square inch

ea = [illegible]

e = [illegible]

ex = [illegible]

[illegible] = [illegible]

f = [illegible]

h = [illegible]

hs = boiler heating surface

i = inlet

j = j^{th} plant component

l = leakage

m = arithmetic mean

mfp = main feed pump

p = main feed pump

pp = boiler pressure parts

pt = main feed pump turbine

s = saturated

sl = steam line

SH = superheated

t = main engines

th = [illegible]

v = [illegible]

C	=	condenser
cl	=	condensate line
cw	=	cooling water
C_{wet}	=	condenser - wet
DSH	=	desuperheated
da	=	deaerator
e	=	exhaust
ex	=	exhaust
F	=	fuel
f	=	make up feedwater
h	=	exhaust hood
hs	=	boiler heating surface
i	=	inlet
j	=	j^{th} plant component
l	=	leakage
m	=	arithmetic mean
mfp	=	main feed pump
p	=	main feed pump
pp	=	boiler pressure parts
pt	=	main feed pump turbine
s	=	saturated
sl	=	steam line
SH	=	superheated
t	=	main engines
th	=	thermal
v	=	sum of variables
w	=	boiler water

w+r	=	walls and refractory
wtr	=	water

Greek and other

Δb	=	change in availability in flow through component indicated by subscript
Δh	=	change in enthalpy in flow through component indicated by subscript
ΔT	=	temperature difference
η	=	efficiency
η_A	=	thermal efficiency of heat added
η_b	=	boiler available energy efficiency
ρ	=	density
Σ	=	the sum of
σ	=	allowable stress
1	=	state point at turbine inlet
$1'$	=	state point at turbine inlet, ideal cycle
2	=	state point at turbine exhaust annulus and condenser inlet
$2'$	=	state point at turbine exhaust annulus and condenser inlet for ideal cycle
2^o	=	stagnation state point for turbine exhaust
3	=	state point of condensate at condenser outlet
$3'$	=	state point of condensate at condenser outlet for ideal cycle
4	=	state point at boiler inlet
$4'$	=	state point at boiler inlet in ideal cycle
$\sim$		is proportional to
$\cong$		is approximately equal to

ACKNOWLEDGMENTS

We are most grateful to Professor C. R. Soderberg for his generosity with his time and patience and for keeping us ever mindful of basic thermodynamic principles, and to Professor S. C. Powell for sharing with us some of his experiences with problems related to ours. Credit is due to Commander J. R. Baylis, USN, for inspiring our curiosity in this problem, and for his continued interest and valued suggestions as the thesis developed. We thank Professor E. G. Frankel for several interesting discussions. Finally, we would take this opportunity to salute our wives, Judith Timlin Hayman and Eleanor E. Otto, for their heretofore unrecognized contributions to our academic work.

I. INTRODUCTION

In the design of naval ships, the portion of total displacement which can be allocated to the various payload functions such as armament, or information gathering devices, is sharply limited. This is particularly the case in destroyer types, which are highly powered and have a large percentage of total displacement devoted to machinery weight. A clear challenge exists to meet certain specifications, with the minimum weight of machinery plus fuel. The principal specifications which are set for the machinery designer are:

(1) Cruising speed and shaft horsepower,

(2) Range at cruising speed,

(3) Maximum shaft horsepower,

(4) Geared steam turbine power plant (authors' assumption).

Efforts to meet this challenge are surpassed in number, at least, by those devoted to a similar one in land power generation. Here power plants of comparable rating are assessed, but the problem has a variation. Minimum power cost is the goal, and not minimum total weight. In both cases efficiency of conversion of fuel into power is of prime importance. In one case the most profitable combination of cost of fuel per unit of shaft work and cost of machinery over the amortization period is sought. In the other, weight of fuel per unit of shaft work plus total weight of machinery is optimized for a given cruising range. The relative penalties in plant cost or weight which we

will pay to obtain fuel economy are different in each case and different optima result. Care must thus be exercised in the application of central station methods to naval steam plant evaluation.

Standardization of main engine top temperature and pressure, and steam conditions for certain auxiliaries is desirable for naval service. Making a great number of units to suit the same steam conditions stands to yield lower cost, better reliability, and higher level of crew training than would fitting the best steam conditions to each individual class of ship. The marine engineer should be able to estimate quantitatively the weight penalty paid for such standardization, as optimum steam conditions change with ships' missions.

The problem undertaken in this paper is to determine which combinations of throttle steam conditions, L.P. turbine exit velocities, and condenser vacua yield minimum combined weight of propulsion machinery and fuel for a given mission. Papers by Meigs [1]* and Michel [2] indicate considerations of the U.S. Navy in solving this problem. No coordinated plant optimization method is published although methods for determining separately certain characteristics of some individual components are given in the Bureau of Ships Design Data Book [3]. "An Analysis of Steam Propulsion Plants for Minimum Weight" by White and Smith [4]

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*-Numbers in brackets refer to references listed at the end of this paper.

gives some useful expressions for component machinery weights. It is unfortunately based on outdated machinery and does not consider fuel consumption.

Semi-analytic optimization methods were proposed by Smyth [5], Wilson and Malouf [6], and Wilson [7], and demonstrated numerically in an example by Wooden and House [8]. These analytic methods are said to be more efficient than the "brute force" method of computing a heat balance for each variation which the designer chooses to consider. They have a disadvantage in that they give the analyst little knowledge of the magnitude of the changes due to specific perturbations. A mathematical method might, for example, select as "optimum" a 1200 psig. plant which was one-half ton lighter than a 1000 psig. plant. In the case of such a flat optimum, the engineer who knew the whole story might base a choice of 1000 psig. steam on factors other than weight, rather than pick the higher pressure to make an insignificant weight saving.

It is the feeling of the authors that analytic methods of weight optimization are an interesting mathematical exercise, but that a more detailed analysis is justified by the importance of making a correct choice of steam conditions and desirable for the insight it yields into the nature of the changes brought about by the various perturbations.

This paper introduces methods of analysis using the concept of available energy, in addition to utilizing the conventional heat balance.

It is shown that the portion of total available energy lost

by irreversibilities in a particular piece of machinery causes that component to be charged with a corresponding weight in increased fuel consumption.

It may be seen that some components are constrained to have constant irreversibilities and need not be considered repeatedly in detail. Some have large irreversibilities and must be considered closely while others are so small that they may be ignored.

The thermodynamic basis of analysis through available energy and its application to the optimization problem are the subjects of the next section.

II. PROCEDURE - AVAILABILITY CONCEPTS

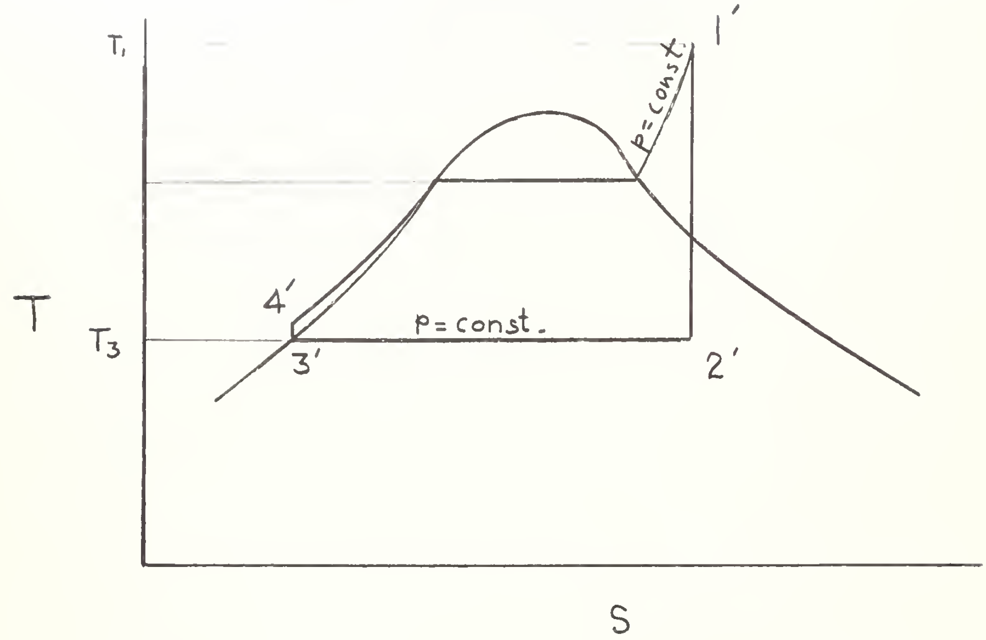
In evaluating a power plant, the engineer is inescapably concerned with overall efficiency, whether his object is cost or weight minimization. The maximum efficiency possible for a heat engine operating between two reservoirs of constant and uniform temperature is that of a reversible engine,

$$\eta = \frac{T_1 - T_2}{T_1}, \qquad (1)$$

where T_1 is the absolute temperature of the heat source, and T_2 is that of the sink. This particular expression gives the Carnot engine efficiency. All reversible engines operating between these same temperatures have identical efficiencies. Formula (1) thus indicates the upper limit of thermal efficiency of a heat engine. Heat addition is not completely accomplished at constant temperature in any practical steam cycle. Instead of the Carnot cycle, most steam power plants are based on variations of the ideal Rankine cycle. An ideal Rankine cycle is shown in Figure I.

Figure I

Ideal Rankine Cycle



Thermal efficiency, less than Carnot, must be computed from the formula,

$$\eta_{th} = \frac{\text{Net work}}{\text{Heat supplied}}$$

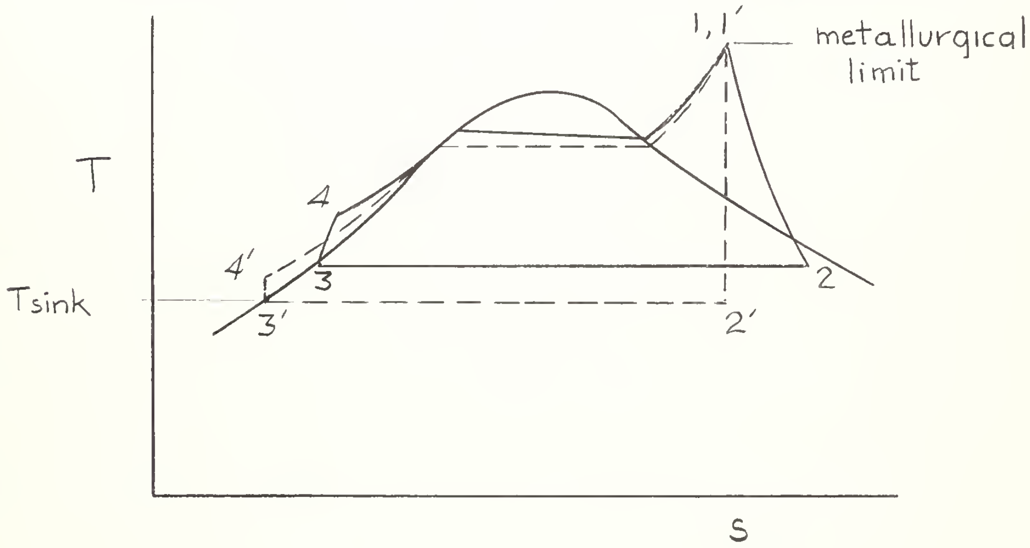
$$\eta_{th} = \frac{(h_1' - h_4') - (h_2' - h_3')}{(h_1' - h_4')} \qquad (2)$$

where $(h_1' - h_4')$ is the heat supplied and $(h_2' - h_3')$ the heat rejected per pound of working fluid. This efficiency is maximized when T_1 is as high and T_3 as low as is physically possible. The upper limit of T_1 is set by the metallurgy of the heat receiving device. T_3 in the ideal case is the sink temperature. The Rankine cycle of Figure I differs from a real one in several ways. These are illustrated in Figure II, where the ideal cycle of Figure I is shown in dotted lines and a

realizable case is depicted by a solid line. For comparative purposes assume that states 1′ and 1 entering the turbine

Figure II

An Actual Steam Cycle Compared with Rankine Cycle



are identical. In the actual case the turbine is irreversible and the entropy of the working fluid increases. State 2 is at a higher temperature than state 2′ because an infinitely large condenser would be required to cool the working fluid to sink temperature. Some subcooling of condensate occurs in the actual case to point 3. Since the pump is not reversible, the condensate increases in entropy in the pumping process. State 4 must be at a higher pressure than 4′ to compensate for pressure losses which occur in the boiler and piping. Every way in which the actual cycle varies from the ideal Rankine cycle causes a further reduction in thermal efficiency. We show next how the failure of real machines to conform to ideal processes can be evaluated and related to the physical characteristics of the

machine. To do this we now examine the concept of available energy.

Availability is defined by Keenan [9] as "the maximum work which can result from interaction of system and medium when only cyclic changes occur in external things except for the rise of a weight." We define the system in the marine propulsion plant as the entire working fluid and the medium as the water of the sea which is stipulated an infinite reservoir of uniform and constant temperature. By arguments which need not be repeated here, it is proved that the decrease in availability per unit mass of fluid between section 1 and section 2 along the path of steady flow is

$$\left(b_1 + Z_1 + \frac{V_1^2}{2g}\right) - \left(b_2 + Z_2 + \frac{V_2^2}{2g}\right) \qquad (3)$$

where $b = h - T_o S$ and T_o is the absolute temperature of the medium. Strictly speaking, the amount by which this decrease exceeds the work delivered to things outside the steady flow system is a measure of the irreversibility of an adiabatic process between 1 and 2. For the purposes of our analysis we assume that changes in Z and $\frac{V^2}{2g}$ can generally be ignored. We further assume that available energy delivered to evaporators and turbogenerators constitutes a loss in availability in that it is parasitic of propellor shaft work. Work delivered by such auxiliaries should still be considered separately from the irreversibilities associated with the machine. Turbogenerator rating is an available energy load on the cycle over which the designer has no control. He may, however, find it profitable

to install a heavier machine in order to reduce availability lost through turbogenerator irreversibilities.

The net decrease in b which occurs when a unit mass of working fluid passes through any component of the plant thus defines the loss in energy which is available to drive the ship, with the exception of the main engines, where

$$\Delta b_t = b_i - b_e - Wk_t \quad (4)$$

Subscripts are defined as follows:

i = inlet ,

e = exhaust ,

t = turbine , and

Δb is "the loss in availability,"

Wk = shaft work .

In the boiler the working fluid undergoes an increase in availability, Δb_A. If this availability could be completely converted to work in the cycle, the thermal efficiency would be,

$$\eta_A = \frac{G_b \Delta b_A}{Q_A} = \frac{\Delta b_A}{\Delta h_A} , \quad (5)$$

where η_A = the thermal efficiency of the heat added [10] (maximum possible thermal efficiency),

Δb_A = increase in availability of steam due to heat added,

Δh_A = increase in enthalpy of steam (assumed at constant pressure),

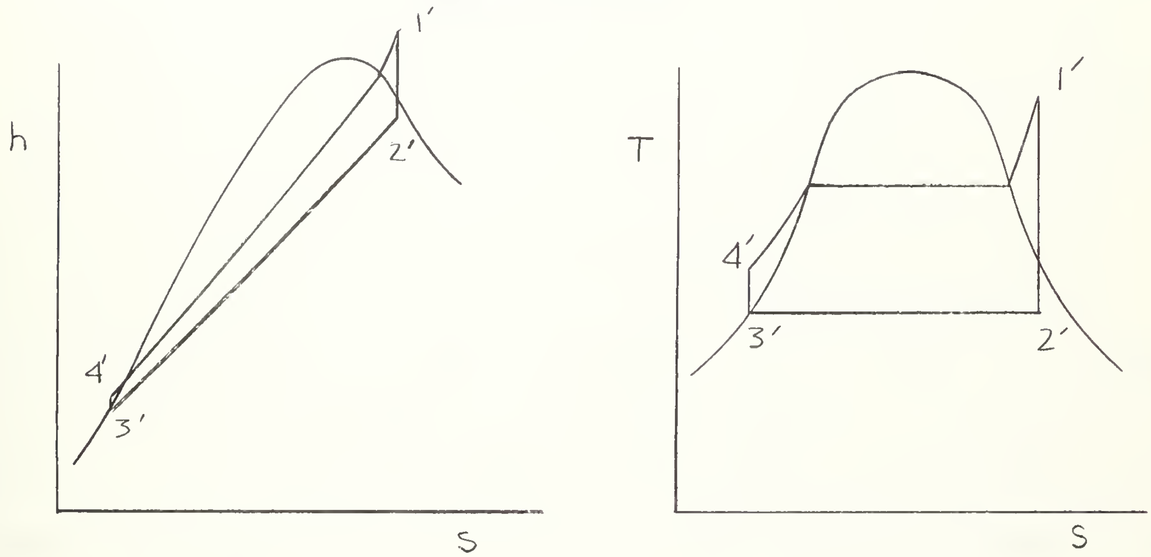
Q_A = heat added, and

G_b = boiler mass rate of flow.

The "heat added thermal efficiency" is the ideal efficiency of any cycle. We demonstrate for the Rankine Cycle, with the help of Figure III.

<u>Figure III</u>

<u>Rankine Cycle</u>



Using formula (5),

$$\eta_{th} = \eta_A = \frac{b_1' - b_4'}{h_1' - h_4'} ,$$

$$\eta_{th} = \frac{(h_1' - T_o s_1') - (h_4' - T_o s_4')}{h_1' - h_4'}$$

$$\eta_{th} = \frac{h_1' - h_2' - P}{h_1' - h_3' - P}$$

since $s_4' = s_3'$ and $s_1' = s_2'$,

$$T_o (s_2' - s_3') = h_2' - h_3' ,$$

and $P = h_4' - h_3'$ = pump work.

It should be pointed out that the "heat added thermal efficiency," η_A, does not give an indication of the efficiency of the

boiler in recovering the available energy in the fuel. First, the temperature of the combustion products may be assumed to be about 3500°F, while the maximum metal temperature is limited to about 1500°F. Second, most of the heat is added at a temperature far below that permitted by the metallurgical limit, for instance, the saturation temperature of 1200 lb. steam is 567°F. Third, there is a pressure drop in the flow through the boiler which represents a loss in availability. Fortunately, for comparative purposes, the efficiency of naval boilers does not fluctuate significantly with changes in cycle parameters. We use throughout a boiler heat efficiency, η_B, at cruising, of 88%, which has proven to be within 1% of trial efficiency in a majority of tests by the Naval Boiler and Turbine Laboratory [11] over a wide range of temperatures and pressures.

The thermal efficiency of the naval steam plant <u>based on the heat added</u> is then,

$$\eta_{th} = \frac{Wk_t}{Q_A} = \frac{G_t \Delta h_t}{G_b \Delta h_A}$$

$$= \frac{\cancel{G_B} \Delta h_A}{\cancel{G_B} \Delta h_A} \quad \frac{G_t \Delta h_t}{G_b \Delta h_A}$$

$$\eta_{th} = \eta_A \quad \frac{G_t \Delta h_t}{G_b \Delta h_A} , \qquad (6)$$

where G is mass rate of flow, and subscripts are:

t = turbine,

b = boiler.

If the state of the feedwater is specified, as, for example, constant deaerating feed heater exit temperature, and if turbine

top temperature and pressure are known, then η_A is a function of the properties of these known states. $G_t \Delta h_t$ is also known since the desired SHP is specified.

$$G_t \Delta h_t = \frac{SHP}{e_m} ;$$

e_m is the mechanical efficiency of the turbine and reduction gears.

