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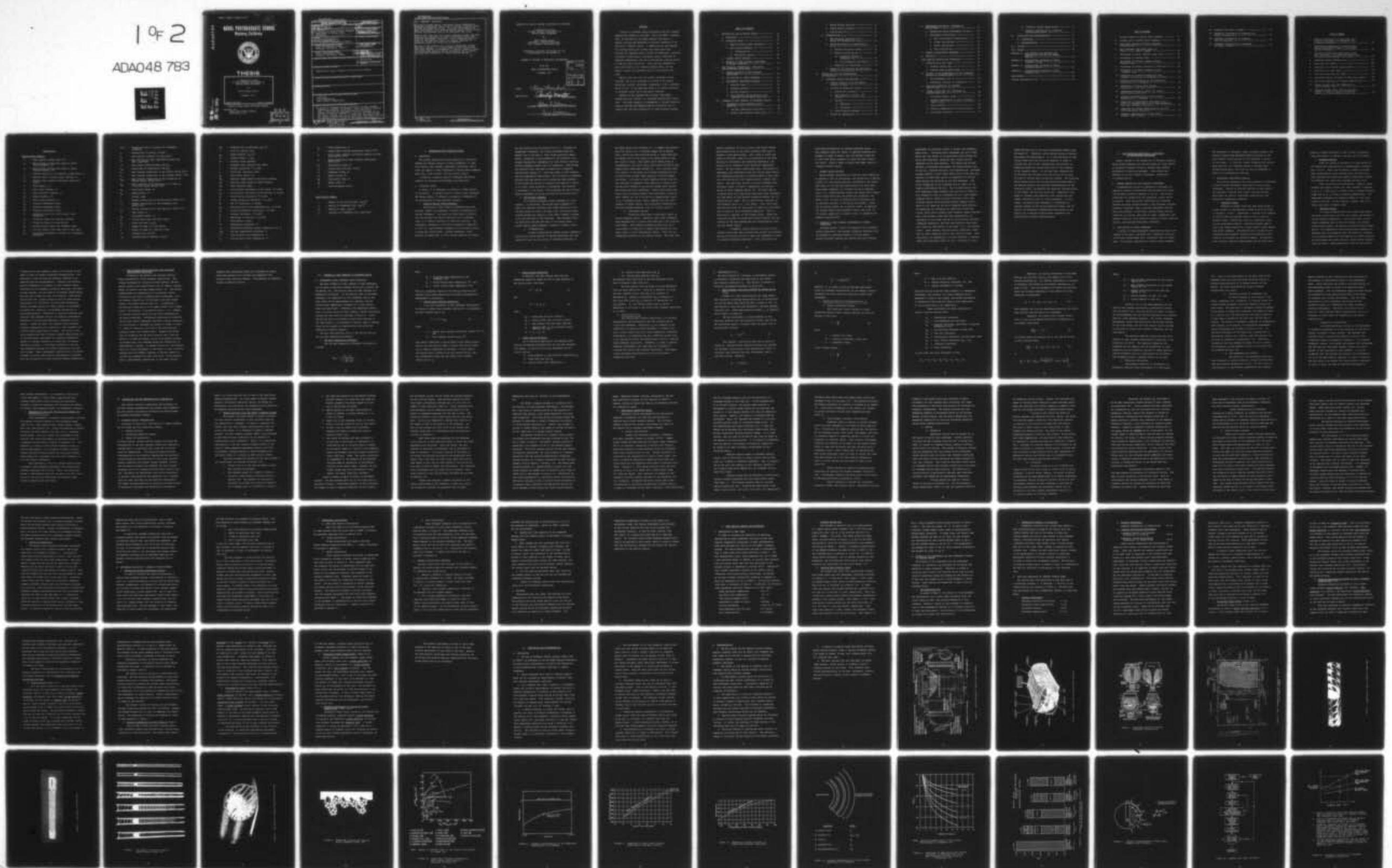

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NAVAL POSTGRADUATE SCHOOL
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THESIS

A FEASIBILITY STUDY
OF HEAT TRANSFER IMPROVEMENT IN
MARINE STEAM CONDENSERS

by

Harry Thomas Search

December 1977

Thesis Advisor: Paul J. Marto

Prepared for:
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Results show that with the present condenser volume, the heat load can be increased by as much as 50 percent using heat transfer improvement techniques. From a different point of view, at the same heat load, a 40 percent reduction in condenser weight and volume may be feasible.

Several of the proposed heat transfer improvement schemes may lead, however, to increased pumping power and/or cost. Continued research is recommended in several promising areas to provide more adequate design information and to improve the long term reliability of these proposed schemes.

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A Feasibility Study of Heat Transfer Improvement in Marine Steam Condensers

by

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Nomenclature

English Letter Symbols

A	-	Heat transfer surface area, ft^2.
A_i	-	Heat transfer surface area based on tubing inside diameter, ft^2.
A_o	-	Heat transfer surface area based on tubing outside diameter, ft^2.
b	-	Interfin spacing in an internally finned tube, in.
c_p	-	Specific heat of cooling water, BTU/lb_m °F.
C_n	-	External heat transfer coefficient correction factor.
D	-	Tube diameter, ft.
D_i	-	Tube inside diameter, ft.
D_e	-	Tube equivalent diameter, ft.
D_H	-	Header diameter, ft.
D_o	-	Tube outside diameter, ft.
E_i	-	Internal enhancement factor
E_o	-	External enhancement factor
ΔE	-	Potential difference, volts
F_1	-	Correction factor for cooling water inlet temperature
F_2	-	Correction factor for tube wall material
F_3	-	Correction factor for tube fouling
f_i	-	Friction factor inside the condenser tubes
f_o	-	Friction factor on the shell side of the tubes
F_d	-	Tube spacing parameter to account for condensate inundation

$F(n)$ - Correction factor to account for condensate inundation

g - Acceleration of gravity, ft/sec^2.

g_c - Gravitational constant, ft $lb_m/lb_f sec^2$

G_v - Mass velocity of vapor based upon minimum flow area, $lb_m/ft^2 sec$.

Δh - Heat removal from the steam, BTU/lb_m

h_{fg} - Latent heat of vaporization, BTU/lb_m

h_i - Heat transfer coefficient on the inside, BTU/hr $ft^2 °F$

h_o - Heat transfer coefficient on the outside, BTU/hr $ft^2 °F$

h_{oN} - Heat transfer coefficient on the outside due to Nusselt theory, BTU/hr $ft^2 °F$

$\bar{h}_{oN}$ - Mean condensate film coefficient for n tubes in a vertical column, BTU/hr $ft^2 °F$

H_B - Tube bundle height, ft.

H_H - Header height, ft.

I - Current, amps

k - Thermal conductivity of cooling water, BTU/hr ft°F

k_f - Thermal conductivity of the condensate film, BTU/hr ft°F

k_w - Thermal conductivity of the tube wall, BTU/hr ft°F

L - Tube length, ft.

L_B - Tube bundle length, ft.

$\dot{m}_c$ - cooling water mass flow rate, lb_m/hr.

$\dot{m}_s$ - Steam flow rate, lb_m/hr

N - Number of tubes in a tube bundle

n - Number of tubes in a vertical column

Δp - Pressure drop, lb_f/ft^2

Δp_i - Pressure drop on tubeside, lb_f/ft^2

Δp_o - Pressure drop on shellside, lb_f/ft^2

p - Pitch of internal fins

p_s - Steam pressure, lb_f/in^2

Pr - Prandtl number, $\mu c_p/k$

PW - Flooding factor exponent

$\dot{Q}$ - Condenser heat transfer rate, BTU/hr

R - Thermal resistance, °F hr/BTU

R - Electrical resistance, ohms

R_B - Tube bundle radius, ft.

Re_e - Reynolds number based on equivalent diameter

Re_i - Reynolds number based on inside diameter

Re_v - Vapor Reynolds number

R_{fi} - Fouling/scale resistance on the inside, °F hr/BTU

R_{fo} - Fouling/scale resistance on the outside, °F hr/BTU

R_f - Total fouling resistance, °F hr/BTU

R_i - Inside convective resistance, °F hr/BTU

R_j - The j^{th} resistance, °F hr/BTU

R_{nc} - Resistance due to non-condensable gas, °F hr/BTU

R_o - Outside condensation resistance, °F hr/BTU

R_p - Promoter resistance, °F hr/BTU

R_w - Resistance of tube wall, °F hr/BTU

S - Tube spacing, inches

ΔT - Temperature difference, °F

ΔT_f - Temperature difference across condensate film, °F

ΔT_{LM} - Log mean temperature difference, °F

T_i - Cooling water inlet temperature, °F

T_o - Cooling water outlet temperature, °F

T_s	-	Steam temperature, °F
U	-	Overall heat transfer coefficient, BTU/hr ft^2°F
U_o	-	Overall heat transfer coefficient based on outside area, BTU/hr ft^2°F
U_v	-	Uncorrected overall heat transfer coefficient, BTU/hr ft^2°F
V_B	-	Tube bundle volume, ft
v_c	-	Cooling water velocity, ft/sec
V_c	-	Condenser volume, ft.
V_H	-	Header volume, ft.
W_B	-	Tube bundle width, ft
W_H	-	Header width, ft.
Z	-	Tube enhancement factor

Greek Letter Symbols

ρ	-	Density of the cooling water, lb_m/ft^3
ρ_f	-	Density of condensate film, lb_m/ft^3
ρ_v	-	Density of vapor, lb_m/ft^3
μ_f	-	Viscosity of condensate film, lb_m/ft-sec

I. INTRODUCTION AND HISTORICAL NOTES

A. OBJECTIVES

The primary objectives of this study are to review the methods for thermal analysis of steam condensers, to identify relevant heat transfer improvement techniques, and to study the impact of these techniques on marine steam condenser heat load, volume, weight, cost and overall design.

In addition, promising areas of condenser exploratory research and engineering development are identified.

B. HISTORICAL NOTES

To begin, it is necessary to establish a common ground for discussion. A capsule history of marine steam condensers and design methods is therefore utilized as a framework for the introduction of some important concepts.

1. Types of Marine Steam Condensers

Historically, three types of condensers have been utilized, namely jet condensers, barometric condensers and surface condensers. Sea water has always been utilized as the cooling water for marine propulsion plant condensers since it is readily available in sufficient quantity. Jet condensers, successors to the device patented by James Watt in 1769 [1], and barometric condensers utilize direct mixing of steam and cooling water. Surface condensers, first patented by Samuel Hall in 1834, provide separate flow paths for the working fluid and cooling fluid [1]. Although not immediately successful, the surface condenser came into general use in the 1860's. Early steam propulsion vessel owners, interested in the economics of the situation, discovered reduced fuel consumption and fuel costs by utilizing surface condensers rather than jet condensers. This economy was realized since less heat is wasted with a separate cooling water system [1]. Since the advent of high pressure and high temperature steam plants, the utilization of separate working fluid and cooling fluid systems has been necessary to facilitate the control of working fluid quality or purity. In so doing, scale formation is eliminated, and corrosion is minimized in the boiler. This precludes the use of direct mixing condensers utilizing sea water as the cooling fluid.

2. <u>The Surface Condenser</u>

In a surface condenser, steam condenses on a cold surface. In particular, the marine steam condenser is a shell and tube heat exchanger with steam condensing on the outside surface of tubes, the shell side, and cooling water flowing through the interior of the tubes, commonly referred to as the tube side of the condenser. Heat transfer occurs by convection from the tube wall to the cooling water. A typical marine steam condenser is shown in Figures 1 and 2.

a. "Condenseritis"

An early problem which plagued surface condensers, particularly during periods of reduced maintenance and long periods at sea, was the failure of condenser tubes due to sea water erosion and corrosion [1]. A common tube material was 70-30 brass (i.e., 70 percent copper and 30 percent zinc); and at one time during World War I, these tubes had an average life of only three to six months before a leak was likely to occur. This defect which became known as "condenseritis" was never really cured until the 1930's when copper-nickel tubes were adopted [1]. Zinc is highly reactive (anodic) in sea water and corrodes preferentially due to electrolysis to many other materials including copper and nickel. Zinc bars or plates commonly called zincs are now installed in condenser water boxes and on ship's hulls and sea water intakes as sacrificial anodes which corrode and require periodic replacement. These zincs protect the ship's hull and piping systems from electrolytic corrosion. Naval vessels now use 70-30 and 90-10 copper-nickel tubes (i.e., 70 percent nickel) which, together with the other elements of proper design, have eliminated "condenseritis" and have extended condenser life up to twenty years [1].

b. Tube Bundle Designs

Volume has always been a significant factor in marine condenser design. Therefore, many older condensers were designed with a circular tube sheet and a full circular tube bundle. This geometry produced minimum size for a given number of tubes due to compact tube spacing and, for small units, a fairly satisfactory design. There are two undesirable features of this design however. The close tube spacing throughout the full circular tube bundle creates more pressure drop on the steam condensing side than is created in more open bundles. Also, since no steam flow lanes are provided, steam flow is restricted in the lower sections of the bundle; and condensate depression, the sub-cooling of condensate below saturation temperature, occurs. For the overall steam power cycle, sub-cooling is unnecessary and undesirable since any cooling below saturation temperature requires an increased amount of heat addition from the boiler, thus lowering overall thermal efficiency. The tube sheet layouts of various condenser manufacturers, shown in Figure 3, demonstrate the artistic nature of steam condenser design [2]. The basic idea is to design steam flow lanes which distribute steam throughout the tube bundle so that maximum condensing benefit will be gained from all tubes. This is particularly important at high steaming rates (i.e., full power). Also, good steam distribution allows for the mixing of steam with the condensate in order to avoid condensate depression. This is particularly important at low steaming rates. Steam flow lanes are provided for distributing steam radially into the interior of the tube bundles and also longitudinally along the full bundle length.