In the formula for thermal efficiency (6), only the total availability added in the boiler $G_b \Delta b_A$ is unknown. If we make use of the identity, Net availability gain in boiler = Net availability lost in all other cycle components, we can rewrite (6) as

$$\eta_{th} = \frac{\eta_A}{1 + \frac{e_m}{SHP} \sum_{j=1}^{h} G_j \Delta b_j} \tag{7}$$

where $G_j \Delta b_j$ is the availability loss (including irreversibilities, heat rejected, and non-propellor work) attributed to the j^{th} plant component. Weight of fuel for a given endurance in miles is then

$$W_F = \frac{R}{V_K} \frac{SHP}{e_m \eta_{th} \eta_B} \frac{2545}{H.H.V.} , \tag{8}$$

where R is range in nautical miles,

V_K is cruising speed, knots,

H.H.V. is the higher heating value at constant pressure,

and η_B is boiler heat efficiency.

Formula (8) can be written,

$$W_F = \frac{K_F}{\eta_{th}} = \frac{K_F}{\eta_A} \left(1 + \frac{e_m}{SHP} \sum_{j=1}^{h} G_j \Delta b_j \right) .$$

It is readily apparent that the weight of fuel carried which is directly attributable to any particular availability loss is,

$$WF_j = \frac{K_F}{\eta_A} G_j \Delta b_j \; . \tag{9}$$

In the weight optimization problem the respective weights of fuel and machinery are of comparable magnitude. The expression for thermal efficiency, (7), need not rely upon an assumed boiler flow but can be computed to the desired accuracy by computing as many of the component availability losses as are deemed significant. The relative importance of the irreversibility associated with each component can be seen from an availability balance, which can be made up using the heat balance for a given power plant. Using a heat balance for a modern destroyer leader, DLG-6, an availability balance was prepared (Table I). This table reveals the relative effect of the various components on efficiency, and consequently, fuel weight. Availability considerations for those components which consume a significant portion of available energy are the subject of the following sections.

Table I.

Availability Balance for DLG-6 at Cruising Condition*

Availability gain:	BTU	Percentage
Main feed pump	4.0	.72
Boiler: a) Main steam	472.2	84.56
b) Desuperheated steam	82.2	14.72
	558.4	100.00
Disposal of available energy:		
Main engine shaft work	288.1	51.59
Electrical output of T.G.'s	22.6	4.05
Irreversibilities		
Main feed pump	18.3	3.28
Forced draft blowers, F.O. service pumps and L.O. service pumps	18.8	3.37
Evaporators	11.1	1.99
Galley, laundry, hot water	7.8	1.40
Fuel oil heaters	.3	.05
Deaerator	8.5	1.52
Air ejectors	7.7	1.38
Turbogenerators	22.6	4.05
Main engines	116.1	20.79
Condenser	13.2	2.36
External Desuperheater	.1	.02
Piping, throttling and leakage	21.3	3.81
Exhaust hood and leaving loss	1.9	.34
	558.4	100.00

*Based on one pound of boiler feed water.

A. Boiler

The efficiency of the boiler in converting the heat in the fuel to energy available to the cycle for doing work is

$$\eta_b = \frac{\text{Energy available to cycle}}{\text{Heat in the fuel}} \tag{10a}$$

The heat efficiency of the boiler is customarily defined as

$$\eta_B = \frac{\text{Heat absorbed by steam generated}}{\text{Heat in the fuel}} \tag{10b}$$

Heat input includes heat in the fuel (H.H.V. = 18,500 BTU/lb.) plus

a) heat added by heating of fuel oil,

b) heat added by heating of combustion air, and

c) increase in heat value at constant pressure over that at constant volume (25 BTU/lb.).

Combining (10b) with (5) we may rewrite (10a) as:

$$\eta_b = \eta_B \eta_A \tag{10c}$$

For the comparative purposes of this paper, we have neglected the effects listed under a) and b) above, since they are small. η_B is taken as a constant 88%. Investigation of the cycle effect of boiler improvements, such as steam air heating operating on the exhaust from forced-draft blowers or bleed steam [13], is a suitable task for a separate availability balance on the boiler. A sample boiler availability balance is given in reference [14].

In the computation of η_A, recognition must be made of the fact that η_A for desuperheated steam is less than that for superheated steam. $\eta_{A(SH)}$ and $\eta_{A(DSH)}$ are set by choice of top steam conditions, and the assumption of constant feed temperature.

The relative flows of superheated and desuperheated steam change with variation of turbine flow, G_t, so that η_A must be computed separately for each case;

$$\eta_A(p_1, T_1, G_t) = \frac{(G\Delta b_A)_{SH} + (G\Delta b_A)_{DSH}}{(G\Delta h_A)_{SH} + (G\Delta h_A)_{DSH}} \qquad (11)$$

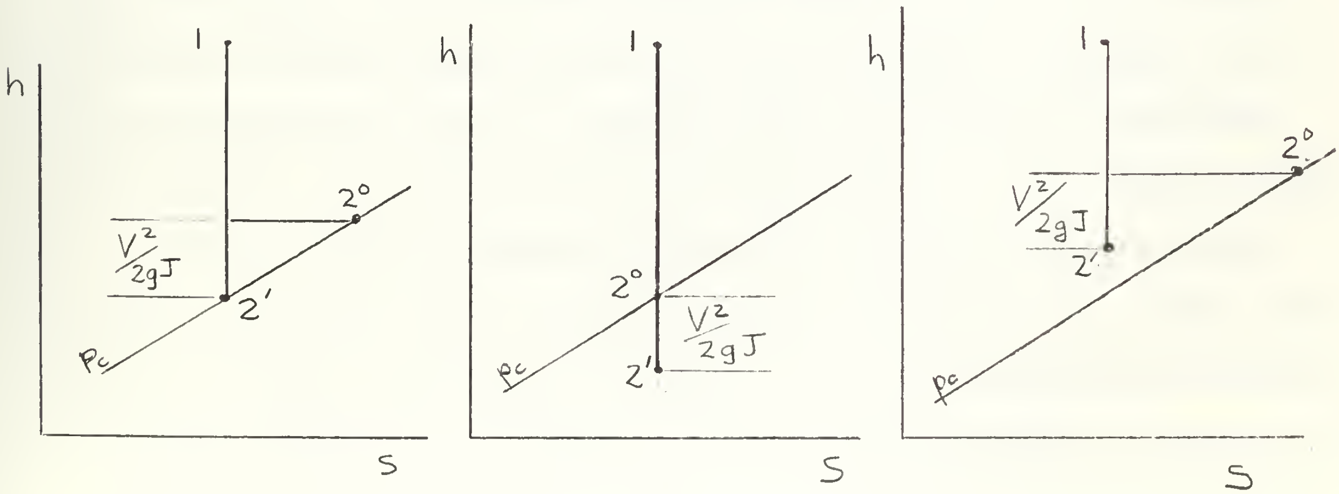
B. Main Engines and Condenser

These two major components are considered together because of their close inter-relationships. The characteristics of both components are dependent on leaving loss and the manner in which it occurs. We discuss leaving loss first, since it is a process which is often not clearly understood.

Consider an isentropic turbine expansion process from state 1 as shown in Figure IV. The condenser pressure is p_c. The steam leaving the exhaust annulus has a certain velocity V_2'. If we assume that the process $2' - 2^0$ connecting the LP turbine exhaust annulus and the condenser flange is adiabatic,

Figure IV

The Exhaust Hood Process



(a) constant pressure (b) reversible (c) Inefficient

then we can evaluate its irreversibility as:

$$I_{2'-2^o} = \left(h_{2'} - T_o s_{2'} + \frac{V_{2'}^2}{2_g J}\right) - (h_{2^o} - T_o s_{2^o}).$$

In this case $V_{2'}$ is not small enough to be neglected. Then, since

$$h_{2^o} - h_{2'} = \frac{V_{2'}^2}{2_g J} ,$$

$$I_{2'-2^o} = T_o(s_{2^o} - s_{2'}). \qquad (12)$$

In a reversible, diffusing exhaust hood, the process is the isentrope $2' - 2^o$ shown in Figure IV(b). In a very poorly designed exhaust hood, friction causes an increase in entropy, $ds > \frac{dQ}{T}$, so great that a pressure loss occurs, process $2' - 2^o$, Figure IV(c).

As might be expected, the cycle effects of the exhaust hood irreversibility all tend toward reduced efficiency and increased component weight. The change in entropy in process $2' - 2^o$ is seen to increase as the pressure change $2' - 2^o$ drops from the isentropic increase of Figure IV(b) to the decreasing tendency of the highly irreversible process Figure IV(c). In reality, the best exhaust hoods which are built today can operate with no net pressure loss at cruising speed and may be considered to follow a constant pressure process line $2' - 2^o$, Figure IV(a).

Privileged information from one leading turbine manufacturer indicates that a slight pressure increase is in fact possible, using a specially designed hood. Pressure loss between turbine annulus and condenser flange is diagrammed in Figure V, where the improved hood is compared with a poorly designed one

Figure V

Exhaust Hood Pressure Loss

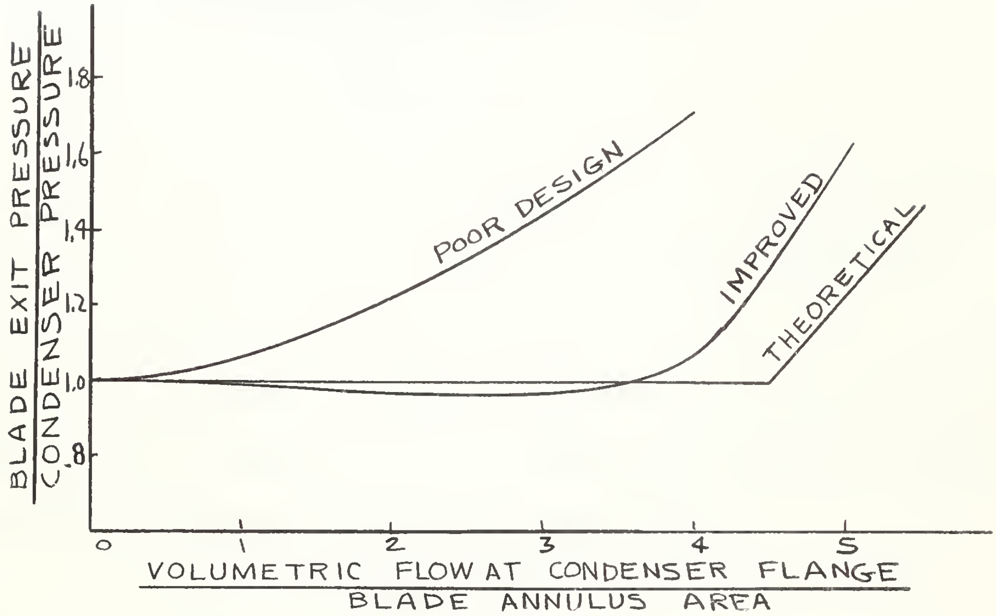
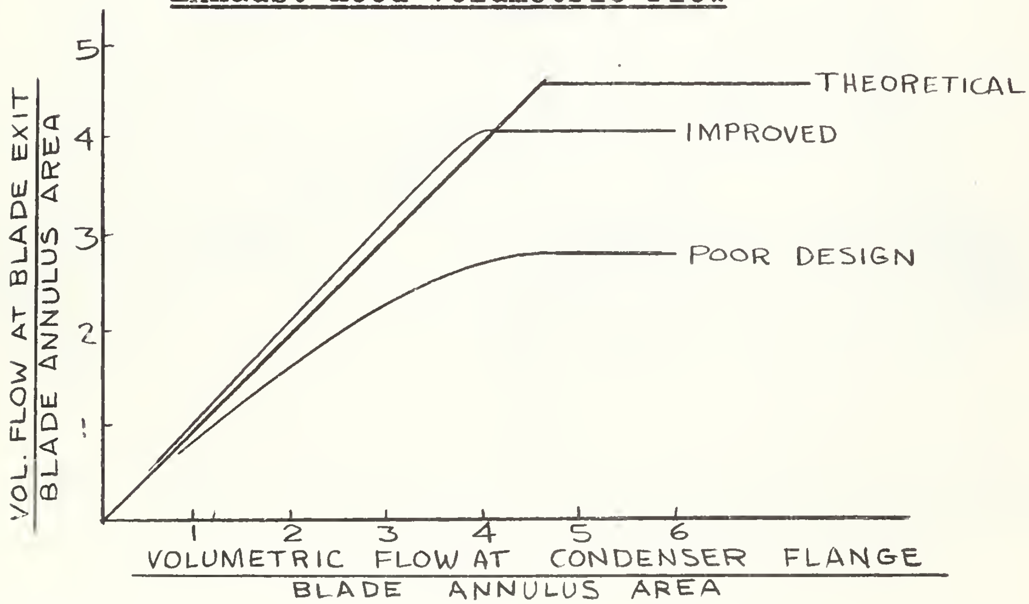


Figure VI

Exhaust Hood Volumetric Flow



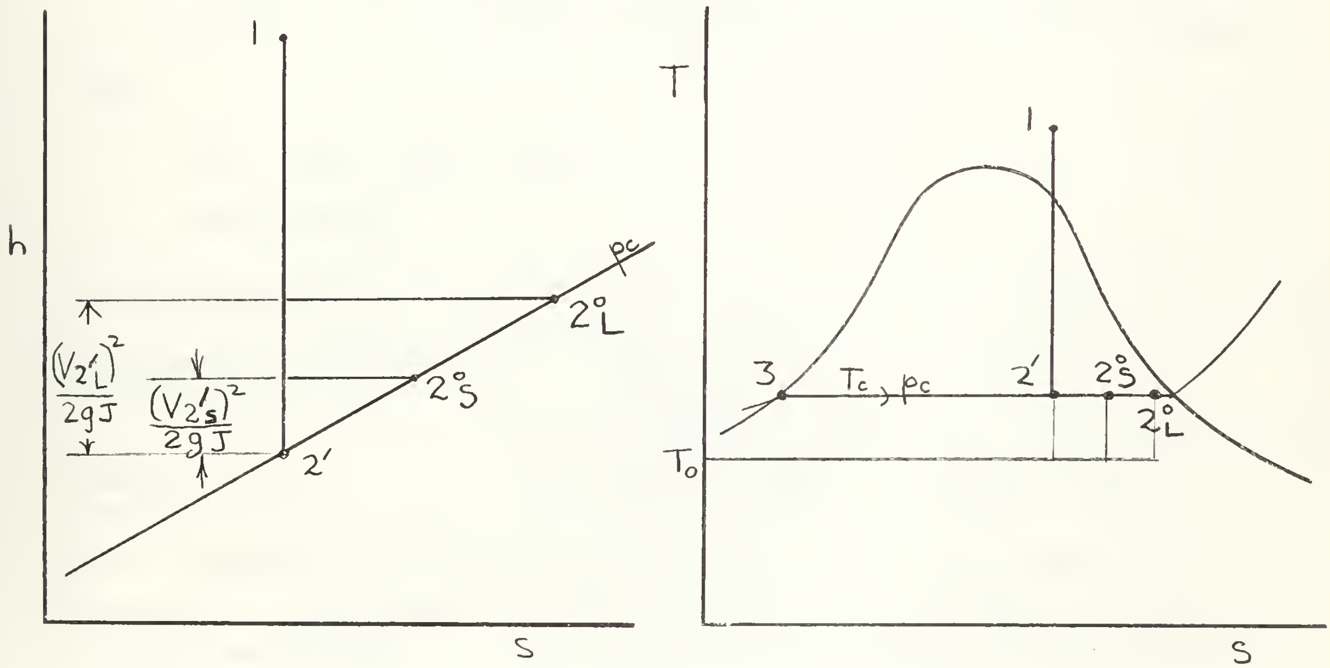
and an ideal one with no loss in available energy.

Volumetric flow for the same cases is shown in Figure VI, where the limitation of sonic blade exit velocity is seen. We have assumed, for the sake of uniformity, that the leaving velocity loss is recovered in enthalpy gain as a constant pressure process. This may not be fully realizable in naval turbines if we permit the presence of reversing stages within the "specially designed" exhaust hood.

We are now ready to consider the availability implications of leaving loss. Assume again an isentropic turbine expansion with a fixed condenser pressure, Figure VII.

Figure VII

Large Versus Small Leaving Loss



$$P_2^{o} = P_2'$$

$$\frac{V_2'^{2}}{2gJ} = h_2^{o} - h_2' \quad , \quad h_2^{o} = h_2' + \frac{V_2'^{2}}{2gJ}$$

We compare a large leaving velocity $V_{2'L}$ and a small one, $V_{2'S}$ which result in stagnation enthalpies $h_{2}o_{L}$ and $h_{2}o_{S}$ respectively.

Now $b_1 - b_3$ is constant.

$$Wk_t = (h_1 - h_2') - (h_2o - h_2') .$$

Turbine work is decreased by the amount of leaving loss.

The irreversibilities of the exhaust hoods are, by (12),

$$I_{hood} = T_o (s_2o - s_2') .$$

$$s_2o - s_2' = \frac{h_2o - h_2'}{T_c} = \frac{V_{2'}^2}{2gJT_c} .$$

$$I_{hood} = \frac{V_{2'}^2}{2gJ} \frac{T_o}{T_c} .$$

Finally, the available energy rejected by the condenser is:

$$(T_c - T_o) (s_2o - s_3) .$$

Tabulating,

Leaving velocity	$V_{2'S}$	$V_{2'L}$
Decrease in available energy, $b_1 - b_3$	$(h_1 - h_3) - T_o(s_1 - s_3)$	$(h_1 - h_3) - T_o(s_1 - s_3)$
Turbine work	$h_1 - h_2o_S$	$h_1 - h_2o_L$
Hood irreversibility	$T_o(s_2o_S - s_2')$	$T_o(s_2o_L - s_2')$
Condenser available energy rejection	$(T_c - T_o)(s_2o_S - s_3)$	$(T_c - T_o)(s_2o_L - s_3)$

We see that the decrease in turbine work due to leaving loss equals the sum of the increases in hood irreversibility and condenser availability rejection.

The effect of a leaving loss on an isentropic turbine can be summarized as follows:

$$\text{Leaving loss} = \frac{V_2'^2}{2gJ}$$

$$\text{Decrease in turbine work} = \frac{V_2'^2}{2gJ}$$

Hood irreversibility,

$$I_{hood} = \frac{V_2'^2}{2gJ} \frac{T_o}{T_c} .$$

Increase in available energy rejected by condenser,

$$\triangle I_c = \frac{T_c - T_o}{T_c} \quad \frac{V_2'^2}{2gJ}$$

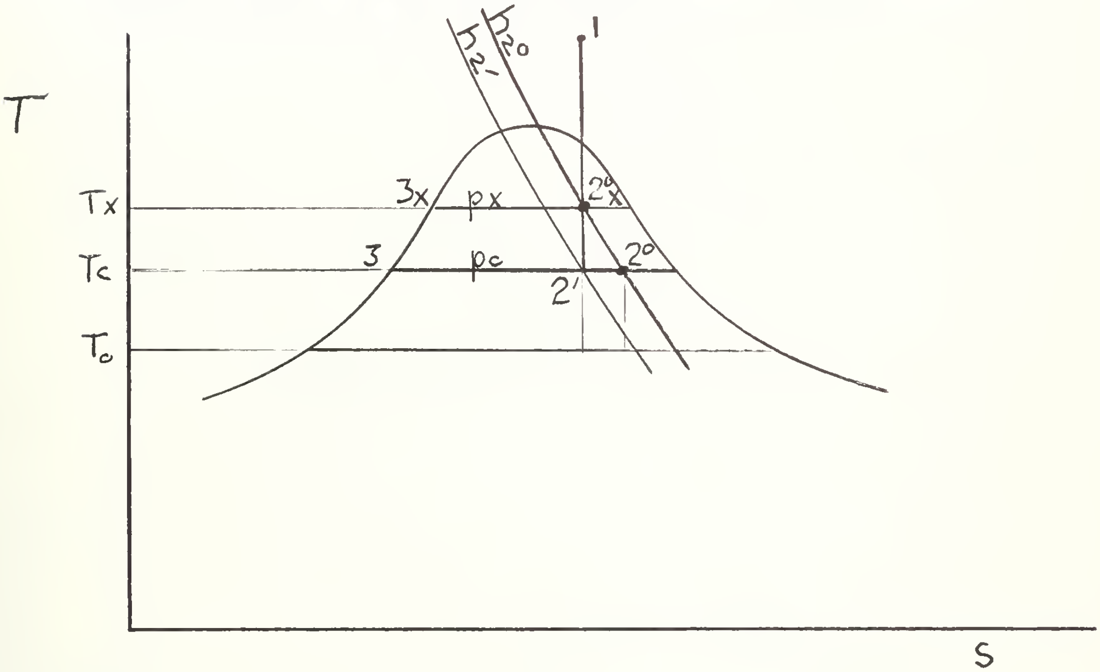
These results are for the case where $p_2' = p_2^o$. It must be remembered that the unfavorable consequences of leaving loss are, in this ideal turbine case, a function of the imperfect behavior of the exhaust hood. There would be no deleterious effect due to leaving velocity if all components were reversible.

For the analysis of this paper, we have chosen to phrase the availability balance of the turbine-condenser combination in another way. We assume that top conditions are fixed, as are steam flow, condition line, and stagnation enthalpy h_2o (see Figure VIII). The consequence of this is that as leaving loss ($h_2o - h_2'$) increases the condenser pressure must be decreased. A maximum on leaving loss is imposed when T_c must be depressed to T_o and the condenser becomes infinitely large. An availability balance in this case yields the following results:

Decrease in turbine work due to leaving loss = 0.

Figure VIII

Leaving Loss with Variable Condenser Pressure



$$I_{hood} = T_o(s_{2^o} - s_{2'}) = \frac{T_o}{T_c} \quad \frac{V_{2'}^2}{2gJ} \; .$$

Increase (due to leaving loss) in available energy rejected by the condenser $= \left(\frac{T_c - T_o}{T_c}\right) \quad \frac{V_{2'}^2}{2gJ}$

$$- (T_x - T_o)(s_{2^o_x} - s_{3x}) + (T_c - T_o)(s_{2^o} - s_3) \; ,$$

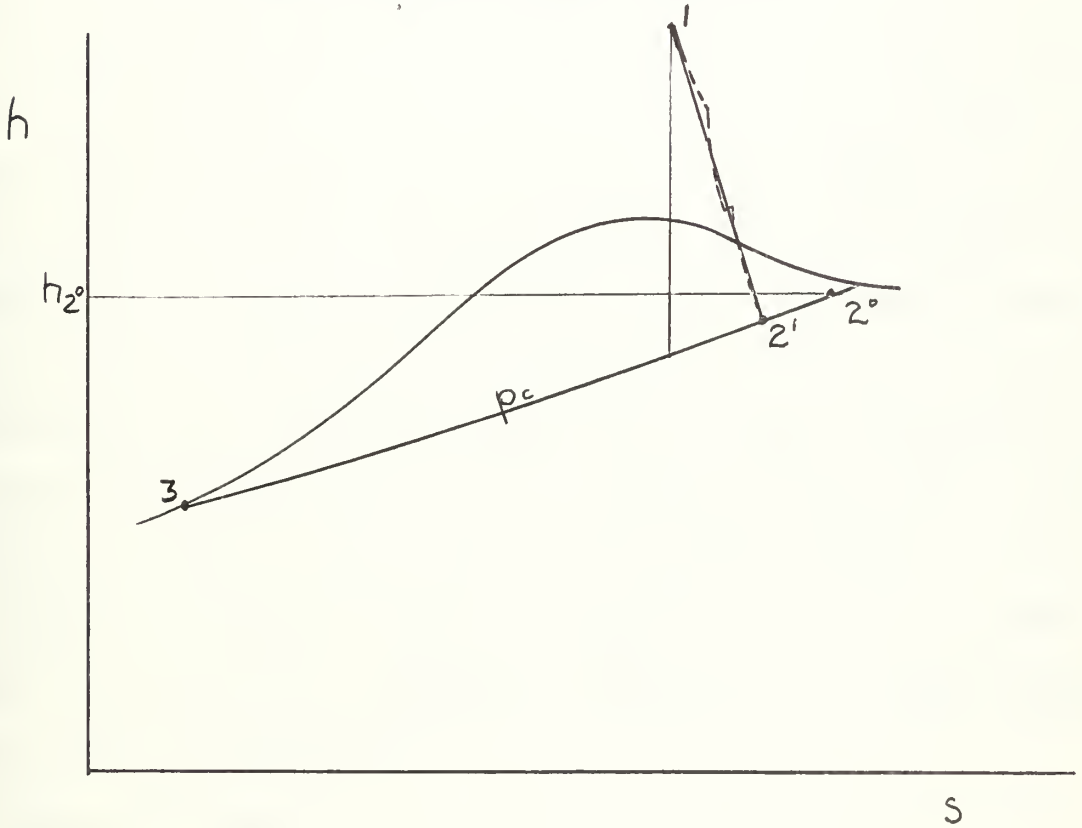
where x indicates the condenser pressure corresponding to a fictitious zero-leaving-loss turbine. Summing the availability losses due to leaving loss, it is found that for any leaving loss, the irreversibility in the turbine and condenser combination is approximately independent of magnitude of leaving loss, to within the accuracy of the Mollier chart.

In computing the variations in turbine and condenser weights with leaving loss, we have assumed that the condition

line can be extended as a straight line through turbine inlet and exhaust annulus state points. This is illustrated in Figure IX, where the actual condition line might be as indicated by the dotted line. For a particular steam flow through the turbine, and with fixed turbine inlet conditions, h_2o is fixed by the requirement of certain SHP.

Figure IX

Main Engine Condition Line



The irreversibility Δb_{13} can be treated as a constant as proved above. We have computed Δb_{13} uniformly for a leaving loss of 4 BTU/lb. Considering Δb_{13} a constant for a particular T_1, p_1, G_T, we can then rewrite equation (7) as:

$$\eta_{th_{H.A.}}(T_1, p_1, G_t) = \frac{\eta_A}{1 + \frac{e_m}{SHP} G_t \Delta b_{1-3} + \frac{e_m}{SHP} \sum_{j=1}^{n-2} G_j \Delta b_j} ,$$

$$\eta_{th_{H.A.}}(T_1, p_1, G_t) = \frac{\eta_A}{\text{const.} + \frac{e_m}{SHP} \sum_{j=1}^{n-2} G_j \Delta b_j} .$$

While the loss in availability computed in this manner may be taken independent of leaving loss, the combined weight of turbine and condenser varies over a range in which is found a minimum, which lies between the high turbine weight characteristic of low leaving loss and the high condenser weight of high leaving loss and high vacuum. These weights are computed for specific values of leaving loss, using the weight formulas developed in Appendix A, and state points picked off the condition line laid down as in Figure IX. Finally, it must be observed that condensate temperature is not independent of leaving loss, and that as a consequence of this fact, additional losses in availability are necessitated elsewhere in the cycle. A numerical example, demonstrating that the total weight change due to elevation of condensate temperature is indeed negligible, is given in Appendix C.

C. Main Feed Pump.

The main feed pump is the only pump in the feedwater system which has a significant net availability loss. The loss in availability in the feed pump engines exceeds the gain in the pump. The amount by which loss exceeds gain is the net irreversibility, which is computed as follows:

$$Wk(pump) = v \Delta p ,$$

$$\text{Available energy (turb)} = \frac{1}{\eta_p \eta_{pt}} v \Delta p$$

$$\Delta b_{mfp} = v \Delta p \left[\frac{1}{\eta_p \eta_{pt}} - 1 \right] ,$$

or

$$I_{mfp} = G_{pt} (b_e - b_i)_{pt} - G_p (b_e - b_i)_p$$

Assumptions:

a) $\eta_p = 0.50 =$ constant

b) No main feed booster pump

c) p_{mfp} exit $= p_o$

d) Pump turbine condition lines parallel to those of DLG-6.

The assumption of constant η_p might bear improving upon, over the range of pressures considered. The amount of pump work in the cycle has a greater effect on cycle efficiency as pump efficiency decreases. Main feed pumps, being inherently inefficient, affect the cycle efficiency significantly, therefore it is important that they be evaluated as accurately as possible.

D. Feed Mixing

Important irreversibilities due to feed mixing occur primarily in the deaerating feed heater in naval plants. Such losses in available energy are easy to compute. Once a heat balance has been made on the mixing point, properties of entering and leaving flows are known and,

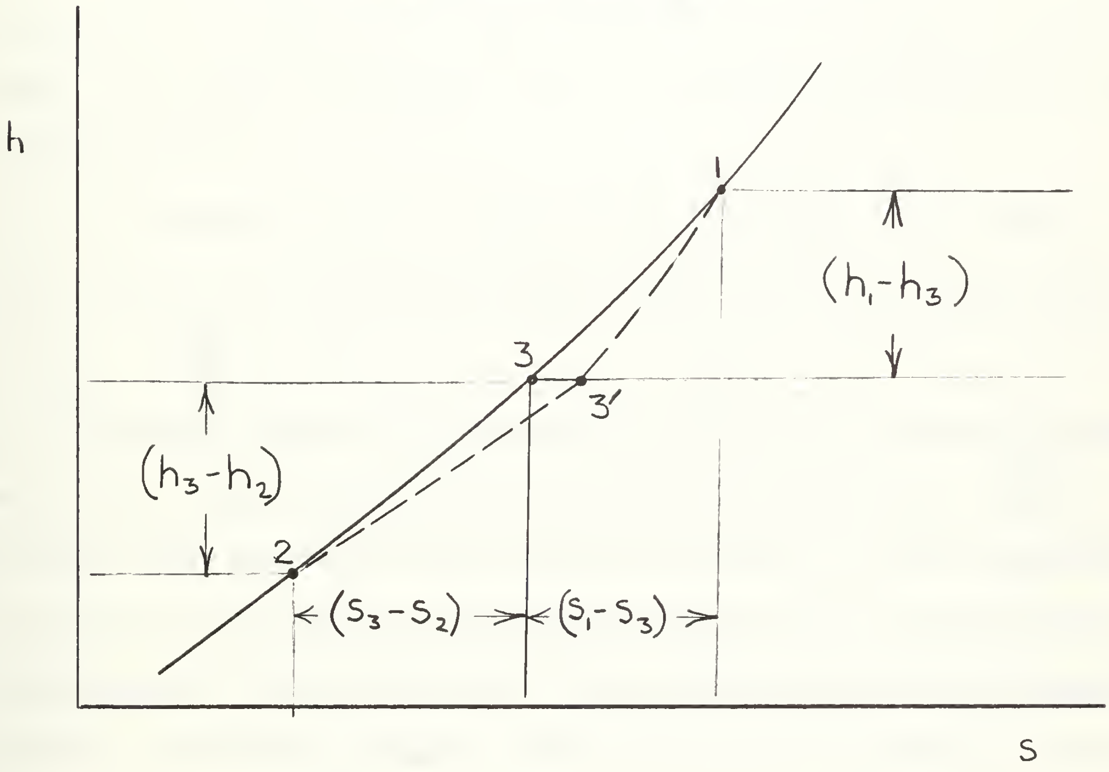
$$I = T_o \sum_{e=1}^{n} G_e s_e - T_o \sum_{i=1}^{m} G_i s_i , \qquad (13)$$

where e = 1, 2...n are the flows leaving the mixing point, and i = 1, 2...m are the flows entering the mixing point.

Formula (13) can be visualized from an example process diagrammed in Figure X. Two fluid flows of equal magnitude, at states 1 and 2, respectively, are mixed. The resultant

Figure X

A Simple Feed Mixing Process



flow is a single stream at state 3. Writing the availability change equations for the mixing process we have:

$$(h_1 - T_o s_1) - (h_3 - T_o s_3) = \text{Decrease } 1\text{-}3$$

$$(h_3 - T_o s_3) - (h_2 - T_o s_2) = \text{Increase } 2\text{-}3$$

Subtracting;

$$(h_1 - T_o s_1) - 2(h_3 - T_o s_3) + (h_2 - T_o s_2)$$

$$= \text{net availability loss,}$$

but $(h_1 - h_3) = (h_3 - h_2)$ by heat balance.

Irreversibility $= 2T_o s_3 - T_o s_1 - T_o s_2$.

Because $\frac{d^2 h}{dS^2} > 0$, $(s_1 - s_3) < (s_3 - s_2)$.

$$\frac{\text{Net loss}}{\text{lb. mixed fluid}} = T_o \left[(s_3 - s_2) - (s_1 - s_3) \right] > 0.$$

If pressure loss occurs in a heat exchange device, the net irreversibility is still given by formula (13). A visualization in Figure X, where the exit state point is at $3'$, shows that the irreversibility is further increased by the pressure loss.

If we make use of the relationship $T_{ds} = dh - vdp$, for a constant pressure, single phase process, we may write,

$$\frac{dh}{ds} = T .$$

Because augmenting steam is bled to D.A. tank pressure at a very high temperature, its availability loss $\int_1^2 \left[T - T_o\right] ds$ is much greater than the availability gain of the condensate. It is for this reason that augmenting steam bled from the desuperheated steam line gives less efficient feed heating than turbine extraction steam. Extraction feed heating is not used in naval steam plants due to operating complications and the extra weight of the feed heaters, piping and valves involved.

E. <u>Pressure or Throttling Loss</u>.

The loss in available energy due to pipe friction or intentional throttling processes is simply $T_o \Delta S$. This, of course, can be evaluated merely by picking the entropies off of the steam chart. An approximate formula which is useful for small

pressure drops is developed in reference [10]:

$$\Delta b_{12} \cong T_o \frac{R}{J} \frac{\Delta p_{12}}{p} .$$

A curve of $\frac{R}{J}$ versus T is given in the same reference. In this curve, a particular fractional pressure loss in superheated steam is seen to cause a much greater availability loss than a like pressure loss in the liquid.

In our analysis, we have assumed throttling for various auxiliary purposes to the same pressure levels as in DLG-6, regardless of top pressure under consideration. The flows and irreversibilities involved were small, and the final result would not be altered by throttling to other pressure levels.

The main steam pressure loss was evaluated for DLG-6 and assumed a constant irreversibility. Bureau of Ships Design Data Sheet DDS 48-1-b considers main steam piping size a variable. This is further discussed in reference [15]. Both the Design Data Sheet and the reference base the optimization of piping size on weight of fuel consumed at full power. This is not consistent with the principle of minimum weight of fuel plus machinery to meet a particular range at cruising speed. In naval ships, where cruising power is of the order of 1/7 of rated full power, main steam piping which will pass full power steam flow at acceptable speeds*, does not contribute significantly to cruising speed irreversibility. Both the irreversibility (0.6 BTU/lb.) and the total weight of piping,

---

* Noise may become excessive if steam velocities exceed about 400 ft./sec. [15].

valves, etc., (16,000# connecting one boiler to one main engine*) are relatively small and not subject to a large practical range of variation. The authors submit that a proper main steam piping weight optimization, which is not made in DDS 48-1-b, and does not affect the optimization of this paper, should balance weight of main steam piping against additional boiler weight required to offset pressure losses at full power.

F. Other Auxiliaries

The necessity of considering minor auxiliaries in detail can be evaluated from the availability balance. If a particular auxiliary function consumes 1% of the available energy added by the boiler, $G_b \Delta b_A$, then neglecting that loss altogether will change the fuel requirement by 1%. If we assume that this 1% availability consuming auxiliary does not change more than 10%, or that we can compute its change to within that accuracy for each change in the main variables, a 0.1% fuel weight error results. Thus for a ship which carries 1,000 tons of fuel, the error in fuel weight which can be attributed to any particular auxiliary will be one ton. In order to maintain this accuracy we have dealt with auxiliaries as follows:

1. Forced draft blowers, fuel oil service pumps, and lube oil service pumps. These auxiliaries all operate on de-

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* By private communication with Mr. Eric Moberg, Piping Design Section, Boston Naval Shipyard.

superheated steam and exhaust to the D.A. tank at 15 psig. in DLG-6. Mass flow rate, exhaust enthalpy, and irreversibility were calculated as for a single machine, using the following assumptions:

a. $(G\Delta h)_{DSH\ Aux.} = G_t \left[\frac{(G\Delta h)_{DSH\ Aux.}}{G_t} \right]_{DLG-6}$

b. $\eta_{DSH\ Aux.}$ = const. = 10%

c. $p_{DSH} \cong p_{SH}$

d. Desuperheated steam has 60°F superheat.

2. Evaporators. Steam flow and irreversibility were computed for every p_1, G_t combination, with constant $[G\ \Delta h]_{evap.}$ as a basis. Auxiliary exhaust steam at 15 psig. is the input to the evaporators. Evaporator drains at 160°F are pumped to the D.A. tank.

3. Galley, laundry, and hot water. These services were considered constant heat loss items, in the same manner as the evaporators. Mass flow rate and irreversibility were computed for each p_1, assuming external desuperheating to 400°F, throttling to 50 psig. and exhaust to the drain tank at 200°F.

4. Turbogenerators. Turbogenerator irreversibilities consume 4% of the available energy added by the boiler at cruising condition in DLG-6. This loss is second only to that in the main engines. For this reason, a weight optimization of the turbogenerators can be expected to be of value to overall plant optimization. We have made the simplifying assumption that turbogenerators are constant weight - constant output machines. Condition lines were drawn parallel to DLG-6 turbo-

generator condition lines, from the various main steam conditions, to a constant vacuum of 3.06" Hg. abs. Irreversibilities involved in the mixing of main and turbogenerator condensate streams were also computed.

5. Air ejectors. Current practice in naval steam plants is to keep air ejector steam flow constant. The ejector steam flow in DLG-6 is about 150% of that indicated in Kent, [16], for condensate flow at rated full power. We have assumed that desuperheated steam is throttled to 150 psig. and used in every case in the same amount as in DLG-6. Irreversibilities attributable to air ejectors, and condensate heating are then computed in every case.

G. Leakage

The amount of working fluid lost by leakage is assumed proportional to top pressure, in our case, $G_\ell = p_1 \left(\frac{G_\ell}{p_1} \right)_{DLG-6}$.

Leakage steam is assumed to have the main engine throttle availability, thus

$$I_\ell = G_\ell \left[b(p_1) - b_f \right],$$

where subscript f indicates make-up feed water.

DETAILS OF PROCEDURE

The determination of minimum total weight of machinery plus fuel involves varying those state points which affect machinery weight or fuel economy, calculating the changes of weight and irreversibility caused by these variations, and assembling the resulting data to find the optimum point.

It is assumed that the optimum plant will be one which operates at the maximum steam temperature which is metallurgically practicable. Turbine throttle pressure, condenser pressure, and LP turbine leaving loss are the main variables considered. Best leaving velocity is picked for a number of condenser pressures, for each top pressure, on the basis of minimum turbine and condenser weight as described in section B.

For each of p x q (p top pressures, q condenser pressures) plants covering the range of the expected optimum, total weight of fuel for various cruising ranges and total variable machinery weight is computed. These weights are then summed to give total non-constant weight. Equations used to compute machinery weights are listed in Table II. Derivations of these weight equations are the subject of Appendix A.

A maximum steam temperature of 1050°F is assumed throughout the calculation.

Table II.

Machinery Weight Equations

Condenser:

$$W_{c(wet)} = \frac{k_1 \; G_c}{(T_o - k_t)} + k_2$$

Main feed pump:

$$W_p = k_3 G_b P_1 + k_4 \; G_b^{3/2} \; P_1^{1/4}$$

Deaerator:

$$W_d = k_5 G_b$$

Main engine: (From [4])

$$W_t = k_6 A_{ex} + k_7 \; A_a \left[.577 \left(\frac{P_1}{P_{1 \; Ref.}} \right) + .423 \right]$$

where

A_a = Last stage annulus area based on 310 fps. axial velocity,

A_{ex} = Exhaust annulus area.