In summary, proper design will provide fairly uniform and minimum vapor pressure drop through the condenser and allow sufficient mixing of vapor and condensate in order to minimize condensate depression. Also, provisions for good steam distribution for maximum condensing benefit at high steaming rates will result in improved condenser performance in terms of weight, cost and perhaps volume due to less tubes being necessary to handle the heat load at full power. Figure 4 shows a typical tube sheet layout of a modern aircraft carrier.

C. CURRENT DESIGN PRACTICE

Marine surface condensers are noted for their simplicity of operation, ease of maintenance, and reliability in service. Current U.S. Navy design practice, documented in the design data sheet [3] and military specification for Naval shipboard steam condensers [4] dictates the use of seamless, plain (i.e., smooth external surface and smooth bore) copper-nickel tubing which has good mechanical properties (See Table 1 [5]), is toxic to adhering marine organisms, and is corrosion resistant in the marine environment. Alloy 706 (90 percent copper and 10 percent nickel) is specified for surface vessel steam condenser tubing, and alloy 715 (70 percent copper and 30 percent nickel) is specified for submarine applications.

D. BENEFITS OF HEAT TRANSFER IMPROVEMENT IN MARINE CONDENSERS

Although surface vessels and submarines have different design constraints, (For example, submarine condensers must be designed to withstand pressure at diving depths.) certain universal benefits are derived from heat transfer improvement in propulsion plants in general and condensers specifically. Increasing shaft horsepower per unit volume and/or per unit weight of the propulsion plant through the use of new technology, generates some flexibility and tradeoffs for the ship designer. The results may be made manifest in decreased displacement and wetted surface area of the hull, improved machinery room accessibility, increased storage capacity, a more robust power plant on the same size platform, or some other option or combination of options. These options are specified by the ship designers. The actual dimensions of the condenser have an impact on the vessel's performance and cost. This is particularly true with submarines where condenser size can have an impact on hull diameter so that an increased cost for a more compact condenser may be offset by overall savings in hull size. Although current design practice ensures thermal performance at full power by the use of generous design margins, the penalties for overdesign are additional weight and volume, which lead to greater than necessary vessel displacement, and machinery rooms with poor accessibility [6]. Overdesign at full power results in vacuum control problems, condensate depression, and high steam entrance velocities with resulting tube erosion at low power (i.e., low steaming rates). Naval vessels, and particularly combatants, spend most of their operating time at less than full power. Also, the penalty for underdesign is a small reduction in maximum power which manifests itself as only a fraction of a knot speed reduction due to the cubic relationship between power and speed [6]. Therefore, narrow design margins on thermal performance are advantageous. It is also desirable to keep cooling water flow rate as low as possible in order to minimize the size of hull penetrations, particularly on submarines where the cooling water side of the condenser is the pressure vessel. At the same time, pressure drop in the tubes must be kept within the limits of the capabilities of cooling water circulating systems. Any finalized design should result from an optimization process based on the desired physical and operating characteristics of the propulsion plant, the physical and operating characteristics of the vessel, and the life cycle cost. Essentially this means to integrate the propulsion plant system with the vessel, considering cost as a major parameter. In this report, the marine steam condenser is considered as a separate element of the marine propulsion system, and all results are presented to demonstrate some aspects of the feasibility of designing marine steam condensers with improved heat transfer characteristics.

II. HEAT TRANSFER AUGMENTATION - MOTIVATION FOR ENGINEERING DEVELOPMENT

Recent interest in the feasibility of designing improved marine steam condensers has been motivated by a wide variety of successful research efforts to enhance heat transfer performance in operating equipment. Such efforts are referred to as heat transfer enhancement, augmentation or intensification.

A. GENERAL RESEARCH ON HEAT TRANSFER IMPROVEMENT

A comprehensive review of augmentation techniques for improving convective single-phase and two-phase heat transfer has been provided by Bergles [7,8,9,10]. The various techniques are grouped into two categories, either active or passive. They are further subdivided into seventeen methods including treated surfaces, rough surfaces, extended surfaces, "enhanced" tubes, displaced enhancement devices, swirl flow devices, coiled tubes, surface tension devices, additives for liquids, additives for gases, mechanical aids, surface vibration, fluid vibration, electric fields, injection, suction, and compound/combination techniques.

B. APPLICATIONS TO STEAM CONDENSERS

A variety of these enhancement techniques has been investigated on the shell (two-phase flow, condensing) side and on the tube (single-phase flow, convective) side of condensers. For specific application to marine steam condensers, the categories of "enhanced" tubes, extended surfaces, and treated surfaces have received extensive coverage in the heat transfer literature and will be addressed in detail in this report. In addition, simply increasing the cooling water velocity in a condenser serves to increase heat transfer performance, and this will also be addressed in detail as an augmentation technique.

C. PERFORMANCE IMPROVEMENT EXAMPLES

This section summarizes the various methods or techniques of heat transfer enhancement applicable to marine steam condensers. Results of previous feasibility type studies are included to provide some insight into the relative magnitudes and types of improvement predicted for each particular augmentation technique.

1. <u>"Enhanced" Tubes</u>

Enhanced or shaped tubes have been manufactured in a wide variety of configurations, three of which are shown in Figures 5, 6 and 7. References 11 through 20 are reports of heat transfer performance tests on various types of enhanced tubes. Withers and Young [21] conducted a type of cost performance analysis, comparing enhanced tube to plain tube design results for a large electric power generating plant condenser. The enhanced tube utilized for the comparison was Korodense, manufactured by Wolverine Tube Division of Universal Oil Products Company. The results, with virtually identical pumping power and the same tubing material, showed reductions in shell volume (20 percent), total tube weight (27 percent) and tube cost (15 percent).

2. Extended Surfaces

Inner fin tubing is available in a wide variety of fin heights and configurations as demonstrated in Figures 8 and 9 [22,23]. Royal [24] discusses a variety of single-phase investigations using internal fins, and points out that seventeen such tubes were tested by Watkinson, Miletti and Tarassoff [25] using both straight and axial fins.

Their conclusions are encouraging. Based on inside diameter and nominal inside area, heat transfer is enhanced by up to 2.7 times that of a smooth tube at constant Reynolds number and up to 1.8 times at constant pumping power. The enhancement decreases with Reynolds numbers in the range 10,000 to 150,000. Friction factors, however, were up to three times those of smooth tubes.

3. Treated Surfaces

If liquid condensate wets the outside tube surface and forms a continuous film over the surface, the process is called filmwise condensation. On the other hand, if the liquid condensate does not wet the surface, but instead forms discrete drops, the process is called dropwise condensation. All commercially manufactured steam condensers are designed under the assumption that the steam will condense in the filmwise mode since this mode can be readily analyzed for performance and occurs naturally unless special preparations are taken. These preparations take the form either of additives to the condensing vapor or of coatings to the tubes in order to promote nonwetting characteristics. The additives or tube coatings are generally referred to as promoters and must be hydrophobic. Generally the dropwise mode of condensation is superior in heat transfer characteristics; but the promoter can add an additional resistance to heat transfer, and the question of promoter effectiveness and long term reliability must be considered. Noble metals such as gold [26], organic materials such as waxes derived from plants [27], and teflon [28] have been successfully tested, although results are widely varying, as discussed by Graham [29], Figure 10, and promoter reliability is highly questionable. References 26 through 46 describe some pertinent heat transfer results with dropwise condensation. Reference 47 provides some additional references on this subject. Graham and Aerni [35] compared teflon promoted dropwise condensation with filmwise on a main condenser for a DLG-10 class guided missile Frigate. The results, with equal pressure drop and 25 percent reduced pumping power due to the performance improvement for dropwise condensation, showed a 13 percent reduction in volume and a 16 percent reduction in weight. A 20 percent reduction in cost was also estimated although the promoter application cost was not included. These improvements resulted from a 30 percent increase in overall heat transfer performance as predicted with the utilization of teflon promoted dropwise condensation.

4. High Strength Tubing Materials with Increased Cooling Water Velocities

Titanium is the primary high strength material being considered for steam condenser applications. The primary advantages of titanium are high strength (200-300 percent greater than copper-nickel) and low density (approximately 50 percent less than copper-nickel) [5]. The primary disadvantages are susceptibility to biological fouling (Titanium is not toxic to adhering marine organisms.) [48], low thermal conductivity (45-65 percent less than copper-nickel), and higher material cost [5]. A comparison of the physical properties of 70-30 copper-nickel, 90-10 copper-nickel, and titanium, is provided as Table 1, [5]. Adamson [5] studied titanium tubing for both steam condensers and sea water evaporators. Performance comparisons on submarine and aircraft carrier condensers resulted in weight reductions of 31-36 percent of condenser dry weight, as shown in Table 2. Costs of titanium at the time of the study were considerably higher than copper-nickel. Adamson also showed that by increasing the sea water velocity from 9 feet per second to 12 feet per second, giving an 82 percent increase in pressure drop, the condenser weight was reduced only by approximately 2 percent. As Adamson points out, the pressure drop is a critical parameter in the design of scoop injection systems which are commonly employed on surface vessels to provide the necessary sea water flow rates. Since pressure drop is approximately proportional to sea water velocity squared, some uncertainty exists as to whether the higher velocities possible with titanium are compatible with existing scoop injection designs. This question is addressed further in Section IV-A-2-d.

III. ELEMENTS OF HEAT TRANSFER IN CONDENSER DESIGN

A. FUNDAMENTAL STEAM CONDENSER DESIGN EQUATIONS

The basic elements of heat transfer in heat exchangers can be found in many heat transfer texts such as Holman [49] and Kreith [50]. The driving force for heat transfer is a temperature difference. In the specific case of the steam condenser, the temperature on the condensing side of the tube bundle can be approximated as a constant, and the cooling water temperature increases as it flows axially along the tube bundle. Therefore, the temperature difference, or driving force for heat transfer, varies continuously through the tube bundle as sketched in Figure 11. Since the driving force is different at each axial position in the tube bundle, it is necessary to obtain a mean or average value for the purpose of characterizing the system and conducting a thermal analysis.

One such characterizing value of the driving force is the log mean temperature difference.

1. Log Mean Temperature Difference

The log mean temperature difference is defined as follows:

$$\Delta T_{LM} = \frac{T_o - T_i}{\ln \left(\frac{T_s - T_i}{T_s - T_o} \right)} \qquad (1)$$

where

T_s = saturated steam temperature in the condenser (°F),

T_i = inlet cooling water temperature (°F), and

T_o = outlet cooling water temperature (°F).

This is a convenient measure of the driving force for heat transfer since these temperatures are readily obtainable by measurement or calculation.

2. Overall Heat Transfer Coefficient

It is also convenient in heat exchanger calculations to define an overall heat transfer coefficient U by expressing the heat transfer rate $\dot{Q}$ as:

$$\dot{Q} = U A \Delta T_{LM} \quad (2)$$

wherè

U = overall heat transfer coefficient (BTU/hr ft^2 °F), and

A = heat transfer surface area (ft^2).

The overall coefficient U may be based on any chosen surface area A, and care must be taken to ensure that an area basis for an overall coefficient is provided. In this thesis, the surface area is chosen to be the outside area A_o, and the corresponding valve for the overall heat transfer coefficient is U_o.

3. Energy Balance Equations

In addition, the heat removal rate from the condensing steam must equal the rate of heat addition to the cooling water, such that:

$$\dot{Q} = \dot{m}_s \Delta h \tag{3}$$

and

$$\dot{Q} = \dot{m}_c c_p \Delta T \tag{4}$$

where

$\dot{m}_s$ = steam mass flow rate (lbm/hr),

$\dot{m}_c$ = cooling water mass flow rate (lbm/hr),

Δh = heat removal from the steam (BTU/lbm),

c_p = specific heat of the circulating water (BTU/lbm °F), and

ΔT = $T_o - T_i$ (°F).

4. Simple Design Procedure

For a particular application, the designer must specify the desired characteristics for the heat exchanger. For example, a typical set of input parameters may be given, such as:

(1) Steam pressure P_s and saturation temperature T_s

(2) Steam mass flow rate $\dot{m}_s$

(3) Cooling water inlet temperature T_i

(4) Cooling water mass flow rate $\dot{m}_c$

(5) Cooling water specific heat c_p.