Boiler:

$$W_{b(wet)} = \left(k_{wtr} + k_{pp} P \right) G_b + k_{w+r} \; G_b^{2/3}$$

RESULTS

The method outlined in the previous section resulted in a tabulation of combined turbine and condenser weights versus leaving loss —— six of these sets for each throttle pressure-flow combination. A sample tabulation for G_t = 73,200 lb./hr. and p_1 = 1000 psia is shown in Table III. In addition, a specific availability loss was tabulated for each throttle pressure-steam flow combination.

The availability analysis of the turbine and condenser in Procedure, part B, indicated that a best turbine-condenser combination could be selected for each throttle pressure-steam flow combination on the basis of minimum weight of turbine and condenser alone without having a significant effect on the total variable weight.

Figures XI through XV indicate that optimum or near optimum turbine-condenser combinations could be selected on this basis for each pressure and steam flow.

The best turbine-condenser combinations were then used in the calculation of the plants described in Table IV Here the main variable weights and corresponding availability losses are tabulated for each p_1, G_t combination. Condenser pressure, p_2, for each plant is listed in the bottom line. Thermal efficiencies, generated from equation (7) are also tabulated, as are fuel weights for ranges of 3000, 5000, 7000 and 10,000 miles computed from equation (8).

From Table IV main variable weights, W_v were plotted versus condenser pressure for each top pressure and range on Figures XVIII, XX, XXII and XXIV.

Best values of W_v were then plotted versus throttle pressure for each range on Figures XIX, XXI, XXIII and XXV.

Figure XXVI has plotted on it optimal curves for throttle pressure, exhaust annulus area, condenser area and condenser pressure versus range. This figure in fact is a digest of the numerical results of this investigation.

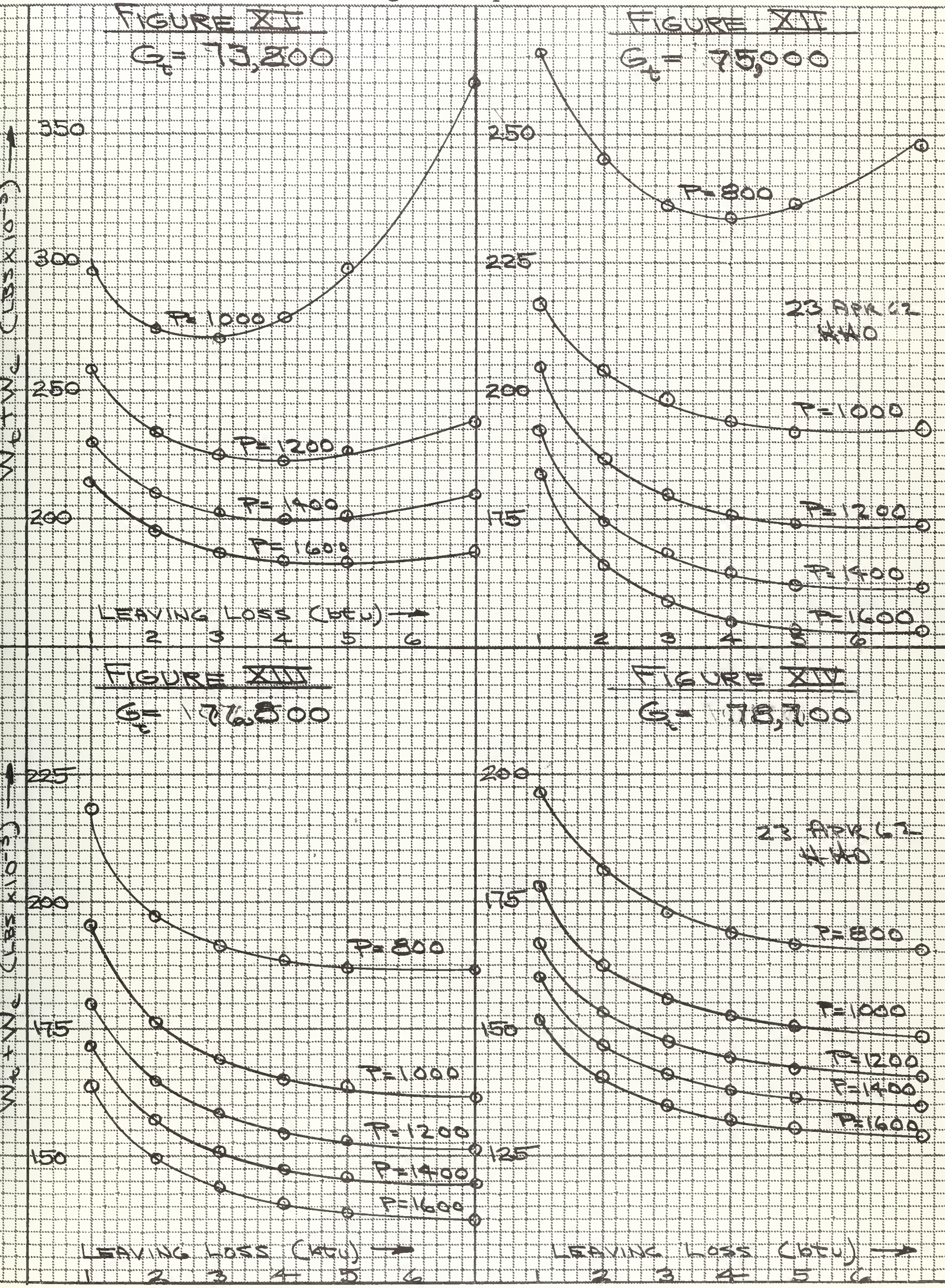
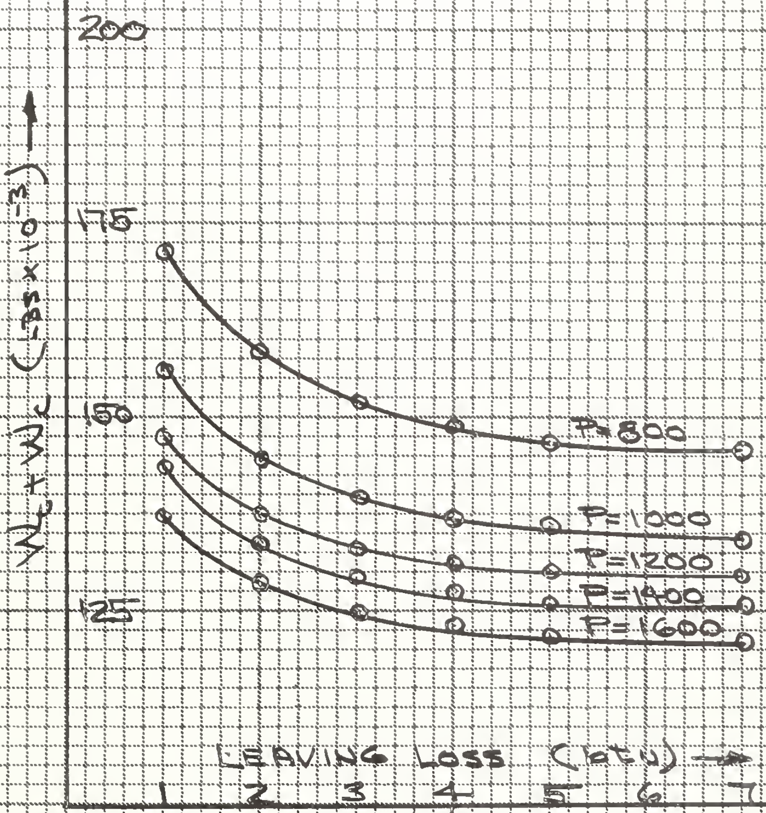


FIGURE XI
G_t = 73,200
P=1000
P=1200
P=1400
P=1600
LEAVING LOSS (btu)
FIGURE XII
G_t = 75,000
P=800
P=1000
P=1200
P=1400
P=1600
23 APR 62
FIGURE XIII
G_t = 76,800
P=800
P=1000
P=1200
P=1400
P=1600
LEAVING LOSS (btu)
FIGURE XIV
G_t = 78,700
23 APR 62
P=800
P=1000
P=1200
P=1400
P=1600
LEAVING LOSS (btu)

FIGURE XV

$G_t = 182,500$



(23 APR 62)

FIGURE XVI

FEED PUMP WEIGHT vs G_b

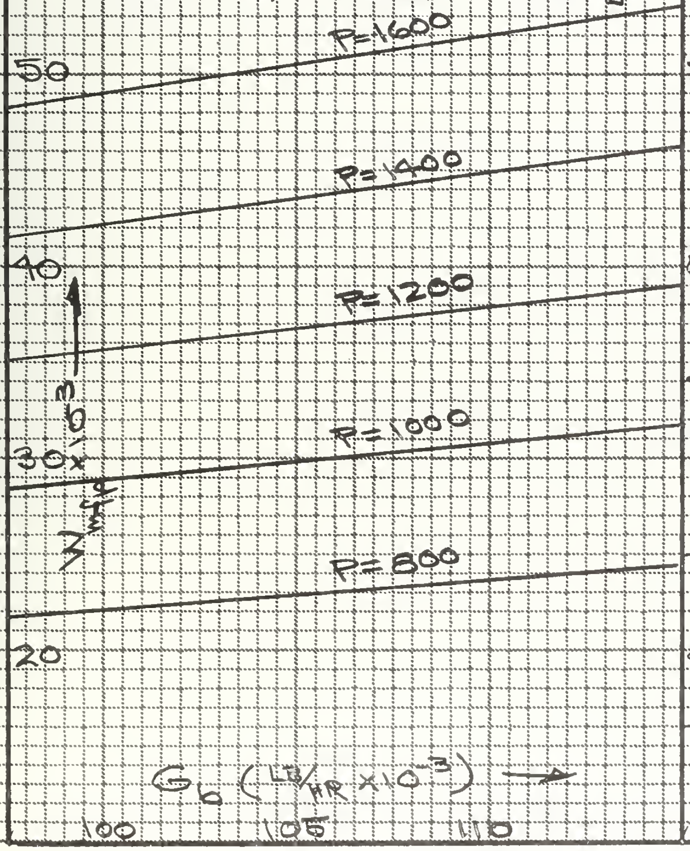


FIGURE XVII

BOILER WEIGHT vs G_b

(4 BOILERS OF 4)

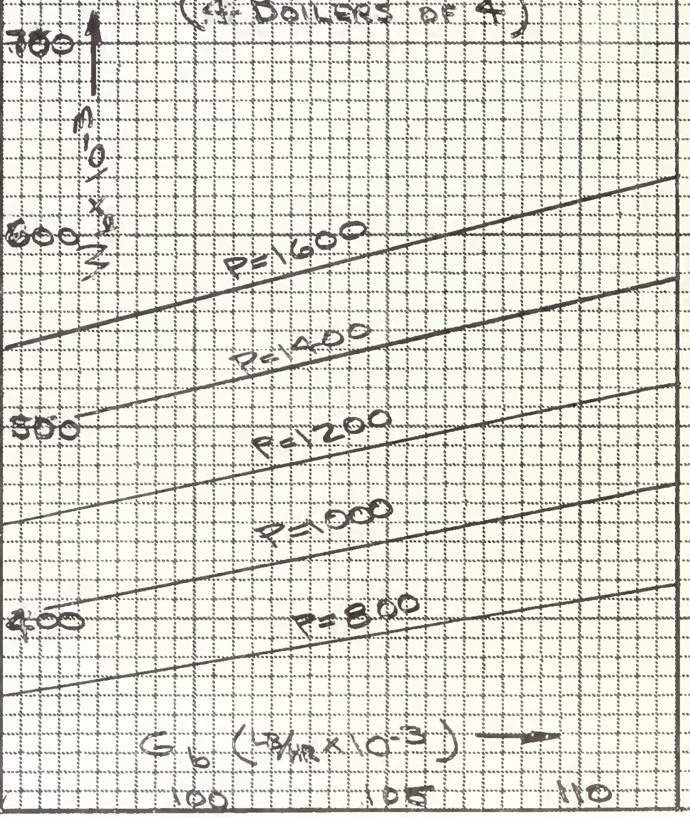


FIGURE XVIII

W_J VERSUS P_2 FOR R = 3000

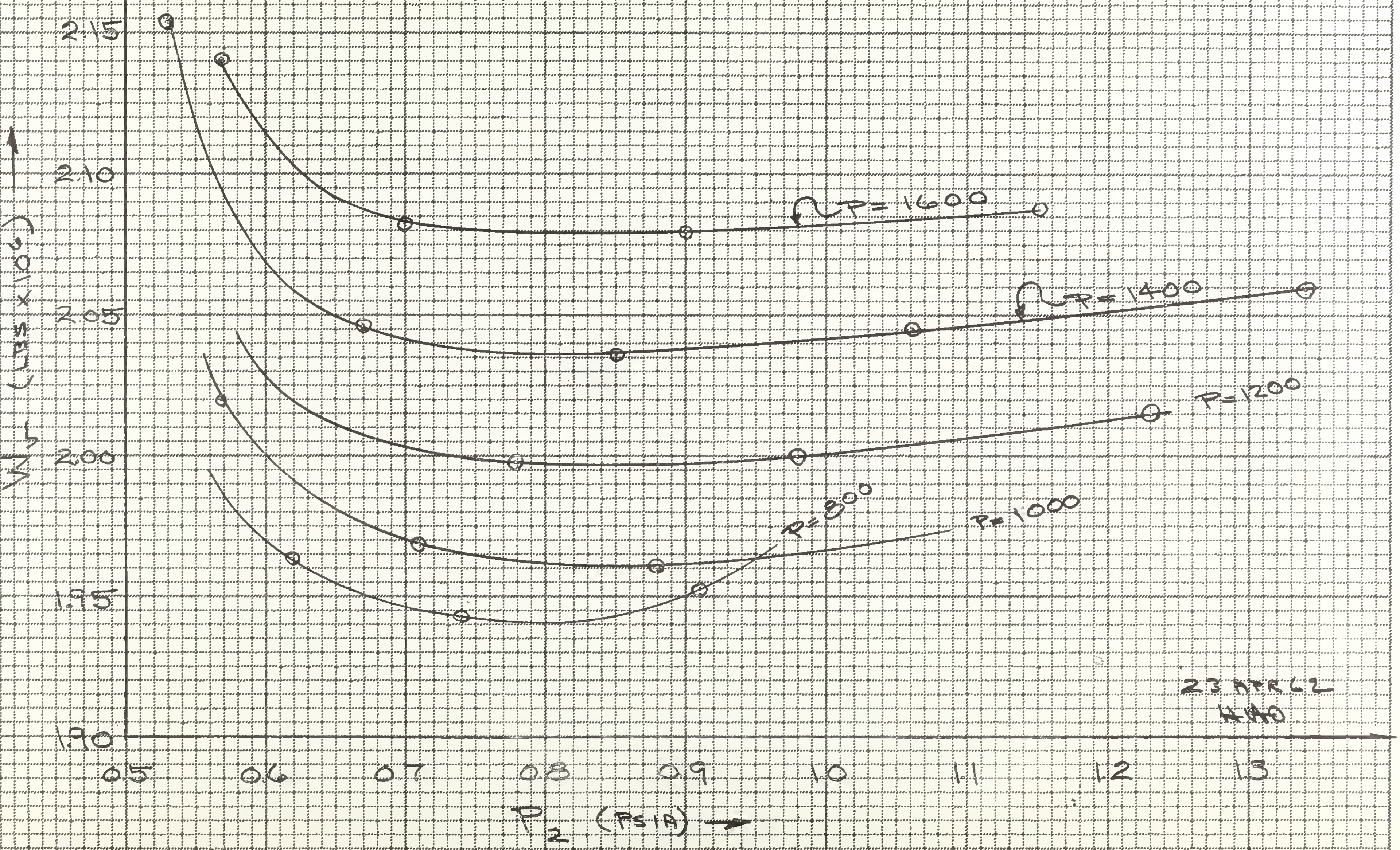
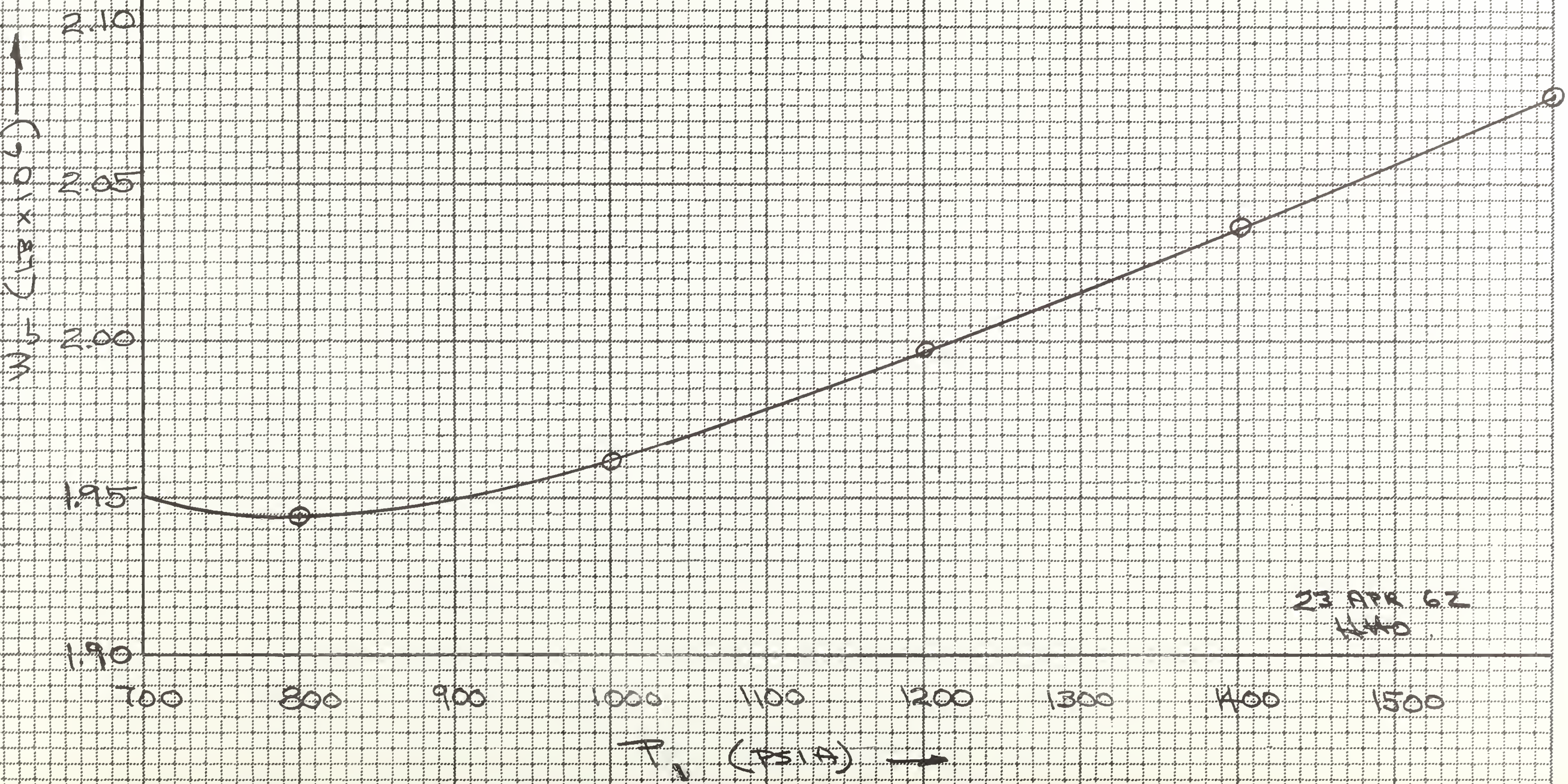


FIGURE XIX

W_s VERSUS P_b FOR $R = 3000$



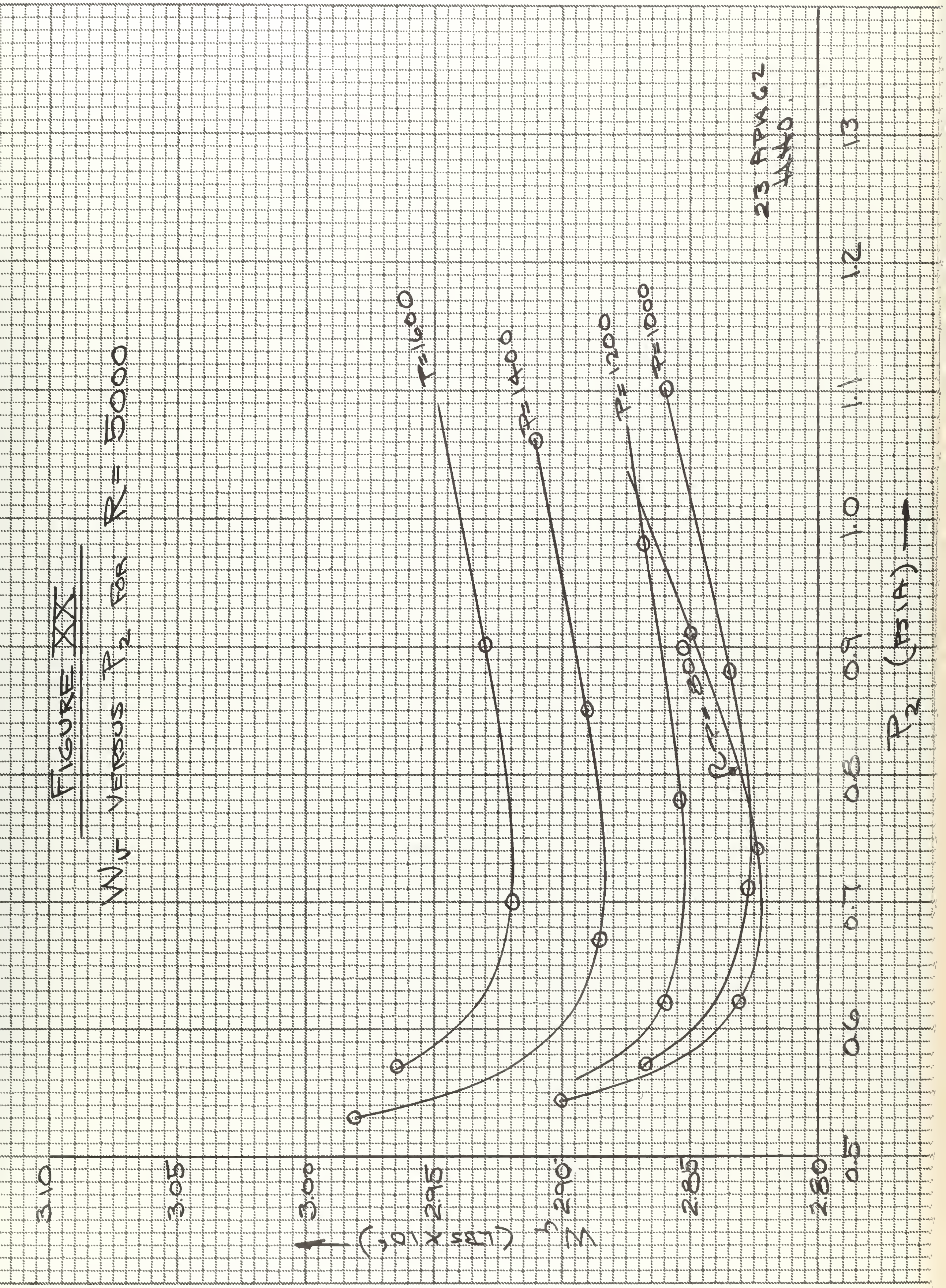


FIGURE XX
Wf VERSUS P2 FOR R = 5000
P=1600
P=1400
P=1200
P=1000
P=800
P2 (PSIA)
Wf (LBS X 10^6)
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
1.3
2.80
2.85
2.90
2.95
3.00
3.05
3.10
23 APR 62

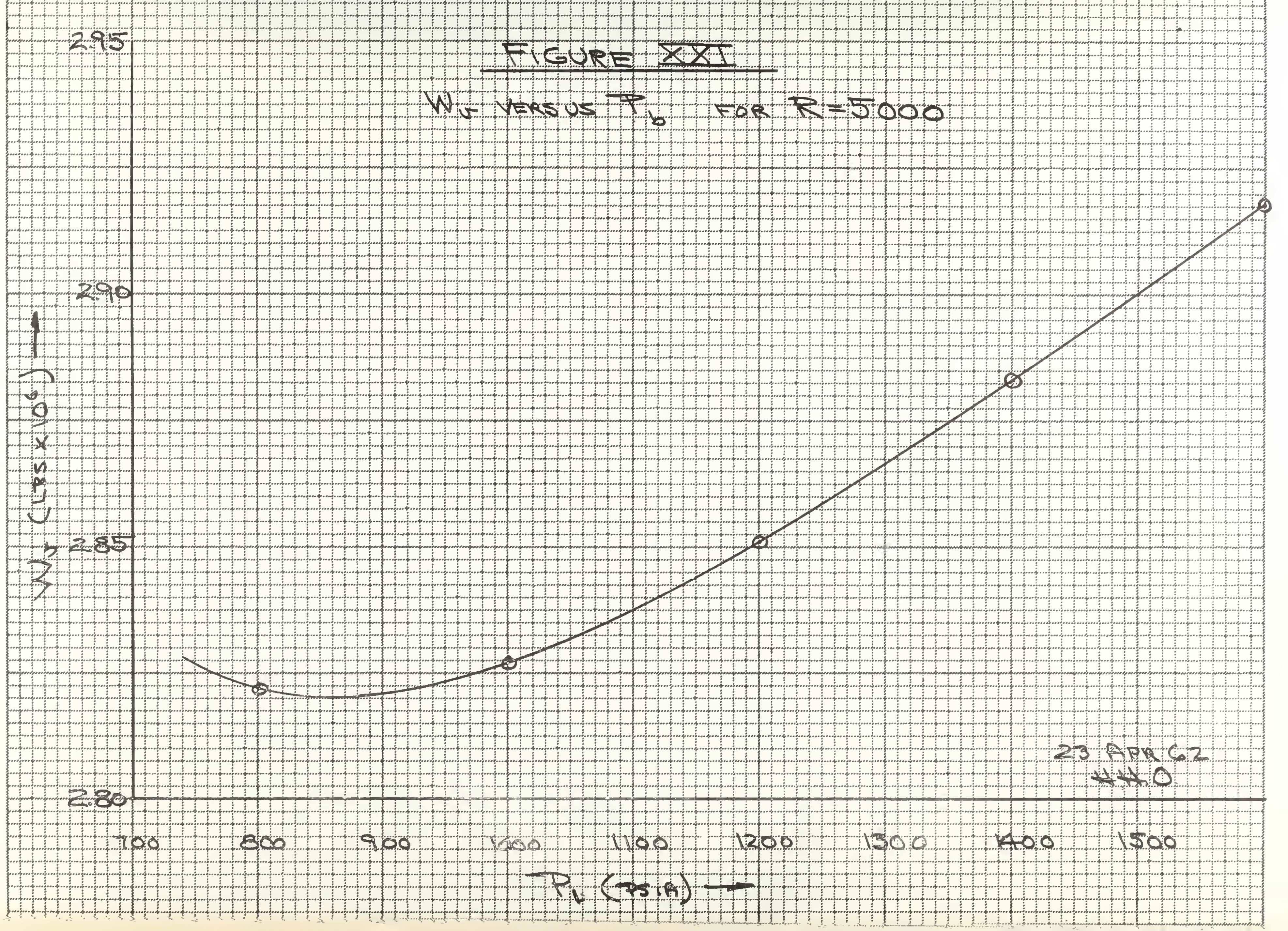


FIGURE XXI
W_J VERSUS P_b FOR R=5000
2.95
2.90
2.85
2.80
W_J (LBS x 10^6) →
700
800
900
1000
1100
1200
1300
1400
1500
P_b (PSIA) →
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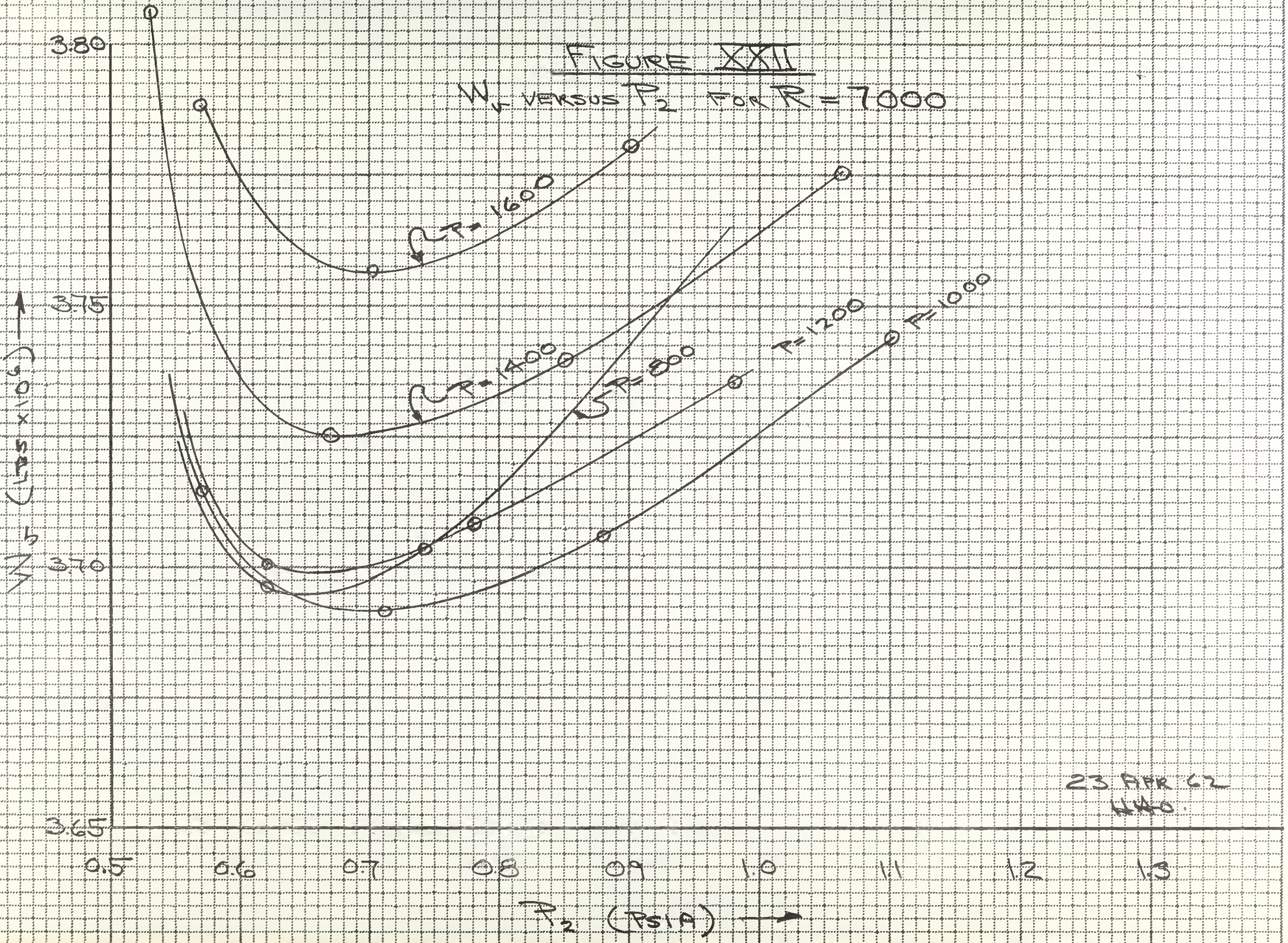
#4.0

FIGURE XXII

W_f VERSUS P_2 FOR R = 7000



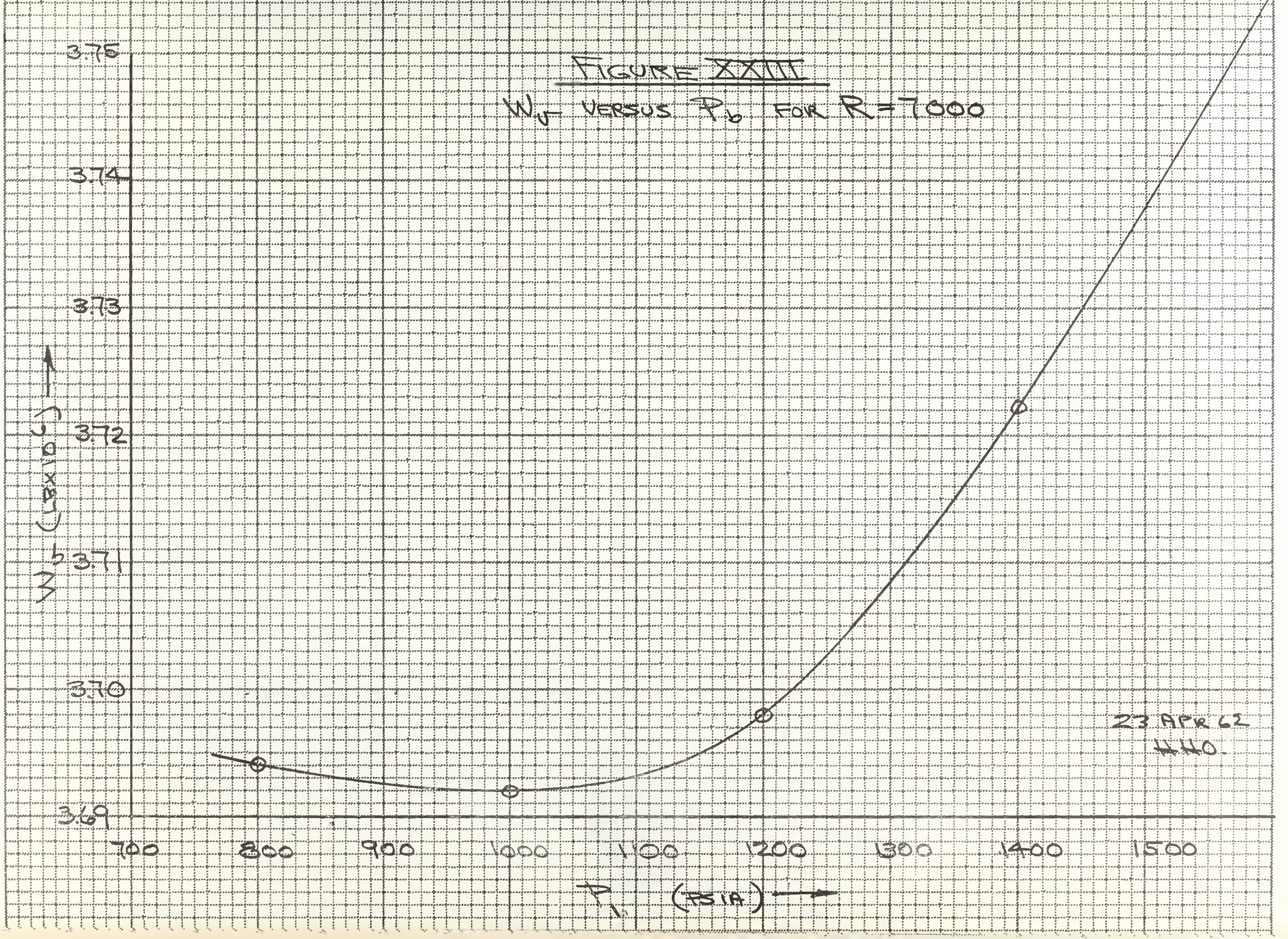


FIGURE XXIII
W_J VERSUS P_b FOR R = 7000
3.75
3.74
3.73
3.72
3.71
3.70
3.69
W_J (LB x 10^7)
700
800
900
1000
1100
1200
1300
1400
1500
P_b (PSIA)
23 APR 62
HHO.

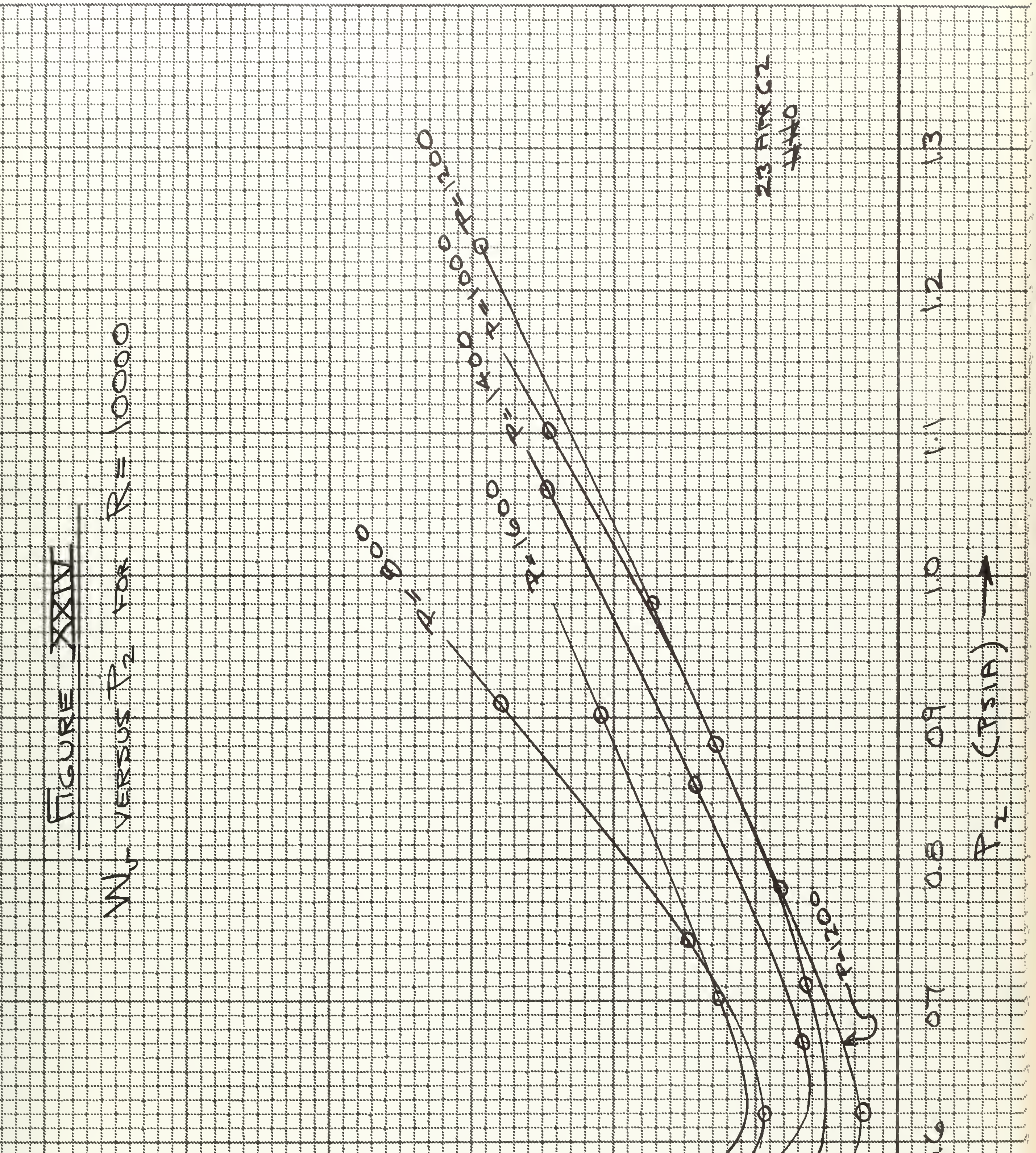


FIGURE XXIV
W VERSUS P2 FOR R= 10000
5.25
5.20
5.15
5.10
5.05
5.00
4.95
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
1.3
P2 (PSIA)
R=1200