The designer would then have to find the required surface area to transfer a heat load of $\dot{Q}$.

The heat removal from the steam Δh can be determined from the steam tables [51]. Using Δh in equation (3) gives the heat transfer rate $\dot{Q}$, and using $\dot{Q}$ in equation (4) determines T_o. Knowing T_o determines ΔT_{LM} in equation (1), and using both $\dot{Q}$ and ΔT_{LM} in equation (2) determines the U_oA_o product. So, the outside surface area A_o can be determined if the value of the overall heat transfer coefficient U_o is known.

5. Significance of U_o

The overall heat transfer coefficient U_o is uniquely determined by the configuration and flow characteristics of the heat exchanger. Physically, U_o is a measure of the capability of a particular heat exchanger to transfer heat over a given surface area for a given temperature difference. Consequently, U_o determines how much condensing surface area A_o is required to handle the required heat load for a specific steam condenser application. Therefore, in order to improve condenser performance, the focus must be on methods to increase the overall heat transfer coefficient. The augmentation techniques described earlier can be utilized to accomplish this task.

B. DETERMINATION OF U_o

The basic problem of a designer or performance analyst is therefore to determine the magnitude of the overall heat transfer coefficient U_o. This section is devoted to outlining the methods available for doing this.

1. Heat Exchange Institute Method for Determination of U_o

Present U.S. Navy specifications for steam condensers contained in military specification MIL.-C-15430J-ships [4] and the design data sheet DDS 4601-1 issued in 1953 [3] were derived in part from the standards of the Heat Exchange Institute [52]. These specifications present U_o in graphical form as shown in Figure 12.

The key relationship in these documents is the empirical equation that the uncorrected overall heat transfer coefficient equals a constant times the square root of cooling water velocity:

$$U_v = Cv_c^{1/2} . \qquad (5)$$

The constant C varies with tube size as shown in Figure 12. Multiplicative correction factors are provided for different cooling water inlet temperatures, Figure 13; different tube materials and wall thicknesses, Table 3; and tube fouling. Therefore,

$$U_o = F_1F_2F_3U_v , \qquad (6)$$

or

$$U_o = F_1F_2F_3Cv_c^{1/2} \quad . \qquad (7)$$

Equation (7) is simple to use and has been used extensively by condenser manufacturers for the design of marine propulsion and shore electrical generating plant steam condensers.

2. Resistance Method for Determination of U_o

a. Thermal Resistance Concept

The electrical circuit analogy is a well documented method of heat transfer analysis utilizing an analogy to Ohm's Law:

$$I = \frac{\Delta E}{R} \qquad (8)$$

where

I = current flow (amps)

ΔE = potential difference (volts), and

R = resistance (ohms).

In the thermal analogy,

$$\dot{Q} = \frac{\Delta T}{R} \qquad (9)$$

where

$\dot{Q}$ = heat flow rate (BTU/hr),

ΔT = thermal potential difference (°F), and

R = thermal resistance (°F hr/BTU).

Since the driving force for heat transfer is a temperature difference, heat transfer in a steam condenser can be expressed in terms of the thermal resistances encountered in transferring heat from the steam at high temperature to the cooling water at low temperature.

These resistances are shown graphically in Figure 14 and are defined below.

R_o = condensation resistance

R_{nc} = non-condensable gas resistance

R_p = promoter resistance (applicable to dropwise condensation only)

R_{fo} = fouling/scale resistance (steam side)

R_w = tube wall resistance

R_{fi} = fouling/scale resistance (cooling water side)

R_f = total fouling resistance ($R_{fo} + R_{fi}$)

R_i = convection resistance

R_t = total resistance.

In this case, the total resistance is then:

$$R_t = R_o + R_{nc} + R_p + R_{fo} + R_w + R_{fi} + R_i .$$

Generally, the fouling resistances on the steam side R_{fo} and the water side R_{fi} are lumped into a total fouling resistance R_f. For filmwise condensation no promoter is necessary, and therefore the promoter resistance R_p is equal to zero. With the assumption of lumped fouling resistance, and filmwise condensation, the equation for total resistance is simplified:

$$R_t = R_o + R_{nc} + R_w + R_f + R_i . \tag{10}$$

This suggests that five key resistances determine the overall heat transfer characteristics of a condenser.

Therefore, the overall heat transfer coefficient U_o can also be determined through the summation of the thermal resistances.

$$\frac{1}{U_o A_o} = \sum_j R_j . \tag{11}$$

For single tubes, the equation for U_o can then be written in the following forms:

$$\frac{1}{U_o A_o} = R_o + R_{nc} + R_w + R_f + R_i \tag{12}$$

or

$$\frac{1}{U_o} = \frac{1}{h_o} + A_o R_{nc} + \frac{D_o \ln D_o/D_i}{2 k_w} + A_o R_f + \frac{1}{h_i} \frac{A_o}{A_i} \tag{13}$$

where

h_o = heat transfer coefficient on the outside (BTU/hr ft^2 °F)

h_i = heat transfer coefficient on the inside (BTU/hr ft^2 °F)

k_w = thermal conductivity of the tube wall (BTU/hr ft °F)

D_o = outside diameter of tube (ft), and

D_i = inside diameter of tube (ft) .

Equation (13) is the same expression for the overall heat transfer coefficient U_o developed by Holman [49] and Kreith [50], with the addition of terms to account for non-condensable gas and fouling resistance as is normally done for steam condenser analysis. For overall condenser performance, single tube calculations can be made throughout the tube bundle, and the local results can be utilized to calculate an overall heat transfer coefficient for the tube bundle.

The generally employed correlations for local analysis of heat transfer performance are described in the literature [15,49,50]. The equations themselves are algebraic and not complex, but the calculation of local properties and the sequential solution of these equations through successive rows of a tube bundle are incredibly time consuming unless a computer program is utilized.

b. Modelling Difficulties

Considerable difficulty is encountered in accurately modelling local performance in a tube bundle

[15]. This is true particularly on the shell side of the condenser where the steam and water two-phase mixture is changing characteristics as it passes through the tube bundle. The flow characteristics of each condenser configuration are unique.

(1) Inundation effects of condensate rain

As steam condenses on the outside of the tubes, condensate rain, inundation, or flooding occurs with greater degree proceeding vertically downward through the tube bundle. The magnitude of condensate inundation depends on the amount of steam being condensed, the number of tubes in a vertical row, the tube spacing, the relative position of the tubes in successive vertical rows, and the use of baffles. The effect of condensate inundation on condensation heat transfer is the subject of considerable disagreement among investigators as is shown in Figure 15 [15]. An excellent discussion of the theory and a presentation of experimental results for a specific tube configuration can be found in Eissenberg's dissertation [15]. Additional information has been gathered by Young [12,13] to account for inundation in tube bundles, as well as the recent work of Catchpole and Drew [19].

(2) Non-condensable gas effects

The effects of non-condensable gases have been discussed by various authors in conjunction with dropwise and filmwise condensation [15,29]. In fact, much of the dispersion in the dropwise condensation heat transfer results reported in the literature has been attributed to the presence of undetermined amounts of non-condensable gases. Steam velocities and normally low concentrations of non-condensable gases are found in an operating marine steam condenser. The non-condensable gases are swept through the condenser with the steam flow and have little effect on condenser heat transfer performance. Very low steam velocities will allow stagnation and buildup of non-condensable gases in a condenser with either filmwise or dropwise condensation. Therefore, the location of air vents is a very important design consideration, and it is necessary to avoid steam flow stagnation in any design configuration.

c. Comparison of Resistances

As stated previously, the key to the improvement of condenser performance is in the reduction of the overall thermal resistance to heat transfer. The designer must therefore seek to minimize the effects of fouling, non-condensable gases and condensate inundation. In a standard condenser, however, the internal and external film resistances and the tube wall resistance together account for over 90 percent of the resistance to heat transfer. It is also noteworthy that the internal and external film resistances can be of the same order of magnitude, as seen in Figure 16. The conclusion to be reached from these points is important. In order to obtain an order of magnitude improvement in heat transfer performance, it is necessary to have significant improvement in both internal and external heat transfer coefficients, while avoiding any significant increase in tube wall resistance and minimizing the effects of fouling, non-condensable gases, and condensate inundation.

3. Necessity for Utilizing the Resistance Method for Determination of U_o

Local performance in condenser tube bundles, including the complex interrelationships of non-condensable gases (such as air), vapor pressure drop, steam velocity, condensate rain, tube spacing, etc., cannot be predicted or studied utilizing the Heat Exchange Institute (H.E.I.) condenser design method. With this method, the achievement of the required tube bundle performance is therefore ensured by a generous fouling factor or factor of ignorance [6]. Consequently, in order to design steam condensers closer to desired requirements, it is necessary to employ the more complex resistance method whereby the local heat transfer relationships are integrated over the entire condenser to give the overall performance.

More importantly, the H.E.I. method cannot be used to predict steam condenser performance when enhanced heat transfer techniques are utilized. Again, the resistance method is advantageous since you can adjust the local value of any of the five key resistances and integrate these effects throughout the tube bundle.

IV. METHODOLOGY FOR THE DETERMINATION OF FEASIBILITY

This section contains a description and discussion of the basic design considerations for marine steam condensers and the criteria utilized for evaluating the effectiveness of proposed improvement schemes.

A. CONDENSER DESIGN CONSIDERATIONS

According to Kreith [50], the design of a steam condenser can be broken down into three major phases:

1. Thermal analysis
2. Preliminary mechanical design
3. Design for manufacture.

As Kreith states, thermal analysis primarily involves the determination of the heat transfer surface area required to transfer heat at a specified rate, given the fluid flow rates and temperatures. The mechanical design involves considerations of the operating temperatures and pressures, corrosion/erosion characteristics of the fluids, relative thermal expansions and related thermal stresses, fouling, and the relation between the condenser and other related equipment such as air ejectors and the scoop injection system. The design for manufacture involves the translation of physical characteristics and dimensions into a minimum cost unit that satisfies all the specified requirements. This phase includes selection of materials and specification of mechanical arrangement and manufacturing procedures.

Table 4 is a more detailed list of some of the significant design considerations. All three phases of design, thermal analysis, preliminary mechanical design, and design for manufacture, are significant for determining the feasibility of designing improved marine steam condensers.

1. Thermal Analysis Using the ORCON 1 Computer Program

The thermal resistance method of analysis accounts for heat transfer enhancement on both the shell-side and the tube-side of a condenser. In order to integrate efficiently the local heat transfer characteristics over the entire tube bundle, it is necessary to employ a computer program. One such computer program, ORCON 1, was developed by Oak Ridge National Laboratory for the analysis of desalinization plant condensers. ORCON 1 is a significant contribution to the heat transfer research effort in the field of steam condenser performance analysis [18,53,54]. It performs a thermal analysis on steam condensers with horizontal tubes in a circular or semi-circular bundle.

The basic assumptions required for the use of ORCON 1 are listed below, as reported by Hafford [53]:

1. Coolant flow is in the tube and makes a single pass through the condenser.
2. The condenser proper is a bundle of tubes of circular or semicircular cross-section with a central void. For purposes of calculation, the bundle is divided into identical "sectors" of 30 degrees each, as illustrated in Figure 17.

3. The tubes are spaced on an equilateral triangle pattern; however, for simplicity the tubes are further assumed to be evenly spaced on an arc with a radius from the center.
4. Baffle options on the shell side consist of either no baffles, or simple baffles at 2, 4, 8 and 10 o'clock.
5. Steam flow in the condenser proper is entirely radial, directed toward the void at the center.
6. At most, only six sectors are calculated, assuming the bundle is symmetrical about a vertical center line.
7. The cooler is optional and when included (as in the case of this study) is rectangular in cross-section and is initially equal in height to the radial length of a sector of the condenser. Steam flow proceeds vertically upward through the cooler tube bank. (NOTE: The cooler calculation is independent of the geometry of the condenser. The code takes no recognition of the physical location of the cooler tubes. However, for the volume, weight and cost calculations in this thesis, the cooler tubes were included.)

Figure 18 shows a simple flow-chart of the ORCON 1 program. The main program calls for all the input data as outlined in Table 5. The bundle geometry is then calculated. The number of rows of tubes and tubes per row for a sector are calculated, as well as the inside and outside diameters for the void and bundle. Tube flooding factors are then found using a theoretical average tube in a row, with its number of tubes vertically above it. Cooler dimensions are then estimated, and all remaining initialization of constants or unchanged parameters for the case is done. The subroutine SECALC is then called which finds all the parameters (such as the temperature, pressure and velocity of the steam) of each of the sectors in the condenser. As part of this process the local value of the overall heat transfer coefficient U is determined using the equations outlined in Appendix A.