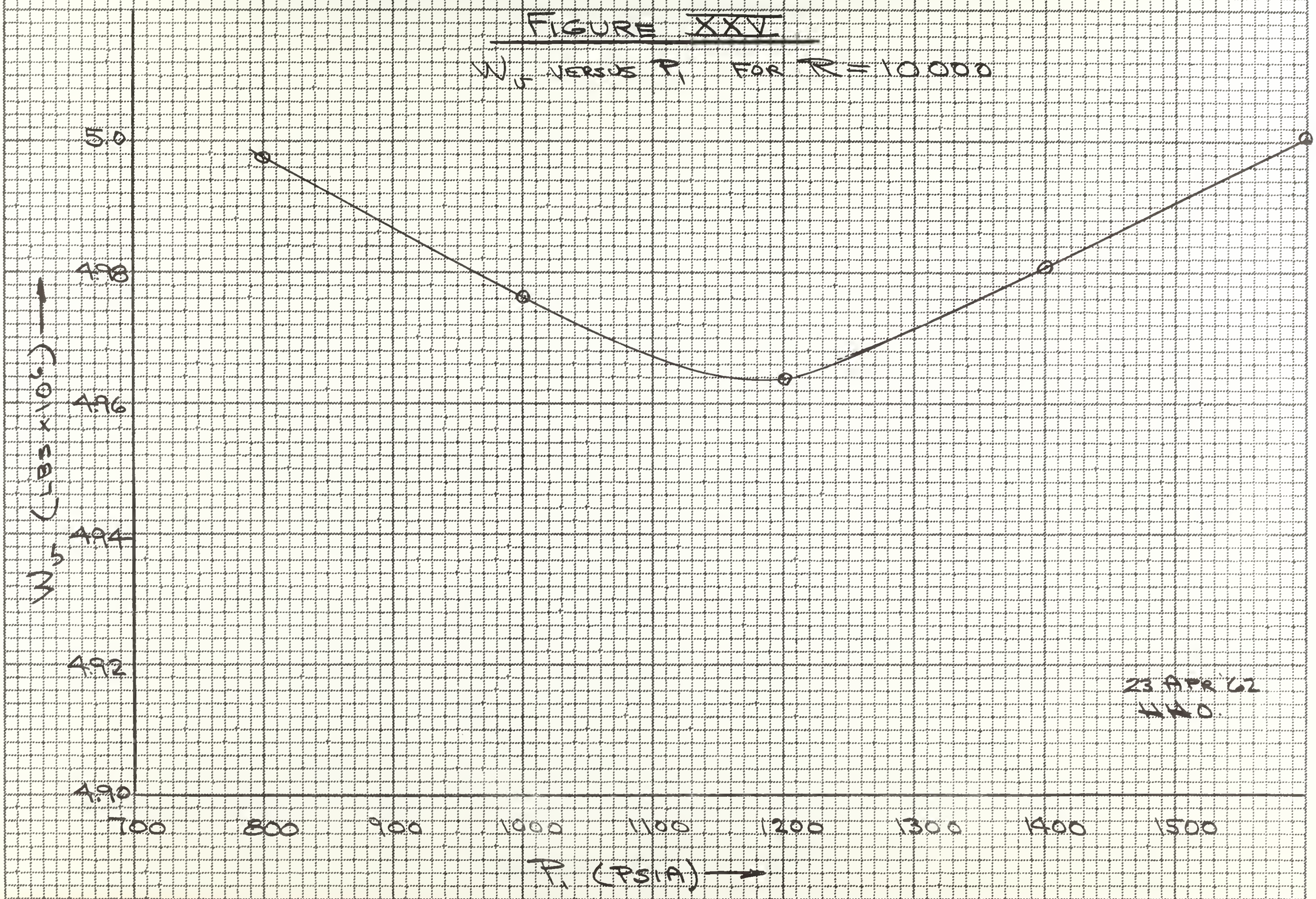


FIGURE XXV
W_5 VERSUS P_1 FOR R = 10,000
5.0
4.98
4.96
4.94
4.92
4.90
W_5 (LBS x 10^6) →
700
800
900
1000
1100
1200
1300
1400
1500
P_1 (PSIA) →
23 APR '62
HHO

FIGURE XXVI

OPTIMAL CURVES

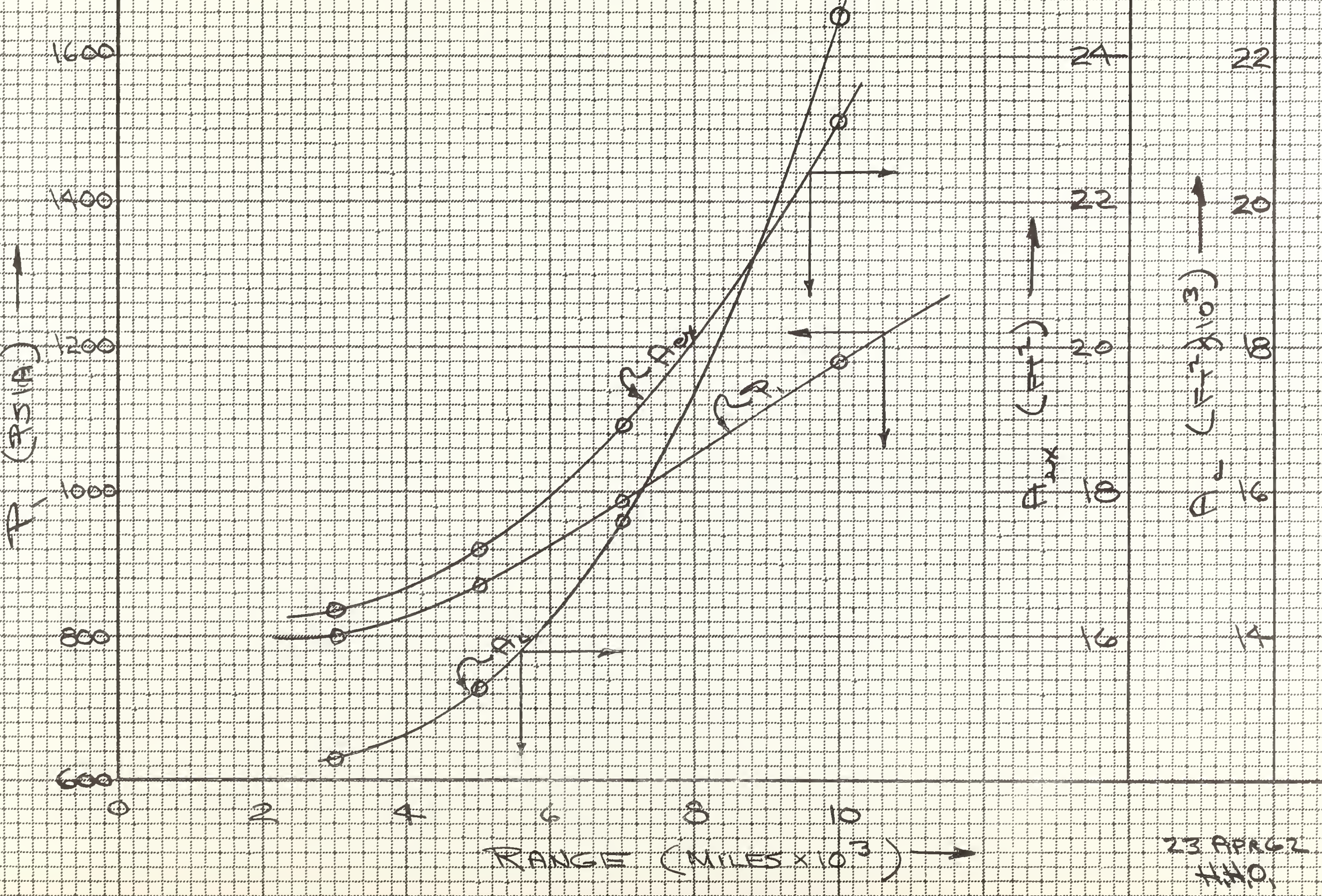


Table III.

Turbine and Condenser Weight Versus Leaving Loss

G_t = 73,200 lb./hr., p_1 = 1000 psia, h_1 = 1534 BTU/lb.

δb_{1-3} = 139 BTU/lb. h_2^o = 1081 BTU/lb.

x $\cong$ 1.6% moisture

Leaving loss, BTU/lb.	1	2	3	4	5	7
P_2, psia.	0.59	0.58	0.57	0.55	0.54	0.51
v_2' ft.^{3}/lb.	542	551	561	582	592	621
A_{ex}, ft.^{2}	49.0	35.3	29.3	26.4	24.0	21.2
W_t x 10^{-3}, lbs.	144.5	114.3	101.6	95.3	90.0	83.9
T_{sat}, ^oF	84.7	84.1	83.5	82.5	81.8	80.2
W_c x 10^{-3}, lbs.	150.7	158.7	168.2	188.8	208.5	285.5
ΣW, x 10^{-3}, lbs.	295.2	273.0	269.8	284.1	298.5	369.4

TABLE IV

WEIGHTS AND AVAILABILITY BALANCES ($P = 800$)

ITEM	ΔbG_i	W_i	ΔbG_i	W_i	ΔbG_i	W_i	ΔbG_i	W_i	ΔbG_i	W_i	ΔbG_i	W_i
MAIN FEED PUMP	0.67	23.4	0.69	23.6	0.70	23.9	0.72	24.2				
BOILERS	61.70	401.4	62.76	407.0	64.00	413.4	65.62	422.0				
TURBINE AND CONDENSER	15.20	233.0	16.40	187.0	17.50	165.5	19.90	145.5				
EVAPORATOR	1.25	—	1.25	—	1.25	—	1.25	—				
DEAERATOR	2.25	5.2	2.08	5.3	1.85	5.4	1.43	5.6				
TURBO - GENERATORS	5.45	—	5.45	—	5.45	—	5.45	—				
GALLEY, H.W. AND LAUNDRY	0.84	—	0.84	—	0.84	—	0.84	—				
OIL FUEL HEATER	0.03	—	0.03	—	0.03	—	0.03	—				
F.D. BLOWERS L.O. PUMPS F.O. PUMPS	1.76	—	1.80	—	1.85	—	1.94	—				
S.H.P.	33.10	—	33.10	—	33.10	—	33.10	—				
OTHER ΔbG	1.17	—	1.15	—	1.45	—	0.98	—				
G_b	—	106.3	—	108.0	—	110.0	—	112.9				
W_{m_r}	—	663.0	—	622.9	—	608.2	—	597.3				
η_{th}	.1918	—	.1889	—	.1855	—	.1813	—				
W_r $R = 3000$	—	1963	—	1943	—	1952	—	1973				
W_r $R = 5000$	—	2830	—	2823	—	2849	—	2889				
W_r $R = 7000$	—	3696	—	3703	—	3745	—	3806				
W_r $R = 10000$	—	4896	—	5023	—	5089	—	5181				
P_2	—	0.62	—	0.74	—	0.91	—	1.38				

NOTE: THROUGHOUT TABLE — UNITS ARE: W_i — LBS x 10^{-3}; ΔbG_i — btu/hour x 10^{6}; P_2 — PSIA; G_b — LBS/HR x 10^{-3}

TABLE IV
WEIGHTS AND AVAILABILITY BALANCES (P = 1000)

ITEM	ΔbG_i	W_i	ΔbG_i	W_i	ΔbG_i	W_i	ΔbG_i	W_i	ΔbG_i	W_i	ΔbG_i	W_i
MAIN FEED PUMP	0.87	29.9	0.88	30.3	0.90	30.5	0.91	30.8	0.93	31.3		
BOILERS	61.47	444.0	62.52	450.0	63.47	455.6	64.46	461.4	66.33	473.2		
TURBINE AND CONDENSER	14.92	270.0	16.20	191.4	17.30	161.5	18.50	148.2	20.80	133.8		
EVAPORATOR	1.25	—	1.25	—	1.25	—	1.25	—	1.25	—		
DEAERATOR	2.25	5.1	2.09	5.2	1.87	5.3	1.66	5.4	1.24	5.5		
TURBO-GENERATORS	5.39	—	5.39	—	5.39	—	5.39	—	5.39	—		
GALLEY, H.W. AND LAUNDRY	0.86	—	0.85	—	0.87	—	0.86	—	0.86	—		
OIL FUEL HEATER	.03	—	.03	—	.03	—	.03	—	.03	—		
F.D. BLOWERS L.O. PUMPS, F.O. PUMPS	1.73	—	1.77	—	1.82	—	1.86	—	1.96	—		
S.H.P.	33.10	—	33.10	—	33.10	—	33.10	—	33.10	—		
OTHER ΔbG	1.05	—	0.97	—	0.98	—	0.93	—	0.79	—		
G_b	—	104.3	—	106.2	—	107.7	—	109.3	—	112.3		
W_{m_v}	—	749.0	—	676.9	—	652.9	—	645.8	—	643.8		
η_{th}	.1963	—	.1930	—	.1906	—	.1878	—	.1828	—		
W_v R = 3000	—	2019	—	1969	—	1961	—	1974	—	2008		
W_v R = 5000	—	2866	—	2830	—	2833	—	2859	—	2917		
W_v R = 7000	—	3713	—	3691	—	3706	—	3744	—	3827		
W_v R = 10,000	—	4983	—	4983	—	5014	—	5072	—	5191		
P_2	—	0.57	—	0.71	—	0.88	—	1.10	—	1.66		

TABLE IV

WEIGHTS AND AVAILABILITY BALANCES (P = 1200)

ITEM	$\Delta b G_i$	W_i	$\Delta b G_i$	W_i	$\Delta b G_i$	W_i	$\Delta b G_i$	W_i	$\Delta b G_i$	W_i	$\Delta b G_i$	W_i
MAIN FEED PUMP	1.07	36.1	1.10	36.7	1.12	37.1	1.12	37.4	1.15	37.7	1.16	38.0
BOILERS	60.50	480.4	62.13	491.4	63.20	498.4	64.15	504.8	65.14	511.4	65.72	515.6
TURBINE AND CONDENSER	14.40	492.9	15.70	221.5	16.80	173.6	18.10	150.9	19.20	140.2	21.5	128.1
EVAPORATOR	1.25	—	1.25	—	1.25	—	1.25	—	1.25	—	1.25	—
DEAERATOR	2.26	5.0	2.12	5.1	1.93	5.2	1.69	5.3	1.45	5.4	1.02	5.4
TURBO-GENERATORS	5.35	—	5.35	—	5.35	—	5.35	—	5.35	—	5.35	—
GALLEY, H.W. AND LAUNDRY	0.88	—	0.89	—	0.89	—	0.88	—	0.88	—	0.88	—
OIL FUEL HEATER	0.03	—	0.03	—	0.03	—	0.03	—	0.03	—	0.03	—
F.D. BLOWERS L.O. PUMPS F.O. PUMPS	1.69	—	1.74	—	1.78	—	1.83	—	1.87	—	1.96	—
S.H.P.	33.10	—	33.10	—	33.10	—	33.10	—	33.10	—	33.10	—
OTHER $\Delta b G$	0.49	—	0.89	—	0.97	—	0.81	—	0.88	—	0.47	—
G_b	—	101.7	—	104.3	—	106.0	—	107.5	—	109.1	—	110.0
W_{m_r}	—	1014.4	—	754.7	—	714.3	—	698.4	—	694.7	—	687.1
η_{th}	.2021	—	.1975	—	.1943	—	.1916	—	.1888	—	.1876	—
W_r R = 3000	—	2248	—	2017	—	1998	—	2000	—	2015	—	2016
W_r R = 5000	—	3071	—	2859	—	2853	—	2867	—	2896	—	2902
W_r R = 7000	—	3894	—	3700	—	3709	—	3735	—	3776	—	3788
W_r R = 10,000	—	5127	—	4963	—	4992	—	5036	—	5097	—	5117
P_2	—	0.50	—	0.62	—	0.78	—	0.98	—	1.23	—	1.88

TABLE IV

WEIGHTS AND AVAILABILITY BALANCES (P = 1400)

ITEM	$\Delta_b G_i$	W_i	$\Delta_b G_i$	W_i	$\Delta_b G_i$	W_i	$\Delta_b G_i$	W_i	$\Delta_b G_i$	W_i	$\Delta_b G_i$	W_i
MAIN FEED PUMP	1.31	43.0	1.33	43.5	1.35	43.9	1.37	44.3	1.39	44.7	1.43	45.5
BOILERS	61.72	531.2	62.77	538.8	63.83	546.4	64.79	553.6	65.74	561.0	67.63	575.2
TURBINE AND CONDENSER	15.10	336.7	16.20	199.9	17.60	161.5	18.80	144.9	19.90	135.0	22.30	125.3
EVAPORATOR	1.25	—	1.25	—	1.25	—	1.25	—	1.25	—	1.25	—
DEAERATOR	2.11	5.0	1.92	5.1	1.74	5.2	1.49	5.3	1.26	5.3	0.77	5.5
TURBO-GENERATORS	5.32	—	5.32	—	5.32	—	5.32	—	5.32	—	5.32	—
GALLEY, H.W AND LAUNDRY	0.90	—	0.90	—	0.90	—	0.89	—	0.90	—	0.90	—
OIL FUEL HEATER	.04	—	.04	—	.04	—	.04	—	.04	—	.04	—
F.D. BLOWERS L.O. PUMPS F.O. PUMPS	1.71	—	1.75	—	1.80	—	1.85	—	1.89	—	1.98	—
SHP	33.10	—	33.10	—	33.10	—	33.10	—	33.10	—	33.10	—
OTHER $\Delta_b G$	0.90	—	0.98	—	0.75	—	0.91	—	0.71	—	0.56	—
G_b	—	103.0	—	104.7	—	106.4	—	107.9	—	109.4	—	112.4
W_{m_r}	—	915.9	—	787.3	—	757.0	—	748.1	—	746.0	—	751.5
η_{th}	.2013	—	.1981	—	.1949	—	.1922	—	.1899	—	.1848	—
W_r R = 3000	—	2155	—	2046	—	2036	—	2045	—	2059	—	2101
W_r R = 5000	—	2980	—	2885	—	2889	—	2910	—	2935	—	3001
W_r R = 7000	—	3806	—	3725	—	3742	—	3775	—	3810	—	3900
W_r R = 10,000	—	5050	—	4983	—	5021	—	5072	—	5123	—	5250
P_2	—	0.53	—	0.67	—	0.85	—	1.06	—	1.34	—	2.00

TABLE IV

WEIGHTS AND AVAILABILITY BALANCES ($P = 1600$)

ITEM	$\Delta_b G_i$	W_i	$\Delta_b G_i$	W_i	$\Delta_b G_i$	W_i	$\Delta_b G_i$	W_i	$\Delta_b G_i$	W_i	$\Delta_b G_i$	W_i
MAIN FEED PUMP	1.57	49.6	1.59	50.2	1.62	50.7	1.64	51.2	1.67	51.7		
BOILERS	62.19	578.2	63.28	587.6	64.31	596.0	65.24	603.0	66.22	615.2		
TURBINE AND CONDENSER	15.50	272.1	16.80	183.6	18.70	153.0	19.20	137.5	20.30	129.1		
EVAPORATOR	1.25	—	1.25	—	1.25	—	1.25	—	1.25	—		
DEAERATOR	1.97	5.0	1.79	5.1	1.55	5.2	1.27	5.2	1.06	5.3		
TURBO-GENERATORS	5.30	—	5.30	—	5.30	—	5.30	—	5.30	—		
GALLEY, H.W. AND LAUNDRY	0.90	—	0.91	—	0.91	—	0.91	—	0.91	—		
OIL FUEL HEATER	0.04	—	0.04	—	0.04	—	0.04	—	0.04	—		
F.D. BLOWERS L.O. PUMPS F.O. PUMPS	1.68	—	1.72	—	1.76	—	1.81	—	1.85	—		
S.H.P.	33.10	—	33.10	—	33.10	—	33.10	—	33.10	—		
OTHER $\Delta_b G$	0.89	—	0.80	—	0.08	—	0.74	—	0.77	—		
G_b	—	103.0	—	104.7	—	106.4	—	106.9	—	109.4		
W_{m_r}	—	904.9	—	826.5	—	804.9	—	796.9	—	801.3		
η_{th}	.2019	—	.1986	—	.1956	—	.1933	—	.1906	—		
W_r R= 3000	—	2140	—	2082	—	2080	—	2087	—	2110		
W_r R= 5000	—	2963	—	2919	—	2929	—	2947	—	2982		
W_r R =7000	—	3787	—	3756	—	3779	—	3807	—	3854		
W_r R= 10000	—	5022	—	5012	—	5054	—	5097	—	5162		
P_2	—	0.57	—	0.70	—	0.90	—	1.15	—	1.45		

IV. DISCUSSION OF RESULTS

The results obtained are extremely dependent on the weight equations used and the thermodynamic assumptions made.

The boiler weight equation is a particularly significant one. The equation (A-1) used in these calculations is believed to hold to a reasonable degree of accuracy over a limited range on both sides of 1200 psia but may show large errors for large departures or different families of equipment.

Table IV indicates that the main feed pump must be charged with an increasing share of the irreversibilities as pressure increases. This is partly compensated for by a corresponding decrease in deaerator irreversibilities. However, this decrease does not nearly compensate for the increase in main feed pump irreversibilities as Table IV indicates.

Figures XIX, XXI, XXIII and XXV indicate that the optimal weight versus throttle pressure curve for a given range has a flat optimum at long ranges but that pressure variations become much more significant at short to medium ranges.

For example, a pressure 400 psi above optimum pressure at a range of 3,000 miles gives a weight increase of 54,000 pounds while the same departure at a range of 10,000 miles gives a weight increase of 34,000 pounds. For ranges of

3,000 and 10,000 miles respectively, the increase in variable weight is 2.78 % and 0.685 % of the total variable weight at optimum pressure. That is, W_v is about four times as dependent on pressure at 3,000 miles as at 10,000 mile range.

It would therefore appear that for a throttle pressure standardization applying to ships with different missions it would be much better to err on the low pressure side than on the high, since for long range missions a small weight penalty is paid for too low a pressure, while at short ranges a relatively much larger penalty is paid for too high a throttle pressure. Added bonuses are decreased initial cost and the diminished maintenance problems that go with lower pressures. Operating costs will, however, increase with decreasing pressure.

Figure XXVI shows optimal pressures rather lower than present United States Navy practice.

Optimum condenser pressures (based on the assumed 75° sea water temperatures) decrease slightly with increasing range and agree very closely with present United States Navy practice (1.35" Hg. for DLG-6 and approximately 1.4" Hg. from Figure XX.) It is interesting to note that the terminal temperature difference is relatively constant over the ranges considered, varying from approximately 11°F to 6°F at ranges of 3,000 and 10,000 miles respectively. At long ranges, the increased condenser weight is offset by the increase in cycle efficiency due to the lower condenser temperature.

V. CONCLUSIONS

The results of the detailed calculations made on thirty perturbations of the DLG propulsion plant do not show a wide diversity of total weight. The best condenser pressure was practically the same for every top pressure. It was proved that the condenser and L.P. turbine can be matched independently. Finally, optimum top pressure was found to vary somewhat below the range of top pressures currently in use, depending on range.

Choice of top pressure is seen to be the least clear-cut, since it must be influenced by the desire to set uniform steam conditions in the fleet and reliability and cost considerations certainly will enter as well. Were selection of top pressure to be based solely on weight considerations, doubt might exist as to the accuracy of the formulas in reflecting the machinery weight penalties inflicted in order to attain the higher fuel efficiency of higher pressure.

This thesis has demonstrated to the investigators that the best choice of top pressure relies on a complicated interplay of many factors. While "brute force" methods may be deplored, choice of top pressure by any less accurate method may easily produce a misleading result.

The methods introduced in this paper are refinements of "brute force". Choice of leaving loss is by simple two-stage dynamic programming, and reduces the total number of

comparisons necessary by an order of magnitude. The availability methods used allow the investigator to weigh each component considered with its true importance to the problem.

The authors are aware that the full solution of the example problem is a matter which involves a great number of considerations which have not even been mentioned thus far. Full power and astern performance must also be carefully evaluated. The true scope of the problem is vast indeed.

The example does demonstrate that availability analysis is a necessary implement to clear thinking about machinery optimization. Available energy concepts based on the second law of thermodynamics do not replace the first law heat balance; they are an essential accessory in the manipulation of the variables involved.

VI. RECOMMENDATIONS

Design of naval propulsion plants for minimum weight and thermodynamic analysis of their cycles requires detailed heat and availability balances. It can be expected that one day this work will be done by electronic computing machines. The device of charging components with a certain weight of fuel depending on their irreversibility will be useful in the comparisons made in the computer programs.

For the present, there are many aspects of the weight optimization problem which can be investigated still further.

Turbine performance may possibly be improved upon. The power level at which maximum turbine stage efficiency occurs is subject to optimization [2]. Even the most desirable number of turbine casings is still open to question; the Royal Canadian Navy is using single casing designs in some destroyer types.

The desirability of increasing top temperature to the limit may be of illusory value in naval plants. Temperature should be made a variable in the same fashion that pressure was in this paper. Higher temperatures promise ideal cycle efficiency gains. Machinery weights will increase and the availability balance may reveal other limitations.

A study of boiler air preheating methods appears to offer still another field for optimization.

In any of the optimization processes the addition of reliability considerations in a quantitative manner, possibly using the statistical methods of operations research, will be a valuable contribution.

In the accomplishment of these suggested studies, as in the investigation of optimum cost or weight relationships, the authors strongly recommend that the investigator make full use of the second law.

VII. APPENDIX

APPENDIX A

DERIVATION OF WEIGHT EQUATIONS

1. Boiler

Two basic assumptions were made in the derivation of the boiler weight equation; i) boiler steam flow at cruising was assumed to be a constant fraction of full power boiler steam flow, ii) boiler heat release rate was assumed to remain constant. If heat release rate is constant:

$$\text{Boiler volume, } V_b \sim L_b^3 \sim G_b \Delta h_A$$

where L_b = a characteristic boiler dimension,

G_b = boiler steam mass flow,

and Δh_A = boiler enthalpy change.

Δh_A is constant,

$L_b^3 \sim G_b$.

Considering walls and refractory, pressure parts and contained water separately and summing we will arrive at the boiler weight equation.

a. Walls and Refractory

$$\text{Wall area, } A_w \sim L_b^2$$

Assuming wall thickness (i.e., refractory and insulation) is constant and that structural weight per square foot is constant, then

$$W_{w+r} \sim L_b^2 \sim G_b^{2/3}$$

where W_{w+r} = wall and refractory weight.

b. Water

It was assumed that boiler water capacity varied directly as G_b — $W_w \sim G_b$,

where W_w = water weight.

c. Pressure Parts

Assuming that the coefficients of heat transfer remain constant and that temperature differentials are approximately the same, heating surface area = $A_{hs} \sim L_b^3$. Now $W_{pp} \sim A_{hs}\, t$, where W_{pp} = weight of pressure parts and t = wall thickness. Using the hoop stress formula:

$$t \sim p_b$$

where p_b = boiler pressure.

$$W_{pp} \sim A_{hs} p_b$$

or

$$W_{pp} \sim G_b p_b \; .$$

Summing the three parts and applying proportionality constants,

$$W_b = k_{w+r}\, G_b^{2/3} + k_{pp} G_b p_b + k_w\, G_b \; . \qquad \text{(A-1)}$$

Equating the three parts of this equation to appropriate parts of the DLG-6 boiler results in:

$$W_b = (.3544 + .00236 p_b) G_b + 75.2\, G_b^{2/3}$$

The above relation is based on a cruising-to-full-power boiler steam flow ratio of 0.144. It appears to hold reasonably well over a fairly wide range in boiler pressures and flows if corrections are made for the flow ratio. Figure XVII shows the variation of boiler weight with pressure and steam flow. These weights were assumed to include all boiler fittings and foundations.

2. Main Engines

White and Smith's [4] steam turbine weight equation was found to correspond very well with the machinery on which this project was based although changes were made in constants. The equation used was:

$$W_t = 2190\ A_{ex} + 16{,}130\ A_a \left[.577\left(\frac{p_1}{p_o}\right) + .423\right] \qquad \text{(A-2)}$$

where A_a = last H.P. stage annulus area based on 310 ft./sec. axial velocity,

A_{ex} = L.P. exhaust annulus area,

p_o = 1200 psig.

3. Main Feed Pump

Using Figure 16 in White and Smith [4] and interpolating linearly since C is almost linear for pressure from 800 to 1600 psi,

$$C = 0.001788\ p' - 0.223$$

where p' = pressure after pump in psig.

Altering Stevens [17] formula to fit 250°F feed water and a boiler cruising to full power steam flow ratio of 0.144,

$$W_{mfp} = 13.94\ CG_b^{5/8} \quad \text{(Pounds)}$$

or $$W_{mfp} = (.0249\ p' - 3.25)\ G_b^{5/8}\ . \qquad \text{(A-3)}$$

This equation proved to give weights which were slightly low (about 10%) for the DLG-6 but was used in calculations as it was felt that weight variations were probably quite close to those which would actually obtain. Weights derived from equation (A-3) are plotted in Figure XVI.

4. Condenser

Here it was assumed that:

i) $G_t \sim$ SHP,

ii) h_2 = constant,

iii) cooling water velocity varies directly with ship speed,

iv) sea water temperature is $75^{\circ}F$,

v) heat transfer coefficient, U = constant,

vi) cooling water temperature rise at full power = $15^{\circ}F$.

Now, for the DLG-6 power ratio, $r_p = \frac{\text{full power}}{\text{cruising power}} = 7$,

and the approximate velocity ratio, $r_v = \frac{\text{full speed}}{\text{cruising speed}} = (r_p)^{1/3} = 1.92$.

For a given number of tubes, $G_{cw} \sim V_k$.

$$\triangle T_{cw} \quad \frac{15}{7} \times 1.915 = 4.1^{\circ}F\ .$$

Using the arithmetic mean,

$$\triangle T_m = \frac{T_s - 75 + T_s - 79.1}{2} = T_s - 77.05$$

Now if U = 490,

$(h_2 - h_3)$ = 980,

$$A_c \simeq \frac{2\ G_t}{(T_s - 77)} \quad . \qquad \text{(A-4)}$$

Now from DLG-6 data,

$$W_{c_{wet}} = 5.06\ A_c + 27{,}250$$

hence,

$$W_{c_{wet}} = 10.12\ \frac{G_t}{(T_s - 77)} + (27{,}250)m \qquad \text{(A-5)}$$

where m is number of condensers.

This equation appears to check very well with fact.

5. Steam Lines

Pressure drop was assumed a constant per cent of throttle pressure

$$W_{sl} = C \times D \times t$$

where C = constant, D = pipe diameter and t = wall thickness. Assuming friction factor, f = constant,

$$\Delta p = \frac{4f \; V^2 L}{2D} \; .$$

Now $\rho = \frac{P}{RT}$, assuming steam behaves as a perfect gas,

but $$V = \frac{Gv}{A} = \frac{4G}{\rho \pi D^2} \; ,$$

so $$V^2 = \frac{kG^2}{\rho^2 D^4} \; .$$

Now $$\Delta p = 2 \; fL \; \frac{p}{RT} \; k \; \frac{G^2}{\rho^2 D^4} \times \frac{1}{D} = (2fL) \frac{RT}{p} \frac{G^2}{D^5}$$

so $$\Delta p = k_1 \frac{G^2}{pD^5} \; , \quad \text{for constant temperature.}$$

Now $$\Delta p = k_2 P \; ,$$

so $$p^2 = k_3 \frac{G^2}{D^5} \; ,$$

hence $$D = \left(k_3 \frac{G^2}{p^2} \right)^{1/5} = k_4 \frac{G^{2/5}}{p^{2/5}}$$

Now using hoop stress,

$$t = \frac{pD}{2\sigma} \; ,$$

$$W_{sl} = \frac{CpD^2}{2\sigma} = k_5 pD^2$$

$$\therefore \quad W_{sl} \quad k_5 \; G^{4/5} \; p^{1/5}$$

This equation was not used in calculations because steam piping weight is rather insensitive to pressure for a given flow and flows used in calculations did not differ by large amounts. In addition the weight of the high pressure steam piping in a plant of the destroyer type is not a very large fraction of the total plant and fuel weight.

6. Condensate Lines

Here $\Delta p = k_1 \frac{V^2}{D}$

but ρ = constant, so $\Delta p = k_2 \frac{V^2}{D}$.

By continuity,

$$V = \frac{4 Gv}{\pi D^2} = k_3 \frac{G}{D^2} .$$

Hence

$$\Delta p = k_4 \frac{G^2}{D^5} .$$

Again keeping $\frac{\Delta p}{p}$ constant

$$D = k_5 \frac{G^{2/5}}{p^{1/5}} \quad \text{or} \quad D^2 = k_5^2 \frac{G^{4/5}}{p^{2/5}}$$

now $W = C \cdot D \cdot t$

and $t = \frac{pD}{2\sigma} = k_6 pD.$

Hence

$$W_{cl} = k_7 p D^2 \text{ , and}$$

$$W_{cl} = k_8 G^{4/5} p^{3/5}$$

7. Water and Circulating Lines

Here pressure and allowable stress are constant, so

$$t = k_1 D .$$

Now $V = \frac{Gv}{A} = k_2 \frac{G}{D^2}$

and $W_w = CDt = k_3 D^2$,

but $D^2 = k_2 \frac{G}{V}$

and $V \cong$ constant.

Therefore $D^2 = k_4 G$.

and $W_w = k_5 G$.

8. Deaerator

Using Stevens' [17] data, deaerator weight is seen to be approximated by a straight line relationship in G_b,

$$W_{da} = k_{da} G_b .$$

In fact, for the range of flows used in the calculations W_{da} could just as well have been assumed constant.

APPENDIX B

SAMPLE CALCULATION

1. Data

Cruising speed : 20 kts.

SHP_{20} : 12,100

$SHP_{(rated)}$: 85,000

T_1 : 1050°F

$e_m = e_t e_g = 0.98 \times 0.95 = 0.931.$

2. Turbine and Condenser (Refer to Figure B-II)

Assume p_1 = 800, 1000, 1200, 1400 and 1600 psia.

For each pressure, G_t = 73,200, 75,000, 76,800, 78,700 and 82,500 lb./hr.

For each (p_1, G_t) combination, leaving loss = 1, 2, 3, 4, 5, and 7 BTU/lb.

Consider the case : p_1 = 1000 psia

G_t = 73,200 lb./hr.

L.L.= 3 BTU/lb.

h_1 = 1534 BTU/lb. from steam tables[18].

$$G_t \Delta h_t = \frac{SHP}{e_m}$$

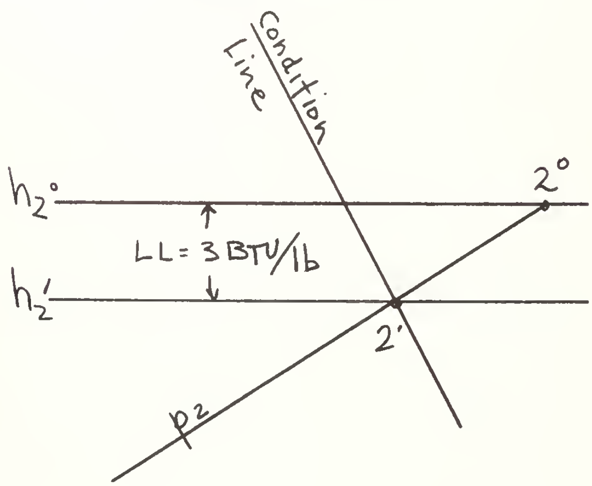
$$\Delta h_t = \frac{12,100 \times 2545}{.931 \times 73,200} = 453 \text{ BTU/lb.}$$

$$h_2{}^o = h_1 - \Delta h_t = 1081 \text{ BTU/lb.}$$

Turbine condition line is laid down from state 1, parallel to DLG condition line, which is assumed a straight line from 1195 psig, 925°F through 1.35 In. Hg. Abs., 6.6% moisture. Stagnation state 2° is picked off steam chart as indicated in Figure B-I.

Figure B-I

Method of Locating State 2^o



S_2^o = 1.996,

p_2 = 0.57 psia, and

x = 1.6% moisture are read from the Mollier Chart. [18]

$V_2' = 561 \frac{ft.^3}{lb.}$ from the steam tables [18].