When these steps are completed for the condenser, a call goes out to the subroutine COOLEX in which the cooler parameters are calculated. After the cooler, the exit fraction of the steam is determined, and an adjustment is made if necessary. If the exit fraction of the steam is not within the required tolerance of the prescribed input value, then a new value of either inlet steam flow rate or tube length is set, and the code returns to the initialization step in MAIN to repeat the calculations. This continues until convergence with the input value is reached. Additional details of this calculational procedure are provided by Hafford [53].

Output data includes a summary tabulation of the overall performance of the condenser, as well as a row by row listing of a variety of parameters such as vapor temperature and velocity, fraction of non-condensables, etc.

The ORCON 1 computer program is a versatile tool for analysis of steam condenser performance. The designer has a wide range of options such as in the selection of tubes and tube spacing, inlet steam conditions, and inlet water properties. This wide variety of options allows the designer to experiment in order to determine the effects of various design modifications. However, the program is limited in bundle geometry and number of baffles and should not be looked upon as a general design program.

The general structure of a computer design method for process heat exchangers has been outlined by Bell [55] and is shown in Table 6. The ORCON 1 program consists of essentially five elements of Bell's nine element general structure. The rating program consists of the main program and analysis subroutines; the input program is a separate subroutine; and the physical/thermodynamics properties program consists of a function subroutine for each required property such as density, viscosity, etc. The design modification program is a limited feature integral with the rating program and capable only of adjusting tube length or steam flow in the process of rating (i.e., determining the heat load performance of) a particular design configuration. The monitor program is also a limited feature consisting of a separate output subroutine and appropriate warning and error messages printed during the rating program analysis steps. Mechanical design, costing, optimization, and systems performance programs are not features of ORCON 1; however, mechanical design and costing are necessary features of a feasibility study.

2. Preliminary Mechanical Design

Mechanical design considerations are particularly important in assessing the risks and uncertainties of improved marine steam condenser designs. The following aspects of design have special significance and impact on the feasibility of proposed improvement schemes.

a. Erosion and Corrosion

Erosion and corrosion problems in condensers have been thoroughly treated by Gilbert [20,56]. Copper-nickel tubing has been specified for marine steam condensers due to its strength (resistance to erosion) and corrosion resistance relative to other materials which are acceptable from a heat transfer point of view. Galvanic corrosion is minimized by the installation of zincs in water boxes. Erosion is limited by maintaining conservative sea water and steam velocities and designing water boxes to minimize turbulence. Current U. S. Navy design practice limits sea water velocity in plain (i.e, smooth bore) copper-nickel tubes to 9 feet per second. This limitation is imposed for the purpose of avoiding erosion problems for the operating life of a condenser. Corrugated and spiral finned tubing have the advantage of providing increased heat transfer rates as a result of increased cooling water turbulence. The disadvantages are the increased pressure drop and the possibility of increased erosion of the tube wall. If the pressure drop is to be held constant, the cooling water velocity in corrugated, or spiral finned tubing will be less than the current U. S. Navy design velocity of 9 feet per second for plain tubing. However, the effects of turbulence must be considered. Basil [57] reported the results of a two year corrosion/erosion study on corrugated copper-nickel. It was demonstrated that water velocities up to 10 feet per second can be employed with corrugated copper-nickel tubing without significant corrosion/erosion damage. It is noted however, that the type of corrugation does have an impact on the degree of corrosion/erosion. It is therefore recommended that the specific tube geometry and material composition be tested or at least thoroughly investigated for any finalized design.

Abrasive erosion caused by suspended abrasive solids, such as sand found in coastal waters, had not been noted as a problem in shipboard condensers. This is probably due to the fact that vessels in port generally operate at low steaming rates which demand only low condenser cooling water flow rates.

Titanium tubing has the advantage of significantly greater strength properties than the copper-nickel allows (See Table 1.). This greater strength permits increased sea water velocities [58]. Titanium has been tested in sea water, brackish water, and dirty river water with immeasurable corrosion rates where brass and copper-nickel alloys were corroded in two to five years [57]. The excellent corrosion and erosion resistance of titanium is reported on by Adamson [5]. Particularly noteworthy is the immunity of titanium to pitting corrosion with sea water temperatures below 250°F [5].

b. Tube Strength, Rigidity, and Vibration

Condenser tubing is subject to various stresses which are highly dependent on the specific marine application. Particularly for submarine applications, the ability of the condenser piping, water boxes, tube sheets, and tubes to withstand pressure at operating depths is a major consideration in condenser design. Tube rigidity is extremely important and dictates the placement of tube bundle support plates to keep tube deflection and tube vibration within acceptable limits. Axial loading must be considered and kept within acceptable limits in order to prevent the tubes from being pushed out of the tube sheet by compressive loads or pulled through the tube sheet by tensile axial loads [59].

Proper engineering analysis and manufacturing techniques are required for maximum condenser reliability [59]. Considerable attention has been given to this subject by Westinghouse Electric Corporation [59,60].

Similar analysis is required for corrugated, internally finned, and titanium tubing. Engineering vibration studies on corrugated tubing were conducted at Auburn University [61] and Oklahoma State University [62] under the sponsorship of Wolverine Tube Division of Universal Oil Products, Incorporated. The studies indicate that the mechanical behavior of corrugated tubing can be analyzed similar to plain tubing, and the results demonstrate the mechanical feasibility of utilizing corrugated tubing for marine steam condenser applications.

c. Fouling

(1) Mechanical

Mechanical fouling must be allowed for in the design of marine steam condensers. Current practice, utilizing the Heat Exchange Institute [52] or design data sheet [3] method for condenser design, employs a multiplicative factor for fouling. This method is basically unsound from the standpoint that the allowed fouling resistance varies directly with the overall heat transfer coefficient while the actual fouling resistance may remain constant or even vary inversely. This could be true particularly where an increased overall heat transfer coefficient is the result of increased cooling water velocity. The selection of fouling factors is particularly important when analyzing different tube geometries for performance comparisons.

Fouling factors are used to a certain extent as factors of ignorance [6]. For the purpose of design comparisons, there is a need for accurate prediction of in-service fouling factors. Starner [63] discusses the impact of fouling factor selection on heat exchanger performance and concludes that further research is needed in this area for the proper evaluation of enhanced surface tubing.

A review of the history and status of fouling research was conducted by Suitor, Marner, and Ritter [64] with the general conclusion that additional research is required in both the area of deposition and removal. The study of shear stress is most significant in understanding the removal process. The study of the complex interrelationships of velocity, wall surface, scale surface, bulk fluid temperatures, and particularly water chemistry is required for improved knowledge and prediction of deposition. Although no literature appears to be available on the study of fouling in internally finned tubing, it would seem that internally finned tubing would be more susceptable to mechanical fouling than either plain or corrugated tubing and that tube cleaning would be more difficult.

(2) Biological

Biological fouling is not a problem with copper-nickel tubing since copper is highly toxic to marine organisms. Titanium tubing, however, is neutral to marine organisms and has been demonstrated to be highly susceptable to biological fouling at very low sea water velocities [65]. Considerable research has been conducted in this area by the Naval Ship Research and Development Center [48] in conjunction with heat transfer performance analysis [5] of titanium tubing and titanium condensers.

Barnacles and mussels are considered to be the most significant problem because of their hardness and prevalence [48]. Various methods have been considered as a prevention or cure for biological fouling including mechanical cleaning, heating the seawater or surface to be protected, maintaining high seawater velocities, or adding chlorine or another toxic substance to the seawater. Mechanical cleaning is time consuming and requires system shutdown. Commercial mechanical cleaning systems such as the M.A.N. method and AMERTAP may be investigated, but the primary usage of these systems to date has been for mechanical fouling. Sea water or surface heating presents problems since saturation temperatures in the condensers are typically less than the minimum temperature (125°F) required for effective biological fouling protection. A high velocity flow system could be developed with the employment of a recirculating loop, but the increased system complexity and uncertainty of complete protection due to the possibility of low velocity or stagnation points in the system make this alternative unattractive.

Intermittent chlorination seems to hold the most promise for prevention of biological fouling. The most critical situation is low velocity flow. This is particularly true during operations in port since there is a general decline in biofouling organisms with depth and distance from shore [66]. Another possibility which has been suggested is the injection of chlorine dioxide [67] which may be preferable from an environmental standpoint but which requires further investigation.

Actual installation of a prototype titanium air ejector condenser for submarine use has been programmed for FY 77-78. With the aid of an operating system and an analysis of condenser operating characteristics, the establishment of firm shipboard fouling guidelines and expected fouling patterns should be greatly enhanced.

d. Scoop Injection Systems

Modern naval surface vessels and some submarines are designed with scoop injection systems capable of supplying sufficient cooling water to the condenser over the entire ship's speed range from approximately 5-7 knots to full power. This does not eliminate the need for a cooling water circulating pump, but the circulating pump can be smaller in size and utilized less frequently. Scoop injection systems have demonstrated in practice that they are higher in performance and more economical than pump circulation systems with the result that modern tankers are being designed with scoop circulation systems [60]. Considerable research was conducted on scoop injection systems during the 1930's, and the design of these systems for naval vessels is still based on the work of Hewins and Reilly published in 1940 [68]. With larger ships and particularly tankers with main machinery spaces far aft in the vessel, the boundary layer thickness on the vessel's hull in the vicinity of the scoop is much larger, and the velocity distribution in the boundary layer more complex than for smaller vessels with amidships scoops. Based upon more recent studies [69], current scoop injection systems for tankers are designed more appropriately for the boundary layer conditions which are encountered as a result of that ship type construction.

The search continues for improved performance scoop injection systems due particularly to the importance of cooling water system drag and expenditure on the overall economics of ship operations [70].

For this report, it is the interrelationships between heat transfer performance and cooling water mass flow, velocity and pressure drop which are important. This is particularly true in considering the case of the increased cooling water velocities permissible with the utilization of titanium tubing. For plain tubing, seawater pressure drop is proportional to almost the square of the water velocity. This means that in order to take full advantage of the improved heat transfer performance, it is necessary to accommodate the increased pressure drop. Adamson [5] expresses concern about the adequacy of current scoop injection system designs and the uncertainty of effectively utilizing cooling water velocities of 10 to 12 feet per second. Certainly there is reason to be more concerned about the situation at cooling water velocities of 15 feet per second. However, more recent studies have demonstrated the definite possibility of improved scoop circulating systems and the attainment of higher pressure differentials. Based on the data from English [70], it appears possible to obtain twice the sea water pressure head currently utilized in steam condenser designs. Further investigation is necessary, and it is not at all clear whether the increased hull drag and power required would be an acceptable tradeoff against the improved condenser heat transfer performance.

3. Design for Manufacture

The types of heat transfer enhancement considered for compact marine steam condensers present some unique fabrication and assembly problems [71]. Corrugated or roped tubing must be fabricated with plain "land" sections for support plate and tube sheet penetrations. The corrugated or roped sections must be the same size or slightly smaller than the plain sections in order to allow tube insertion through the tube sheets and support plates. This does not present difficulties since the corrugated or roped type tubing is formed from a plain type tube. Internally finned tubing does not create any significant condenser assembly problems although there are fabrication problems with titanium. Plain titanium tubing requires no significant condenser assembly considerations other than the method for securing the tubes in the tube sheet [5]. Installation costs for rolling titanium tubes into a monel tube sheet should be similar to copper-nickel tubing, although a closer tolerance would have to be maintained on the tube sheet holes. If tubing is secured by welding, tube-to-tube sheet fabrication costs would be significantly (two or three times greater than rolled copper-nickel tubing) increased due primarily to the necessity of utilizing a titanium tube sheet [5].

A significant assembly problem may exist with a dropwise promoted tube unless the promoter could be sprayed or otherwise applied after tube bundle was assembled. The promoter would necessarily have to be hard and resilient in order to avoid damage due to mechanical abrasion when installing the tubes in the tube sheets and support plates. The assembly technique must be addressed based upon the promoter coating characteristics and susceptibility to damage.

B. PERFORMANCE EVALUATION — MEASURE OF EFFECTIVENESS

1. Determining Design Performance Criteria

In order to evaluate the relative performance of marine steam condenser designs, some measures of effectiveness (i.e., design performance criteria) must be established. Bergles, Blumenkrantz, and Taborek [72] provide some practical criteria to define the merits of heat transfer augmentation techniques in single phase flow. One of their criteria which has practical significance aboard naval vessels is to fix the heat load, seawater pressure drop, and flow rate (i.e., pumping power) with the objective of reducing heat exchanger size. For the purposes of this study, four measures of effectiveness are considered, and comparisons are made relative to a standard or baseline design. The four measures of effectiveness for condenser designs are as follows:

1. Heat load capacity at equivalent pumping power
2. Weight at equivalent heat load
3. Volume at equivalent heat load
4. Cost at equivalent heat load

In the last three cases, the sea water pressure drop is held constant, and the seawater flow rate is allowed to vary as necessary in order to accommodate the desired heat load.

The four measures of effectiveness are selected due to their importance in naval ship design. Equivalent pumping power allows the utilization of existing scoop injection systems and cooling water circulating pumps in designing a higher heat load propulsion plant. Weight, volume, and cost are significant parameters in naval ship design and are of interest in designing propulsion plants to provide a specific amount of shaft horsepower for a naval vessel. In certain cases, full advantage of an augmentation technique cannot be obtained unless cooling water pressure drop is allowed to increase. This is particularly true when the performance improvement is the result of higher cooling water velocities allowable with the utilization of titanium tubing. To obtain these higher pressure heads, existing scoop injection systems may have to be re-designed as discussed earlier.

2. Performance Calculations

a. Heat Load Capacity Calculations

Heat load capacity at equivalent pumping power is taken directly from the output data of ORCON 1 following the procedure mentioned later in Section IV-C.

b. Volume Calculations

Volume calculations are based on the tube bundle geometry equations from ORCON 1. A sample calculation is provided in Appendix B.

c. Weight Calculations

Total standard condenser dry weight is determined from a parametric estimate of weight versus condensing surface area as shown in Figure 19. With augmented tubes, the condenser dry weight is determined from the same parametric curve, but the estimate is modified based on the weight of the augmented condenser tubes compared to the standard condenser tube. Condenser operating weight or wet weight is utilized for comparison of designs in this report. The wet weight is calculated by adding condenser dry weight and the cooling water weight in the tubes and headers. The headers are assumed to be half cylinders, with the diameter determined from the tube bundle geometry equations from ORCON 1. The weight of steam and condensate in the condenser is neglected due to the fact that hotwell dimensions were not determined. A sample calculation is provided in Appendix C.

d. Cost Calculations

Total standard condenser cost is determined from a parametric estimate of cost versus condensing surface area as shown in Figure 20. The augmented condenser cost is determined from the same parametric curve, but the estimate is modified based on the cost of the augmented condenser tubes compared to the standard condenser tube. The parametric curve utilized for the cost comparison was based on assembled cost at the plant (i.e., installation and shipping costs not included). A sample calculation for cost is provided in Appendix D.

C. OVERALL CALCULATIONAL PROCEDURE

The basic steps which were followed in this study to analyze and evaluate specific enhanced heat transfer schemes in condensers are listed below.

1. Specify a standard or baseline design utilizing an operational condenser for a model, and obtain an ORCON 1 analysis (including number of tubes, steam flow rate, etc.) as a basis for comparison.

2. Specify a heat transfer augmentation technique to be compared with the standard design.

3. For a chosen augmentation technique, calculate required tube length and number of tubes for input to ORCON 1 while maintaining the pumping power equal to that of the standard design. If the maintenance of equal pumping power is not realistic based upon the other design parameters,

increase the pumping power by multiplying by 1.5 or 2.0 for purposes of comparison. Obtain an ORCON 1 analysis for the new geometry.

4. Compare heat load capacity for the augmented designs with the standard design at equivalent or constant pumping power.

5. Next, assume that the calculated heat flux for a given design is constant throughout the condenser, and reduce the number of tubes (and hence the area) in order to obtain a heat load equivalent to the standard design. Because of this reduction in the number of tubes, while keeping the pressure drop through the tubes constant, the total seawater flow rate will be altered, thereby changing the pumping power from the standard design.

6. For this case of equivalent heat load, calculate and compare weight, volume and cost for the standard and augmented condenser designs.

7. Assess the augmented design risks and uncertainties relative to the performance comparisons.

D. EPILOGUE

Paraphrasing Bell [55] again, the engineer has final responsibility for verifying and accepting each design. The computer should be looked upon as a tool for the use of the designer, and the results produced from any computer design program should be thoroughly checked particularly with something as complex as a steam condenser. The simplifying assumptions utilized in this report are exceedingly rough, and various researchers would disagree on the correct equations and data to be utilized for performance analysis. It must be noted, however, that this report is a feasibility study and not a finalized design. Any finalized marine steam condenser designs would need to be analyzed in much greater detail in order to ensure that the correlations, assumptions and results are strictly applicable to the specific design.

V. CASE ANALYSIS RESULTS AND DISCUSSION

A. DESCRIPTION OF TEST CASES

In order to evaluate the feasibility of designing improved marine steam condensers utilizing various heat transfer augmentation techniques, sixteen representative cases were analyzed with the aid of the ORCON 1 computer program. The tubing descriptions and mode of condensation used in these cases are briefly described in Table 7. For each representative case, input parameters for use in ORCON 1 were generated based, in general, on cooling water pressure drop and cooling water mass flow rate equivalent to the standard design, as described in Section IV-B-1. Appropriate heat transfer enhancement factors on the inside surface (tube-side) and outside surface (shell-side), as defined by the heat transfer correlations presented in Appendix A, were also generated for use in ORCON 1. A listing of specific input data is contained in Table 8. In addition, the following input parameters were held constant in all test cases:

Steam saturation temperature:	125.4 °F
Sea water inlet temperature:	75.0 °F
Tube spacing/diameter ratio:	1.60
Percentage of tubes in cooler:	6 percent
Fouling resistance:	0.0003 hr ft^2 °F/BTU
Non-condensable gas flow rate:	22.5 lb_m/hr
Exit steam fraction:	0.02 percent

1. Standard Design Case

The standard or baseline case is an approximation of a modern marine steam condenser from a 1200 PSIG steam propulsion plant such as on the aircraft carrier U.S.S. JOHN F. KENNEDY. The actual tube sheet layout and steam lanes, as shown earlier in Figure 4, as well as the shell configuration, could not be duplicated due to the geometric limitations of ORCON 1. However, a schematic representation of the KENNEDY condenser was made for use in ORCON 1, and this representation is sketched in Figure 21. In addition, tubing design parameters, such as thermal conductivity and dimensions, and the steam and cooling water inlet characteristics are consistent with the actual design [73].

2. Improved Heat Transfer Cases

The fifteen test cases with improved heat transfer were chosen based upon the enhancement techniques discussed in Chapter II. In particular, test cases 1, 2 and 3 used corrugated copper-nickel tubing as manufactured by Wolverine Tube Division of Universal Oil Products [74]. In these cases the nominal tube outside diameter was increased from 5/8 inch to 3/4 inch and 1.0 inch, respectively. Test case 4 used plain copper-nickel tubing with dropwise condensation assumed on the outside surface. Test cases 5, 6 and 7 used thin-walled titanium tubing with sea water velocities of 9.0, 12.0 and 15.0 feet per second, respectively. Test case 8 was identical to case 7 except that dropwise condensation was assumed on the outside surface. Test cases 9, 10 and 11 used corrugated titanium tubing similar to cases 1, 2 and 3. The last four cases, 12, 13, 14 and 15 used internally finned copper-nickel tubes as manufactured by Noranda Metal Industries. In these cases, the nominal tube outside diameter was kept at 3/4 inch, but the internal fin characteristics were altered between cases 12 and 13 and also between bases 14 and 15 [25]. Film condensation was assumed for cases 12 and 13, while dropwise condensation was assumed for cases 14 and 15.

B. EFFECTS OF NON-CONDENSABLE GAS AND CONDENSATE FLOODING ON STANDARD DESIGN

The effects of non-condensable gas and condensate flooding, or inundation, are important for the design and operation of steam condensers. Therefore, a number of variations were tried on the standard design case with the intent of observing trends and determining the sensitivity of the heat load capability of a steam condenser to these factors. These results are summarized in Table 9 and are discussed below.

1. Non-condensable Gas

As shown in Table 9, the effects of non-condensable gas concentration up to three times the design level had little effect on heat load carrying capability in this instance. However, a factor of ten increase in concentration of non-condensables resulted in a 50 percent reduction in heat load performance. The influence of non-condensables is clearly non-linear with concentration.

2. <u>Condensate Flooding, or Inundation</u>

Condensate inundation has a significant effect on heat load performance as shown by the results with and without baffles and also by the series of tests with varying numbers and size of tube bundles in the condenser. Clearly, providing baffles and designing with smaller (therefore shallower) tube bundles gives a thinner condensate film on the lower tubes, thereby increasing the heat transfer rate. Perhaps the most significant result, however, is the approximately 16 percent variation in predicted performance depending on which flooding correlation is utilized to account for condensate flooding. The variation in existing flooding data, as shown in Figure 15, demonstrates the need for additional investigation in this important area.

C. HEAT LOAD COMPARISON AT CONSTANT PUMPING POWER

The various heat load capacities of each test case as a percentage increase (+) or decrease (-) relative to the standard design are tabulated in Table 10. The best heat load performance for each augmentation category is summarized below:

1. <u>Internal Enhancement</u>:

Increased Velocity with Titanium	+ 7.2%
Internally Finned Copper-Nickel	+ 9.9%
Corrugated Titanium	+33.4%
Corrugated Copper-Nickel	+33.7%

2. External Enhancement:

Dropwise Condensation on Copper-Nickel	+30.1%

3. Internal and External Enhancement:

Increased Velocity with Dropwise Condensation on Titanium	+35.5%
Internally Finned and Dropwise Condensation on Copper-Nickel	+50.7%

These results demonstrate the significance of improving both internal and external heat transfer coefficients. Since the internal and external coefficients are of the same order of magnitude in the standard case (See Figure 16), much more dramatic heat load performance improvement is realizable if both coefficients are improved. Also, since the tube material offers a significant resistance to heat transfer, it is important to select tube wall thicknesses which are mechanically adequate but not overdesigned for the particular application. In the case of titanium, even with a wall thickness of 0.022 inches, heat load performance deteriorates except for sea water velocities greater than 9 feet per second. These increased velocities increase the pressure drop and pumping power, however, and may not be feasible in shipboard operation unless improved scoop injection systems [68], and/or larger circulating pumps, are installed. The heat load performance for the finned tubing selected for analysis is significantly less than that of the corrugated tubing. Based on the available data [25,74], improvement might be anticipated with better selection of tubing size and fin configuration for this particular application. Dropwise condensation applied to the outside of the tubes can be very effective in improving condenser heat load performance. These dropwise results, however, presume a ten-fold increase in h_o for copper-nickel over the film condensation case and an increase of two and one half for titanium. These enhancement factors must be verified experimentally for tube banks, however, before a more detailed comparison can be made.

In general, these comparisons show that the condenser heat load can be increased by approximately 50 percent at a constant pumping power, using combined internal and external enhancement techniques.

D. VOLUME, WEIGHT AND COST COMPARISON AT CONSTANT HEAT LOAD

For each of the enhanced heat transfer cases, Table 11 lists the percentage deviation of volume, weight, and cost from the standard design. This comparison is made at constant heat load and pressure drop, except in case 5 where the pressure drop is increased by 50 percent and in cases 6 and 7 where the pressure drop is increased by 100 percent. In order to maintain a constant heat load, the sea water mass flow rate is adjusted as discussed in Section IV-C, so its percentage deviation is also included.

1. Corrugated Copper-Nickel (Cases 1,2,3)

Corrugated copper-nickel tubing improves condenser performance in terms of weight and cost reduction if 3/4 inch or 1 inch diameter tubes are utilized, but a penalty is paid in terms of increased volume. This is particularly so for the 1 inch diameter tube case which shows the best weight reduction (6.7 percent) and the best cost reduction (9.9 percent) at the expense of increased tube length and bundle radius. This volume penalty could be reduced or reversed by reducing the tube spacing to diameter ratio S/D, but a case study would have to be run in order to determine the effect of S/D on heat transfer performance.

The primary virtue of corrugated copper-nickel tubing is in the increased internal heat transfer coefficient due to the turbulent nature of the cooling water flow. Young, Withers and Lampert report only slight improvement in the external heat transfer coefficient [13]. The baseline design wall thickness was utilized to ensure strength, and therefore wall resistance remains essentially the same as the baseline design.

2. Dropwise Condensation Promoted on Plain (Standard) Copper-Nickel (Case 4)

Dropwise condensation promoted on plain copper-nickel produces weight reduction (18.5 percent), cost reduction (13.0 percent), and significant volume reduction (24.5 percent). The above-mentioned cost reduction, however, is overly optimistic because the cost of promoter application on the tube has not been included.

The major advantage of dropwise condensation promotion is the increased external heat transfer coefficient with copper-nickel (an order of magnitude increase even when including the promoter resistance [45]). Although the internal heat transfer coefficient and tube wall resistance are the same as for the baseline condenser, it is an advantage that tubing size and cooling water pressure drop are not affected by promoting dropwise condensation. The increased performance is directly related to a reduction in the number of tubes and corresponding reductions in weight and volume.

The major disadvantages are the susceptibility of dropwise promoters to contamination (with resultant loss of dropwise promotion) and the uncertainty of promoter reliability and cost.

3. Plain Titanium (Cases 5,6,7)

At a sea water velocity of 9 feet per second, plain titanium tubing (5/8 inch diameter) with reduced wall thickness (BWG 24; 0.0220 inch) produces a dramatic weight reduction (26 percent, wet) and a small volume reduction (6 percent) at the expense of greater cost (21 percent). From an overall design viewpoint, the law of diminishing returns seems to be in effect for cooling water velocities above 9 feet per second. The percentage decrease in weight and cost is less than 2 percent for a velocity increase from 9 to 12 feet per second. It is also noteworthy that in order to benefit from the increased heat transfer coefficients possible with higher sea water velocities (12 and 15 feet per second), it is necessary to pay the penalty of substantially increased cooling water pressure drops (multiplicative factors of 1.5 and 2.0, respectively, as shown in Table 8). A brief discussion of the implications of increased cooling water pressure drop in conjunction with scoop injection systems is provided in Section IV-A-2-d.