Leaving loss = 3 BTU = $\frac{V_2'^2}{2gJ}$

$$V_2' = \sqrt{2 \times 3 \times 778 \times 32.2} = 387 \text{ ft./sec.}$$

$$A_{ex} = \frac{v_2' G_t}{V_2'} = \frac{561 \times 73,200}{387 \times 3600} = 29.5 \text{ ft.}^2$$

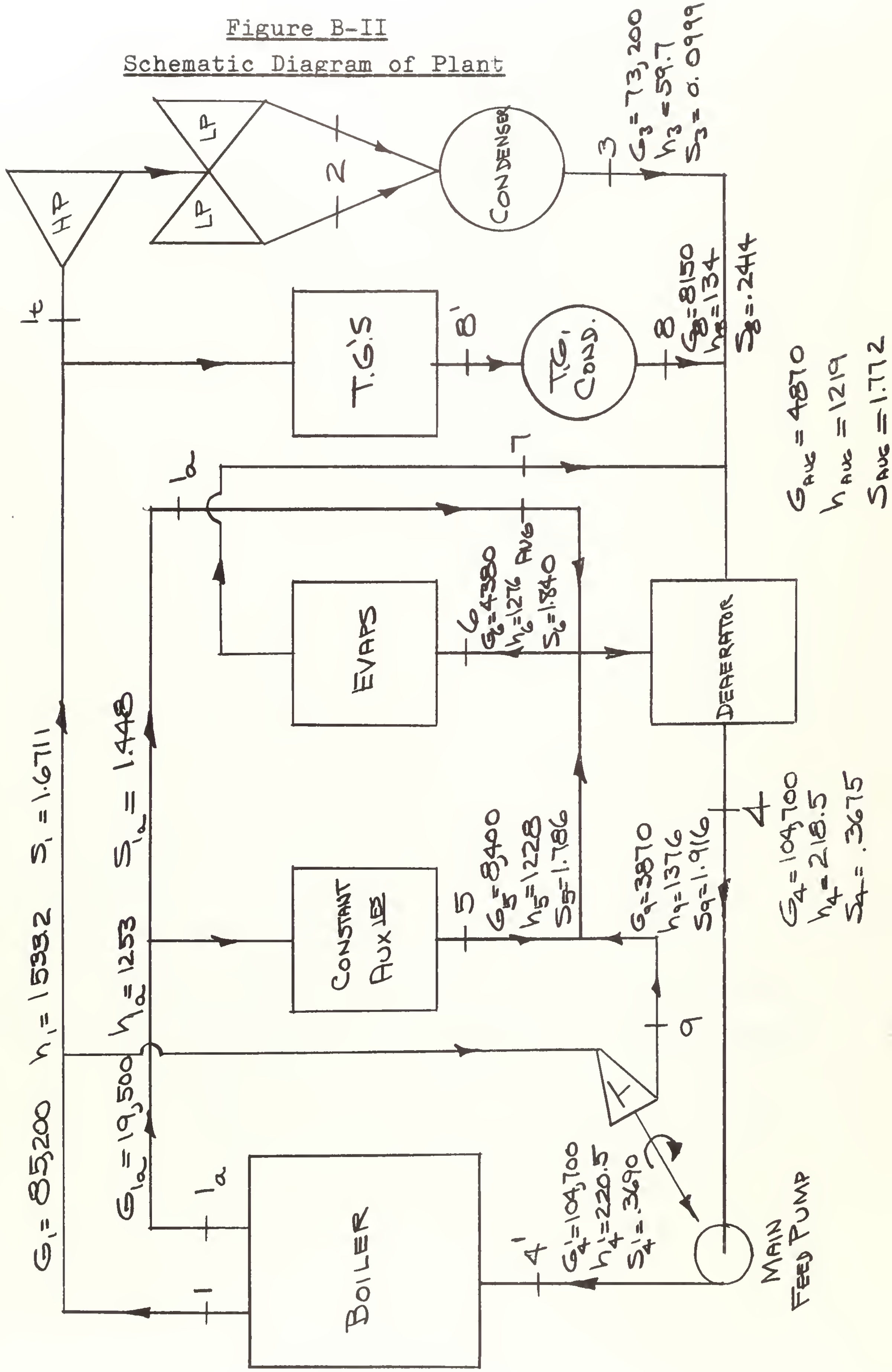
In sizing A_a, the annulus area of the last H.P. stage, the exhaust condition was assumed to lie on the condition line at the point where $h_a = h_1 - \frac{\Delta h}{2}$.

$$h_a = 1538 - \frac{453}{2} = 1312 \text{ BTU/lb.}$$

$$v_a = 39.2 \text{ ft.}^3\text{/lb.}$$

Figure B-II

Schematic Diagram of Plant



$$A_a = \frac{G_t v_a}{V_a}, \quad \text{and } V_a = 310 \text{ ft./sec. constant.}$$

$$A_a = \frac{73,200 \times 39.2}{310 \times 3600} = 2.57$$

$$W_t = 2190\ A_{ex} + 16,130\ A_a\ [.577\ \frac{p_1}{1200} + .423] \qquad \text{(A-2)}$$

$$W_t = 101,600 \text{ lbs.}$$

T_s = 83.5°F corresponding to p_c = .57 psia.

$$W_c = \frac{10.12\ G_t}{T_s - 77} + 2 \times 27,250 = 168,200 \text{ lb.}$$

Irreversibility for all turbine-condenser combinations with G_t = 73,200, p_1 = 1000, is approximately $\Delta b_{1-3} - Wk_t$, computed for L.L. = 4 BTU/lb.

$$\Delta b_{1-3} - Wk_t = h_2^{\,o} - h_3 - T_o(s_1 - s_3)$$

$$= 1081 - 50.5 - 535\ (1.6711 - .0979)$$

$$= 189 \text{ BTU/lb.}$$

$$I_{t+c} = 189 \times 73,200 = 13.84 \times 10^6 \text{ BTU/hr.}$$

$$W_{t+c} = 101,600 + 168,200 = 269,800 \text{ lbs.}$$

This is plotted vs. leaving loss and compared with other turbine-condenser combinations for this p_1 and G_t. See Figure XI. Optimum leaving loss for this p_1, G_t combination can now be picked based on minimum condenser plus main engine weight. It corresponds to the calculated set : p_2 = 0.57, T_3 = 83.5°F, S_3 = .0999. Leakage irreversibility is added to that of turbine and condenser as follows:

Leakage (DLG-6) = 2080 lbs./hr.

b_1 = 639.8 BTU/lb.

Availability of make up feed, b_f = -1.9

$$I_{\ell} = 2080 \times 641.7 \times \frac{1000}{1200} = 1.11 \times 10^6 \text{ BTU/hr.}$$

$$I_{t+c\ total} = 13.84 + 1.11 = 14.95 \times 10^6 \text{ BTU/hr.}$$

3. Turbogenerators

TG output same as for DLG-6. Main feed booster pumps and main condensate pumps use about 10 Kw, so TG output can be assumed constant. It was also assumed that:

$$G_{tg} \Delta h_{tg} = \left(G_{tg} \Delta h_{tg}\right)_{DLG-6} = 2.56 \times 10^6$$

$$G_{tg} \Delta s_{tg} = \left(G_{tg} \Delta s_{tg}\right)_{DLG-6} = 3510$$

$$p_{2tg} = 3.06 \text{ In. Hg. abs.}$$

Condition line was plotted on Mollier chart [18] parallel to DLG-6 TG condition line. Properties of state 8′ were read off the chart. (See Figure B-II for state points.)

$$s_{8'} = 2.102$$

$$h_{8'} = 1221$$

$$\Delta h = 1534 - 1221 = 313$$

$$\Delta s = + .431$$

$$G_8 = \frac{2.56 \times 10^6}{313} = 8150 \text{ lb./hr.}$$

and for p_{2tg} = 3.06 In. Hg. abs.,

$$h_{8'} = 82.4$$

$$s_{8'} = 0.1550$$

T.G. air ejectors' condensate at 200°F, 390 lb./hr. DSH steam.

$$T_{1a} = T\ (\text{Sat, 1000 psig}) + 60^\circ = 605^\circ F$$

$$p_{1a} = 1000 \text{ psig.}$$

$$h_{1a} = 1253$$

$$s_{1a} = 1.448$$

After air ejector,

$$h_8 = 134.3$$

$$s_8 = .2414$$

$$\triangle h_{(AE)} = 1085$$

$$\triangle s_{(AE)} = 1.154$$

$$I_{tg} + Wk_{tg} = \triangle b_{tg} G_{tg} + \triangle b_{(AE)} G_{(AE)}$$

$$= 5.21 \times 10^6 + .18 \times 10^6$$

$$I_{tg} + Wk_{tg} = 5.39 \times 10^6 \text{ BTU/hr.}$$

4. Desuperheated Auxiliaries

F.D. blowers 1000 psig.

F.O. and L.O. service pumps 600 psig. throttled.

$\eta_{th} = 0.10$

Exhaust to 15 psig.

h_{1a}	= 1253	$h_{1a(600)}$	= 1253
s_{1a}	= 1.448	$s_{1a(600)}$	= 1.496
h_{9s}	= 984	$h_{5s(600)}$	= 1017
$h_{1a}-h_{9s}$	= 269	$(h_{1a}-h_{5s})_{600}$	= 236
$h_{1a}-h_9$	= 27	$(h_{1a}-h_5)_{600}$	= 24
h_9	= 1226	$h_{5(600)}$	= 1229
s_9	= 1.784	$s_{5(600)}$	= 1.787

$$G_{aux} \triangle h_{aux} = \frac{G_{b \text{ assumed}}}{G_{b \text{ DLG-6}}} [G_{aux} \triangle h_{aux}]_{\text{DLG-6}}$$

$$(G \triangle h)_{\text{FDB, DLG-6}} = 110 \times 10^3 \text{ BTU/hr.}$$

$$(G \triangle h)_{\text{600, DLG-6}} = 132 \times 10^3 \text{ BTU/hr.}$$

$$G_{b \text{ assumed}} = \frac{G_t}{.75} = \frac{73,200}{.75} = 97,500 \text{ lbs./hr.}$$

$$(G\Delta h)_{FDB} = \frac{97,500}{111,600} \times 110 \times 10^3 = 96.1 \times 10^3$$

$$(G\Delta h)_{600} = \frac{97,500}{111,600} \times 132 \times 10^3 = 115.2 \times 10^3$$

$$G_{FDB} = \frac{96.1 \times 10^3}{27} = 3.6 \times 10^3 \text{ lbs./hr.}$$

$$G_{600} = \frac{115.2}{24} = 4.8 \times 10^3 \text{ lbs./hr.}$$

To compute I for forced draft blowers and pumps:

$$I = [\Delta h - T_o(\Delta s)] G$$

$$I_{FDB} = 3.6 \times 10^3 [27 + 535(.336)] = .745 \times 10^6$$

$$I_{600} = 4.8 \times 10^3 [24 + 535(.339)] = \frac{.984 \times 10^6}{1.729 \times 10^6 \text{BTU/hr.}}$$

5. <u>Main Feed Pump Turbines</u>

Operates on main steam, exhausting to 15 psig, parallel to DLG-6 main feed pump condition line.

h_1 = 1534

s_1 = 1.672

h_9 = 13.76

s_9 = 1.916

Δh = 158

G_b = 97,500 lb./hr. assumed

η_p = 0.50

Feed pump turbine, $G_9 = \dfrac{V\Delta pG_b}{\eta_p \Delta h}$

$$G_9 \cong \frac{.01696 \times 1000 \times 97,500 \times 144}{.50 \times 158 \times 778} = 3.87 \times 10^3 \text{lbs./hr.}$$

6. Evaporators

$G_{ev} \Delta h_{ev} = 5.024 \times 10^6$ const. (DLG-6)

$h_7 = 128$ BTU/lb. (condensate at $160^{\circ}F$)

$$h_6 = \frac{G_5 h_5 + G_9 h_9}{G_5 + G_9} = \frac{3.87 \times 1376 + 3.6 \times 1229 + 4.8 \times 1226}{3.87 + 3.6 + 4.8}$$

$h_6 = 1276$

$\Delta h = 1276 - 128 = 1148$

$G_6 = 4376$ lb./hr.

To compute I evaporators:

$$I_{ev} = G_{ev} [h_6 - h_7 + T_o(s_6 - s_7)]$$

$$= 4376 [1148 - 535 (1.840 - .178)]$$

$I_{ev} = 1.24 \times 10^6$ BTU/hr.

7. Deaerating Feed Heater

Augmenting steam must be of flow rate such that $T_4 = 250^{\circ}F$.

Augmenting steam is a) throttled to 150 psig.

b) desuperheated to $400^{\circ}F$

c) throttled to 15 psig.

From Mollier chart [18];

$h_{aug} = 1219$,

$s_{aug} = 1.772$.

Flow from drains is 6010 lb./hr. const. at $200^{\circ}F$.

Heat balance on D.A. tank:

$$(G_5 + G_9 - G_6) h_6 + G_6 h_7 + G_8 h_8 + G_3 h_3 + G_{aug} h_{aug} + G_{const} \times h_{const}$$

$$= [G_5 + G_9 + G_8 + G_3 + G_{aug} + G_{const}] h_4$$

$$G_{aug} = \frac{(G_5+G_9+G_8+G_3+G_{const})h_4 - G_3h_3 - G_8h_8 - G_6h_7 - G_{const}h_{const} - (G_5+G_9-G_6)h_6}{h_{aug} - h_4}$$

$$= \frac{(8.40+3.87+8.15+73.2+6.01)218.5 - 73.2\cdot 59.7 - 8.15\cdot 134 - 6.01 \times 168 - 4.38\cdot 128 - (8.40+3.87-4.38)1276}{1219 - 218.5}$$

$$G_{aug} = 4.87 \times 10^3 \text{ lbs./hr.}$$

$$G_4 = G_5 + G_9 + G_8 + G_3 + G_{aug} + G_{const} = 104{,}690 \text{ lbs./hr.}$$

$$\frac{I_{da}}{T_o} = G_4s_4 - G_9s_9 - G_5s_5 + G_6s_5 - G_7s_7 - G_8s_8 - G_3s_3 - G_{aug}s_{aug} - G_{const}s_{const}$$

Tabulating

$G_{const}s_{const}$	=	6.01 x .294	=	1.77
G_9s_9	=	3.87 x 1.916	=	7.41
G_5s_5	=	8.40 x 1.786	=	15.00
G_7s_7	=	4.38 x .178	=	.78
G_8s_8	=	8.15 x.2414	=	1.96
G_3s_3	=	73.2 x 0.0999	=	7.32
$G_{aug}s_{aug}$	=	4.87 x 1.772	=	8.63
				42.87
G_4s_4	=	104.69 x 0.3675	=	38.47
G_6s_5	=	4.38 x 1.840	=	8.06
				46.53

46.53 - 42.87 = 3.66

$I_{da} = 3.66 \times 10^3 \times 535 = 1.96 \times 10^6$ BTU/hr.

Additional I for aug. steam, $.29 \times 10^6$ BTU/hr.
(from next section)

$I_{da(tot)} = 2.25 \times 10^6$ BTU/hr.

8. <u>Galley, Laundry, and Hot Water; Fuel Oil Heaters</u>

These functions utilize 400°F externally desuperheated steam:

a) Galley, etc., 2500 lb./hr., throttled to 50 psig,

b) F.O. Heaters 300 lb./hr., throttled to 150 psig,

from desuperheated steam line with augmenting steam.

External desuperheater mixes boiler feedwater at 250°F with 150 psig desuperheated steam. External desuperheater not shown on Figure D-I.

Let p = steam from internal desuperheater,

q = feedwater to external desuperheater.

$$G_q = (G_{aug} + 2800)\left(\frac{h_p - h_{aug}}{h_{aug} - h_p}\right)$$

$$G_q = 7670\left(\frac{1253 - 1219}{1219 - 218.5}\right)$$

$$G_q = 261 \text{ lbs./hr.}$$

$$I(\text{external desuperheater}) = G_q T_o \Delta S_q + 7670\, T_o \Delta S_{DSH}$$

$$I\ e.d. = 261 \times 535 \times (.363 - 1.600) + 7670 \times 535(1.600 - 1.448)$$

$$I\ e.d. = 450{,}400 \text{ BTU/hr.}$$

I e.d. is charged to the various services by proportion :

Augmenting steam,

$$I_{aug} = \frac{4870}{7670} \times .450 \times 10^6 = .286 \times 10^6 \text{ BTU/hr.}$$

This must be added to D.A. tank irreversibility. Other services have a constant irreversibility after external desuperheater plus a portion of the external desuperheater irreversibility.

$$I\ \text{Galley, etc.} \quad \left(.71 + \frac{2500}{7670} \times .450\right) 10^6 = .86 \times 10^6 \text{ BTU/hr.}$$

$$I\ \text{F.O. Heater} \quad \left(.01 + \frac{300}{7670} \times .450\right) 10^6 = .03 \times 10^6 \text{ BTU/hr.}$$

9. <u>Main Feed Pump</u>

$G_p = G_4 = 104,690$ lbs./hr. (from Sec. D-7)

I_{net} = availability loss in turbine - availability gain in feedwater

$$= 3.87 \times \frac{104.6}{97.5} \times [158 - 535\ (1.672 - 1.916)]$$

$$- 104,600 \times 3.19 = 0.87 \times 10^6 \text{ BTU/hr.}$$

Weight of main feed pump:

$$W_{mfp} = (.0249 \times 1000 - 3.25)(104,600)^{5/8}$$

= 29,900 lbs. all installed pumps.

10. <u>Boiler</u>

Available energy supplied by boiler = total available energy consumed by plant.

Summary of available energy disposition:

Main feed pump	0.87×10^6 BTU/hr.	
Turbine and condenser irreversibility	14.95	
Evaporators	1.24	
Deaerator	2.25	
Turbogenerators	5.39	
Galley, etc.	0.86	
Fuel oil heaters	.03	
Forced draft blowers and pumps	1.73	
Turbine work	33.10	
	60.42	
Constant miscellaneous	1.05	(From DLG-6
$G_b \Delta b_b$ =	61.47	availability balance)

Since we know boiler steam conditions and flow, $G_b \Delta b_b$ can be checked:

$$h_1 = 1533.2$$

$$s_1 = 1.6711$$

$$h_{1a} = 1253$$

$$s_{1a} = 1.448$$

$$h_4' = 218.5$$

$$s_4' = 0.3675$$

$$\Delta b_{4'-1} = 617$$

$$\Delta b_{4'-1a} = 578$$

$$G_b \Delta b_b = G_1 \Delta b_{4'-1} + G_{1a} \Delta b_{4'-1a}$$

$$G_1 = G_3 + G_9 + G_8$$

$$G_{1a} = G_4' - G_1$$

$$G_b \Delta b = 85{,}220 \times 616 + 19{,}470 \times 457$$

$$= 52.58 + 8.89 = 61.47$$

Boiler Weight

$$W_b = (.354 + .00236 \cdot 1000)\ 104{,}690 + 75.2\ (104{,}690)^{2/3} \qquad \text{(A-1)}$$

$$= 444{,}000 \text{ lbs. (wgt. of 4 installed boilers.)}$$

Heat added efficiency

$$\eta_A = \frac{\Delta b_b}{\Delta h_b}$$

$$\eta_A = \frac{61.47 \times 10^6}{85{,}220 \times 1314.7 + 19{,}470 \times 1035}$$

$$\eta_A = 0.463$$

Overall thermal efficiency

$$\eta_{th} = \eta_A \eta_B \quad e_m \quad \frac{G_t \Delta h_t}{G_b \Delta b_b}$$

$$\eta_{th} = 0.463 \times 0.88 \times 0.931 \times \frac{33.10}{61.47} = \frac{12.55}{61.47}$$

$$\eta_{th} = 0.204$$

Fuel Weights

$$\frac{W_F}{1000 \text{ Mi}} = \frac{50 \text{ hrs}(@\ 20 \text{ kts}) \times 12{,}100 \text{ SHP} \times 2545 \frac{\text{BTU}}{\text{SHP-hr}}}{18{,}525 \text{ BTU/lb.} \times \eta_{th}}$$

$$\frac{W_F}{1000 \text{ Mi}} = 408{,}000 \text{ lbs.}$$

APPENDIX C

EFFECT OF

CONDENSATE TEMPERATURE ON TOTAL VARIABLE WEIGHT

Our purpose here is to show that leaving loss can be optimized on the basis of minimum weight of turbine and condense alone, and that the remainder of the variable weight is effectively independent of condensate temperature.

Augmenting steam is used in sufficient quantity to raise condensate temperature to 250°F in the D.A. tank.

$$G_{aug} = \frac{G_{cond}(218.5 - h_{cond})}{h_{aug} - 218.5} \qquad \text{(C-1)}$$

For the example of Appendix B,

$$G_3 = \frac{73.2}{99.8} G_{cond.} \qquad \text{(flow to D.A. tank)}$$

$$\Delta t_{cond.} \cong 0.73 \; \Delta t_3 \; .$$

Using the numbers of Appendix B in (C-1),

$$4.87\ (1219 - 218) = 99.8\ (218.5 - h_{cond})$$

$$h_{cond.} = 169.8$$

$$s_{cond.} = .297$$

$$\frac{I_{aug}}{T_o} = 104.7 \times .368 - 99.8 \times .297 - 4.87 \times 1.448$$

$$I_{aug} = \frac{1.02}{4870} G_{aug} \times 10^6$$

$$\Delta G_{aug} = \frac{G_{cond} \; \Delta t_{cond}}{1001}$$

$$\Delta I_{aug} = \frac{99{,}800 \times 0.73 \Delta t_3}{1001} \times \frac{1.02}{4870} \times 10^6$$

$$\Delta I_{aug} = 14,900 \; \Delta t_3 \text{ BTU/hr.}$$

$$\frac{\Delta W_F}{1000 \text{ mi}} = 6,630 \times \Delta I_{aug} \times 10^{-6} \qquad (9)$$

Assuming a 5000 mile mission and the condensate temperature variation of the 1000 psia, 73,200 lb./hr. family, we tabulate:

$$\Delta W_F = 495 \, \Delta t_3$$

Leaving loss	1	2	3	4	5	7
Δt_3	+1.2	+0.6	0.0	-1.0	-1.7	-3.3
ΔW_F, lbs.	-594	-297	0	495	+843	+1,633
ΔW_{t+c}, lbs.	25,400	+3,200	0	14,300	28,700	99,600
$\frac{\Delta W_F}{\Delta W_{t+c}}$	.023	.093	-	.035	.029	.016

It is evident that the effect of condensate temperature on fuel weight is negligible compared with the change in weight of condenser and turbine combination with leaving loss.

APPENDIX D

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