Titanium is non-toxic to marine organisms and therefore susceptible to biological fouling unless special precautions are taken. A discussion of this problem is provided in Section IV-A-2-c.

The greater cost of titanium is a significant disadvantage. The cost analysis was performed utilizing code welded tubing which is cheaper than seamless. The welded titanium should be suitable for surface vessel applications, but additional considerations must be taken into account for submarines due to the pressure environment and the cycling from atmospheric to depth pressure. Further investigation may be necessary for application of welded titanium tubing to submarine applications.

The primary virtues of titanium are its excellent strength properties coupled with very low density. However, its thermal conductivity is very low compared with copper-nickel. The properties of titanium are compared to those of other materials in Table 1.

4. Dropwise Condensation on Plain Titanium (Case 8)

Plain 24 BWG (0.0220 inch wall) titanium coated with a dropwise promoter was analyzed with a cooling water velocity of 15 feet per second. The results show dramatic decreases in both volume (36.7 percent) and weight (44.0 percent) with essentially no change in cost (although the cost of promoter application is not included). This case is identical to case 7 except for the assumption of dropwise condensation. A dramatic improvement in volume is evident when these two cases are compared. This improvement is due to heat transfer enhancement on both sides of the tube simultaneously. The internal heat transfer coefficient is enhanced by the increased sea water velocity, and the external heat transfer coefficient is enhanced by the promotion of dropwise condensation. Unfortunately, the technological feasibility of applying a dropwise promoter coating on a titanium tube has not been verified experimentally.

5. Corrugated Titanium (Cases 9,10,11)

Using the nominal 5/8 inch diameter tube, a dramatic weight reduction (26 percent) and a volume reduction (9 percent) occurs. These reductions are at the expense of a particularly significant cost increase (36 percent). For the larger tubes, a volume increase occurs, similar to the corrugated copper-nickel cases. As pointed out earlier, this volume penalty in the case of the larger diameter tubing could be reduced or reversed by reducing the tube spacing to diameter ratio S/D, but a case study would have to be run in order to determine the effect of S/D on heat transfer performance.

The significant weight reduction is due primarily to the titanium. An additional performance improvement is possible if the sea water velocity is increased beyond

5.2 feet per second. Titanium tubing should be able to withstand increased turbulence at higher velocities; however, much larger pressure drops would be required.

6. Internally Finned Copper-Nickel (Cases 12,13)

Using a nominal 3/4 inch diameter copper-nickel tube, with straight inner fins, a weight reduction (11 percent) occurs, at the expense of a volume increase (8 percent) and a cost increase (4 percent). For the spiral fin tube, the performance decreases when compared to the standard design. This is due to the lower sea water velocity necessary in this case (6.63 compared to 9 feet per second) to offset the increased equivalent friction factor due to the presence of the fins. The straight fin tube outperforms the spiral fin tube due primarily to the thinner wall thickness. In each of these tubes there is an increased potential for mechanical fouling and greater cleaning difficulties will be encountered, particularly with spiral fins.

7. Dropwise Condensation on Internally Finned Copper-Nickel (Cases 14,15)

Internally finned tubing, coated on the outside with a dropwise promoter, produces dramatic weight reduction (30 percent) and substantial volume reduction (23 percent) with probable increased but uncertain cost. A further reduction in volume is realizable in this case with a reduced spacing to diameter ratio S/D, although the effect of S/D on heat transfer performance must be determined, as mentioned earlier.

The dramatic improvement of cases 14 and 15 when compared to the previous two cases is due to the heat transfer enhancement on both sides of the tube. However, the uncertainties associated with dropwise promotion and the fouling and cleaning question associated with internally finned tubing need to be considered.

VI. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

1. The use of condenser thermal analysis codes, such as ORCON 1 as developed by the Oak Ridge National Laboratory, are particularly advantageous in studying the feasibility of using enhanced heat transfer techniques in marine condensers.

2. Marine condenser heat loads at constant pumping power can be increased by approximately 50 percent using heat transfer enhancement techniques.

3. At a constant heat load, a reduction in condenser weight and volume of approximately 40 percent can occur if dropwise condensation is promoted on the outside of 5/8 inch diameter titanium tubes through which sea water flows at 15 feet per second. This reduction, however, occurs at the expense of pumping power (approximately 300 percent increase) and may incur an increase in cost.

4. A similar reduction in weight and volume, near 30 percent, can occur if dropwise condensation is promoted on the outside of 3/4 inch diameter, internally finned copper-nickel tubes with a sea water velocity of 7 feet per second. This enhancement technique also gives a reduction of 34 percent in pumping power and may decrease condenser cost as well. The influence of fouling inside these internally finned tubes is a significant uncertainty in this scheme, however.

5. The replacement of 5/8 inch diameter, copper-nickel tubes with thin walled titanium tubes, at the same sea water velocity, gives a dramatic reduction in condenser weight (near 30 percent) at the expense of cost (near 20 percent). Increasing sea water velocity from 9 to 15 feet per second provides a small additional improvement in volume and weight at the expense of a significant increase in pumping power. High sea water velocities are only attractive if heat transfer is enhanced on the outside of the tube as well.

6. Corrugated copper-nickel tubes can be used to decrease condenser weight and cost at equivalent heat load and pumping power. This is particularly true for 1.0 inch diameter tubes. These reductions, however, are less than 10 percent and occur at the expense of increased condenser volume (near 20 percent). The condenser volume, however, may be reduced if the analysis is made at tube spacing to diameter ratios less than the value of 1.60 which was used in this study.

7. Promoting dropwise condensation on conventional copper-nickel tubes can reduce condenser weight and volume by as much as 20 percent, at constant heat load and pumping power. A large uncertainty exists, however, as to the long term effectiveness of existing dropwise promoters.

8. The availability of condenser cost data is rather limited, especially in regard to fabrication. This limited data leads to large uncertainties in the costs which were calculated during this study.

B. RECOMMENDATIONS

1. The Navy should further develop existing thermal analysis computer codes such as ORCON 1 for condenser use. Such codes can be written to analyze both two and three dimensional effects, as well as a variety of enhanced geometry techniques.

2. The effect of tube spacing to diameter ratio on condenser design should be further studied, particularly with enhanced heat transfer schemes.

3. An experimental program should be carried out to investigate the heat transfer performance of a variety of tube enhancement techniques. A systematic comparison of various tube types should be made under simulated marine condenser environments.

4. The application of ultra thin dropwise promoters on both copper-nickel and titanium tubing should be studied in regard to method of application, heat transfer performance, reliability and cost. The influence of condensate drainage and high steam velocities on dropwise condensation heat transfer coefficients should also be examined.

5. Modified scoop injection systems should be considered to provide a higher pressure head for condenser sea water cooling. Of course, the influence of these systems on hull resistance must also be taken into account.

6. Continued testing of titanium-sea water corrosion is important for future use of this material. The particular effect of biological fouling should be continuously evaluated.

7. In regard to computer-aided ship design, the Navy should develop computer codes to optimize condenser designs with regard to weight, volume, cost, pumping power, etc., at a constant heat load.

8. The Navy, perhaps with the Department of Energy (OTEC Program), should sponsor a workshop on marine condenser designs of the future. This workshop would include representatives of the Navy, government, industry and universities to explore future trends in condenser designs.

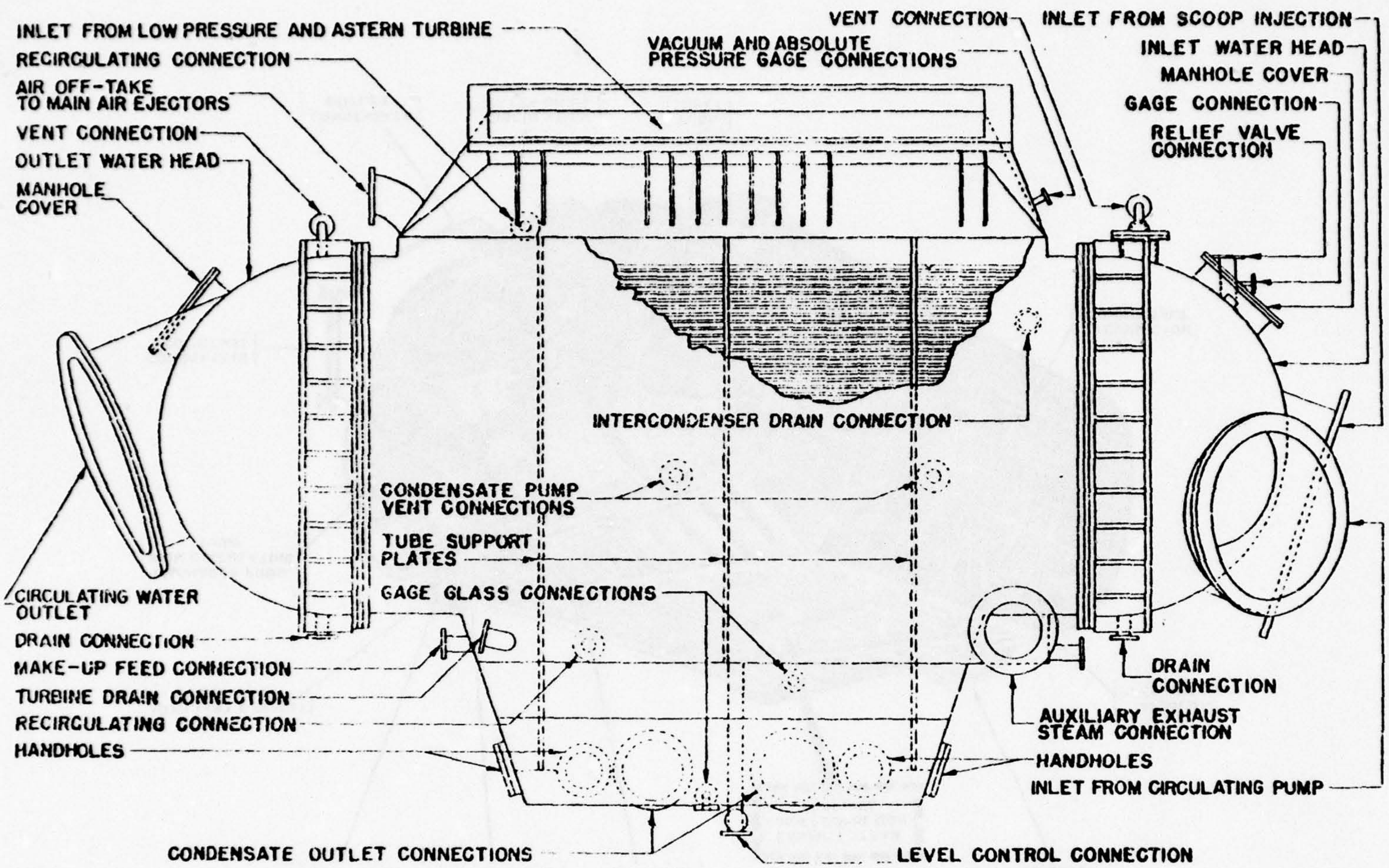


Figure 1. Outline Drawing of Typical Main Condenser

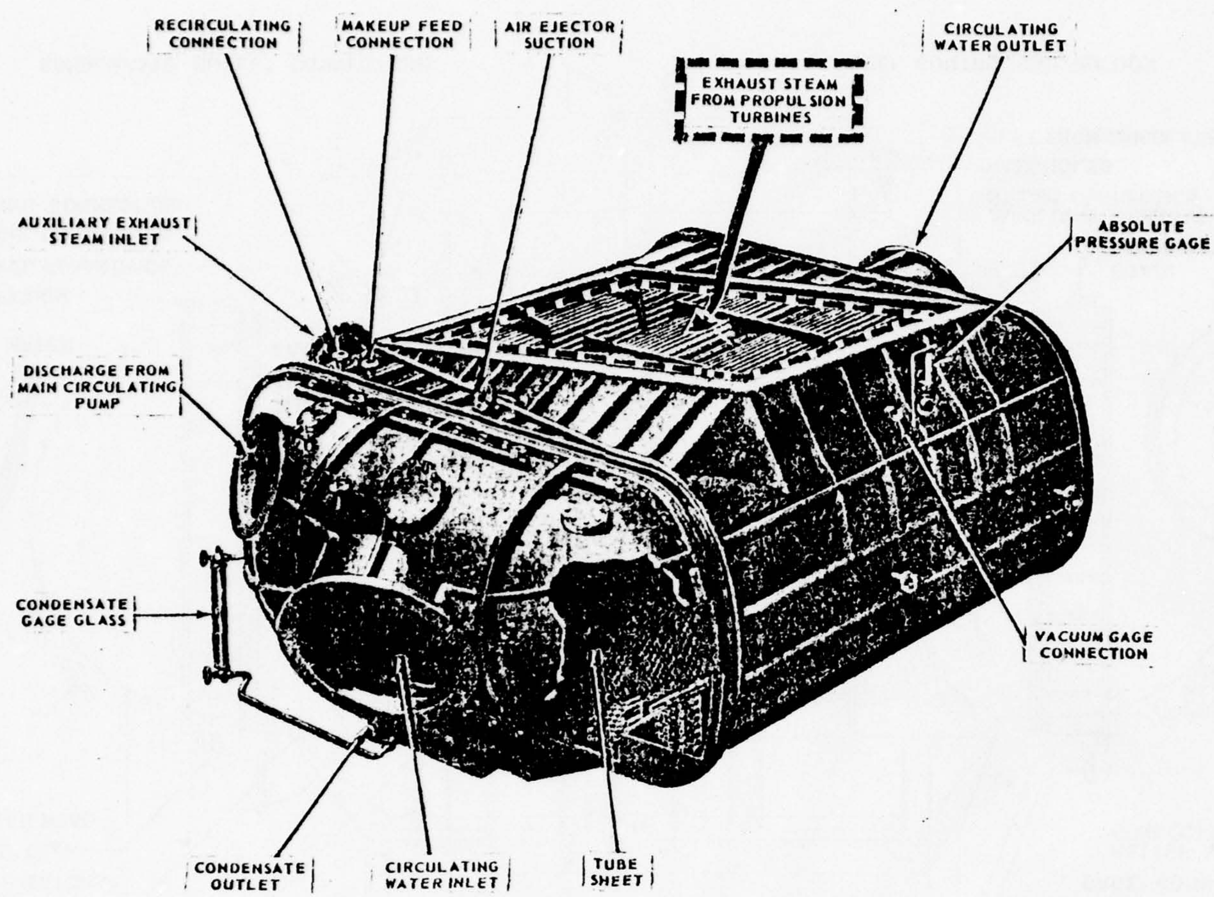


Figure 2. Cutaway View of Typical Main Condenser

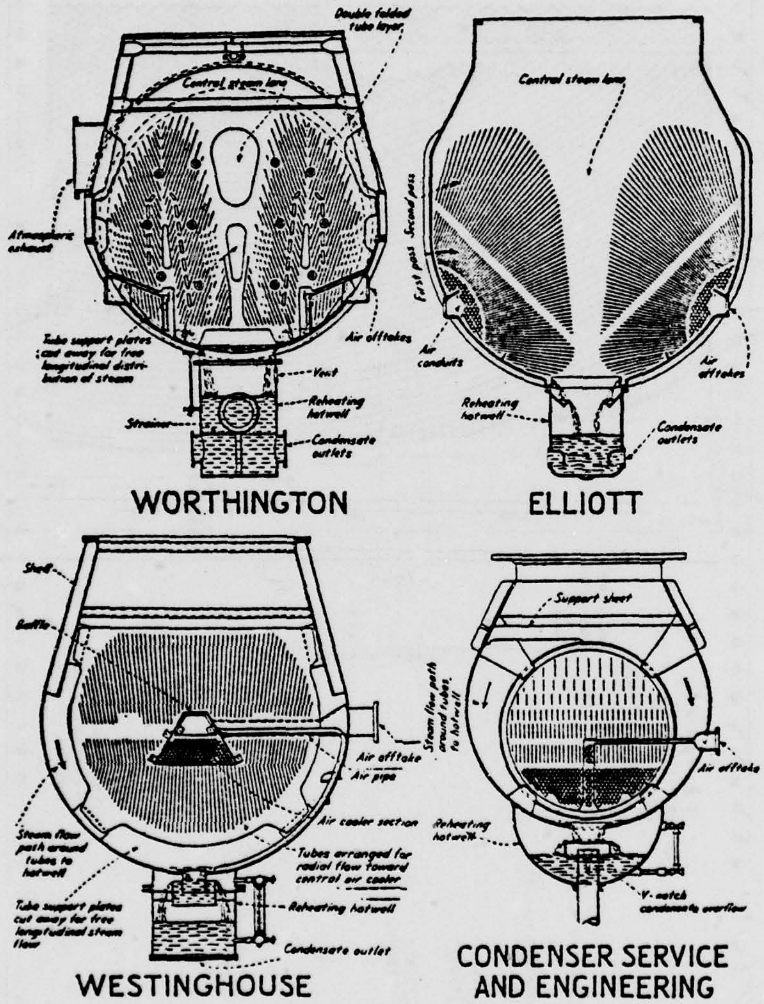


Figure 3. Tube Sheet Layouts of Various Condenser Configurations

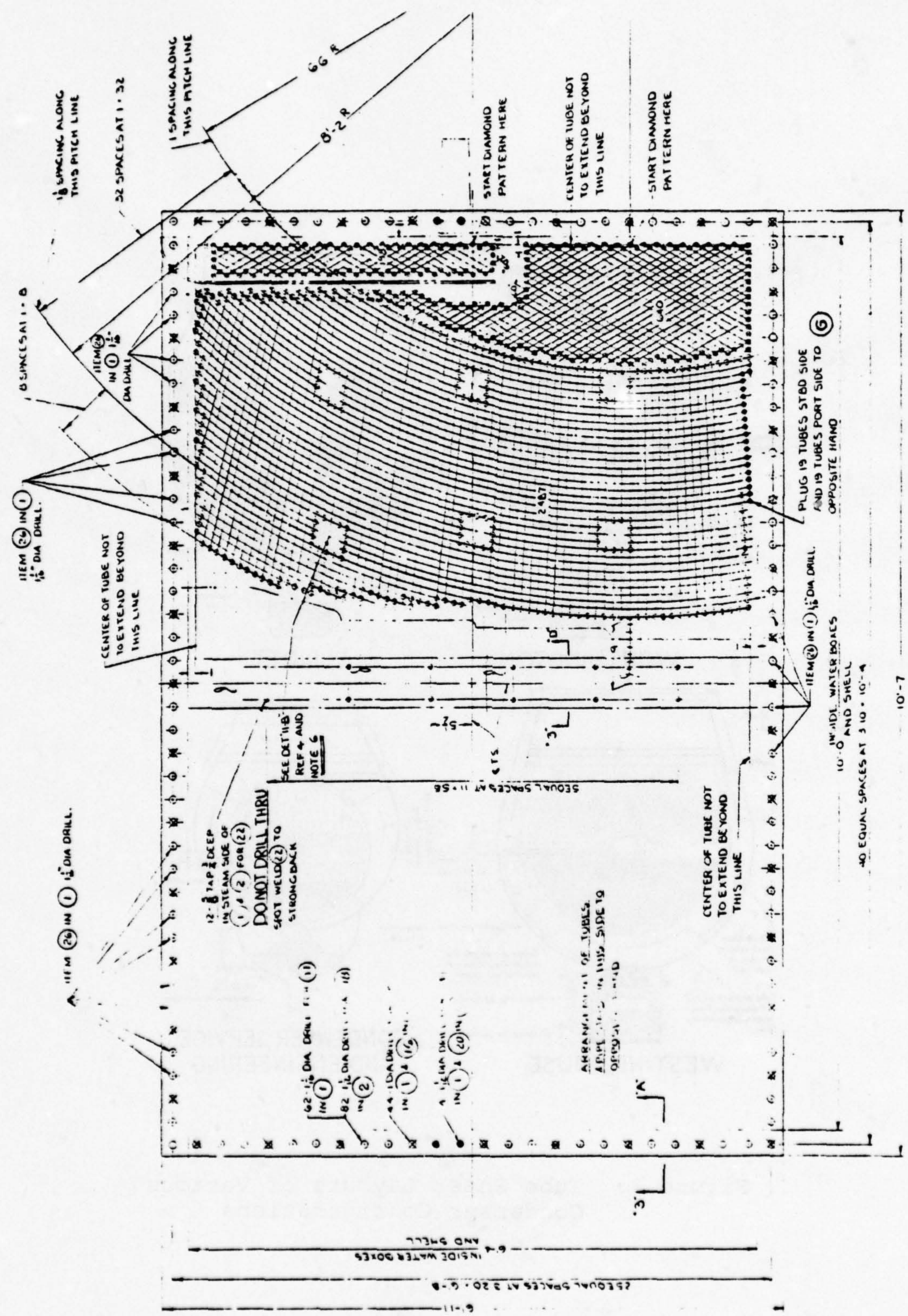


Figure 4. Main Condenser Tube Sheet Layout on CVA-67, U.S.S. JOHN F. KENNEDY

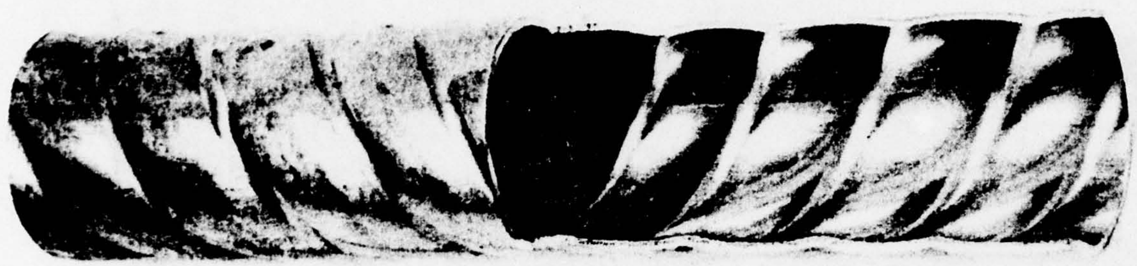


FIGURE 5. Photograph of Spiral Indented (Rope) Tube

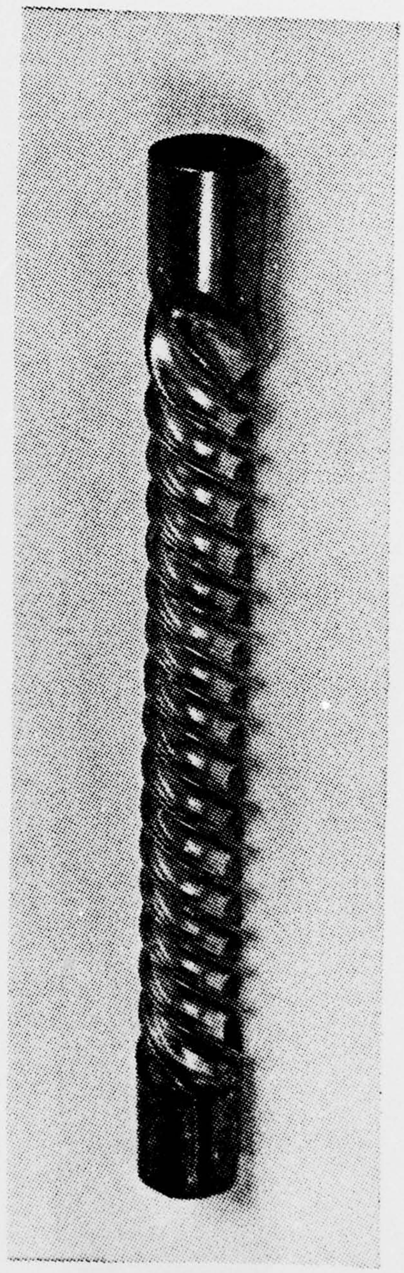


FIGURE 6. Photograph of Tubotec Tube

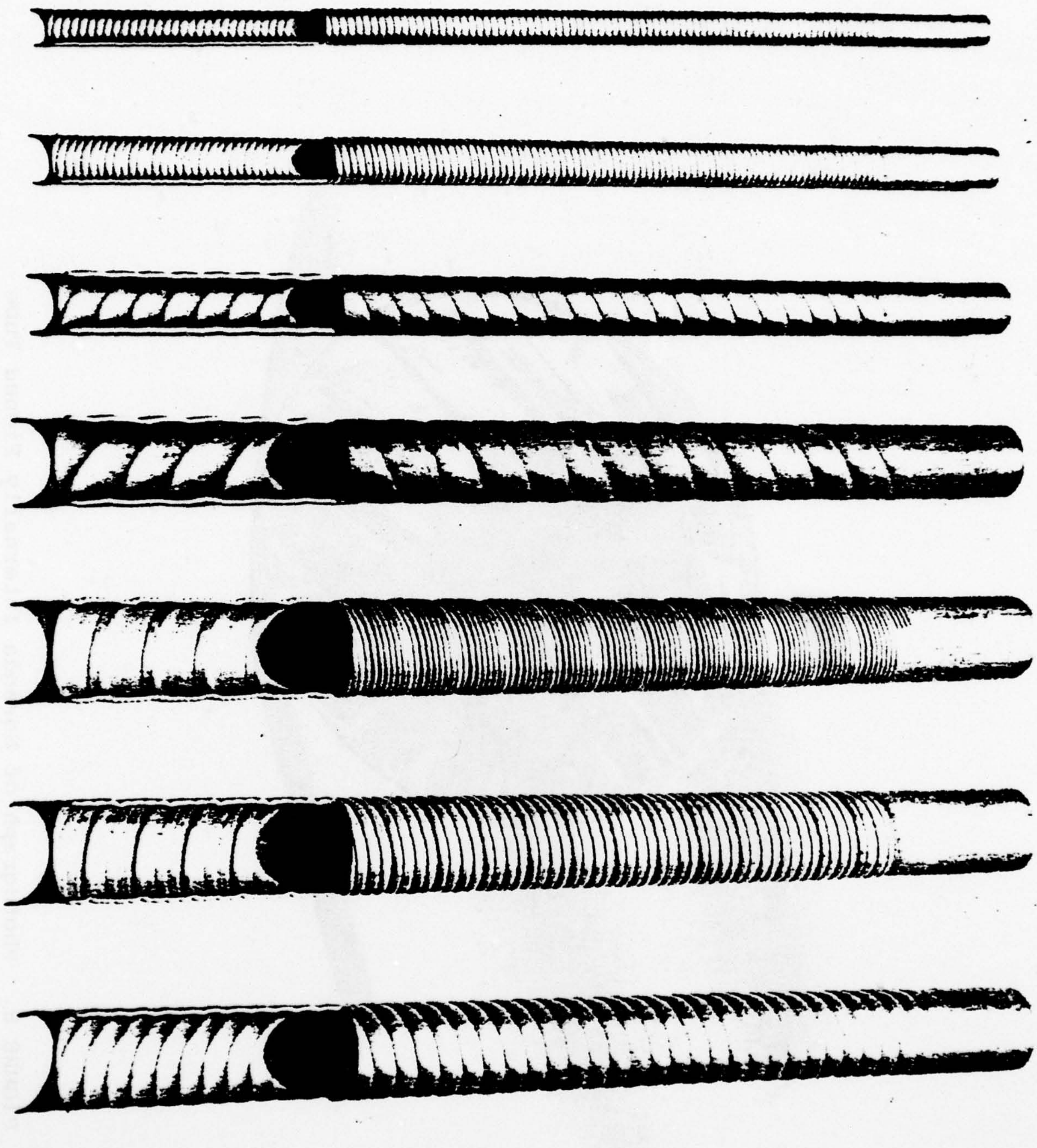


FIGURE 7. Photograph of Yorkshire Imperial Metals Enhanced Tubes [20]

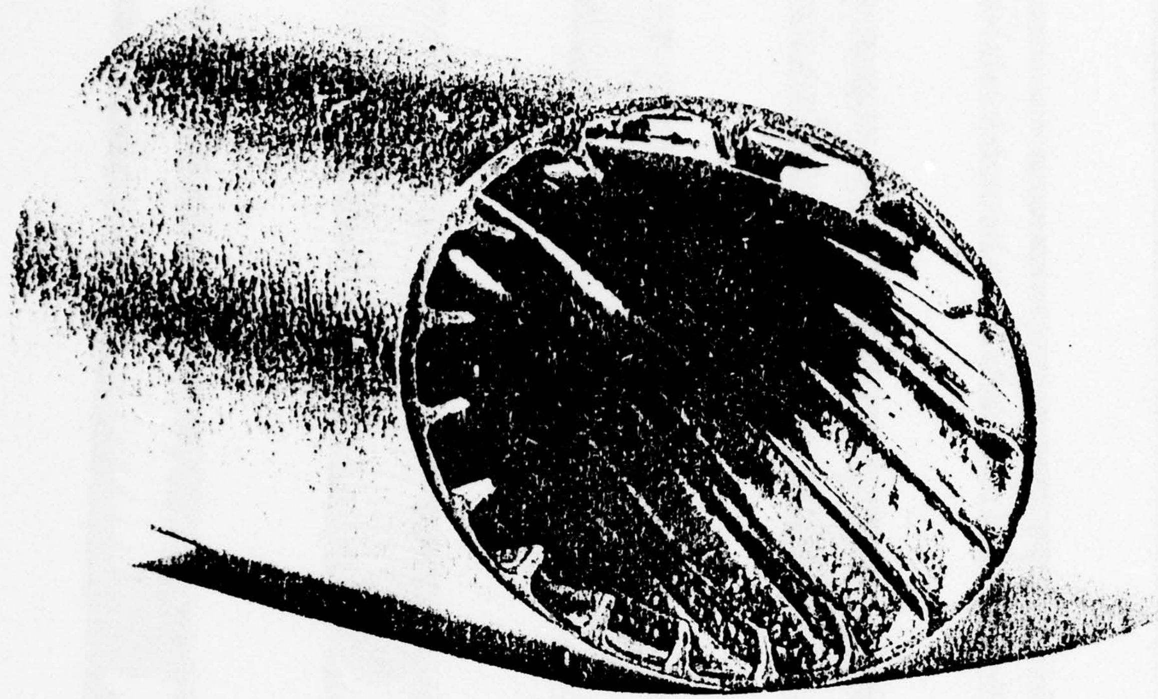


FIGURE 8. Photograph of Noranda Internally Finned Tube

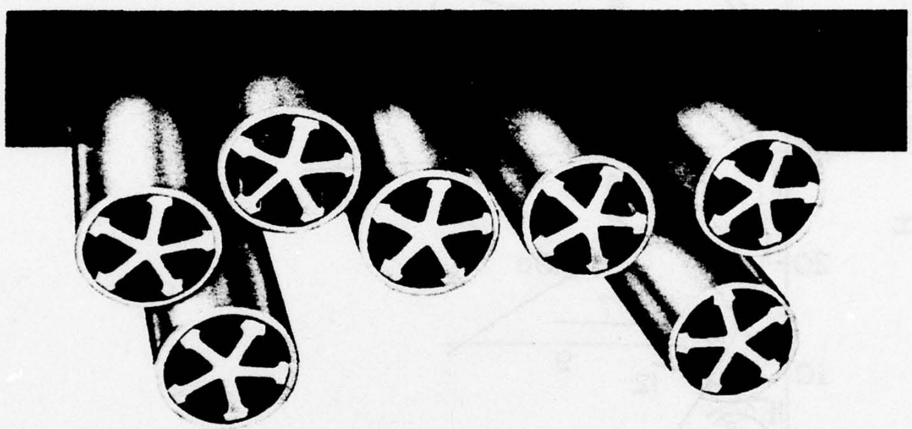
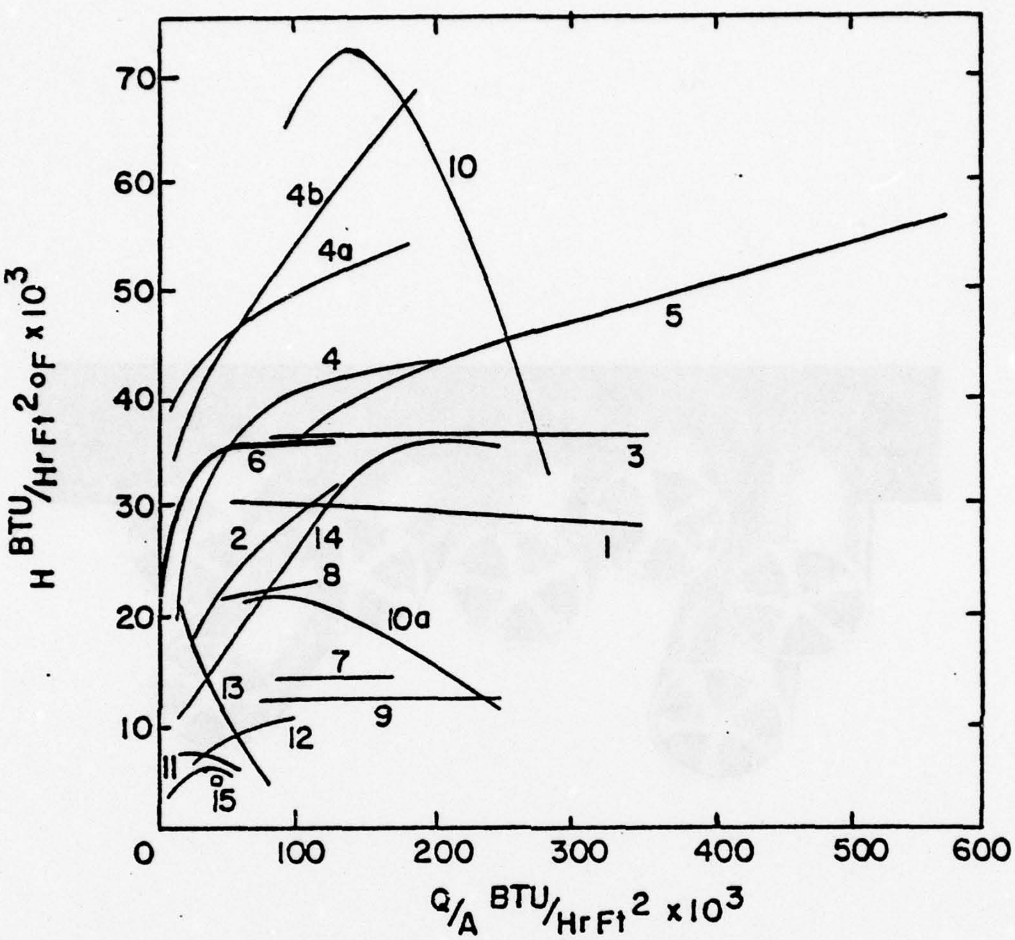


FIGURE 9. Photograph of Yorkshire Imperial Metals 'INNESTAR' Tubes [23]



1 KRAUSE [37]
2 HAMPSON & OZISIK [30]
3 WENZEL [38]
4 TANNER et al. [39]
5 LeFEVRE & ROSE [36]
6 PRESENT WORK
7 NAGLE [40]
8 GNAM [41]
9 FITZPATRICK [42]
10 SHEA & KRASE [43]
11 KIRSCHBAUM [44]
12 COSTAS [45]
13 WELCH & WESTWATER [7]
14 KAST [8]
15 FATICA & KATZ [5]

NOTE: Numbers in brackets refer to the original Bibliography of Graham [29].

FIGURE 10. Comparison of Dropwise Condensation Heat Transfer Coefficients as Presented by Graham [29]

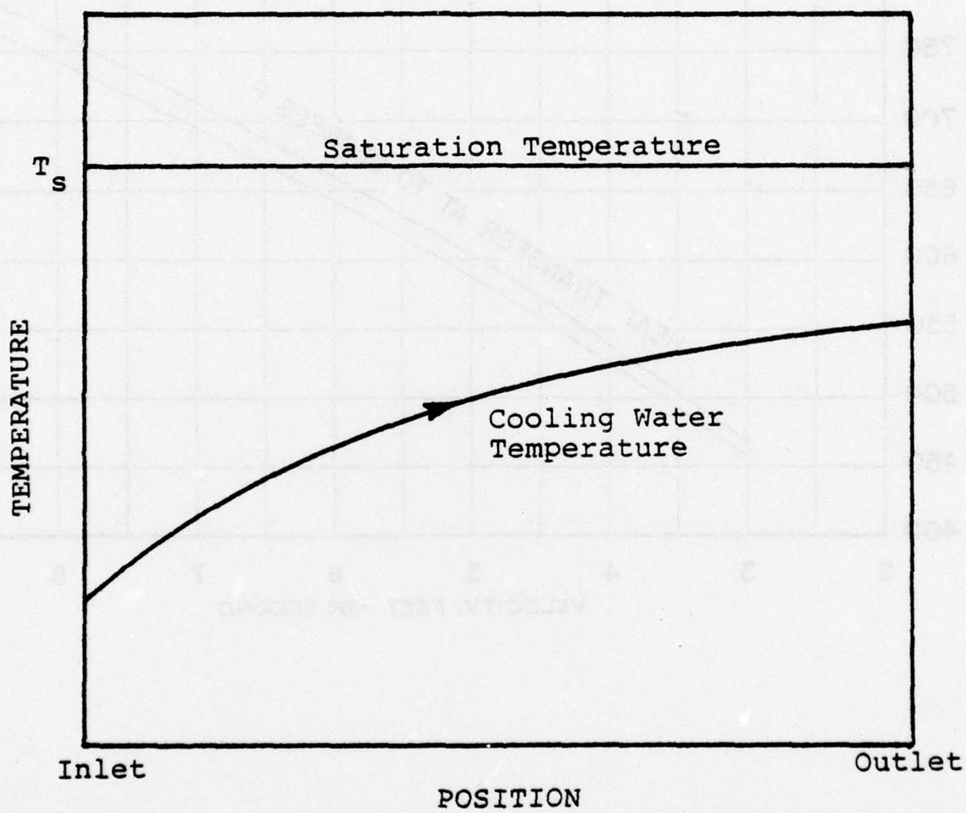


FIGURE 11. Schematic Representation of the Temperature Distribution in a Condenser

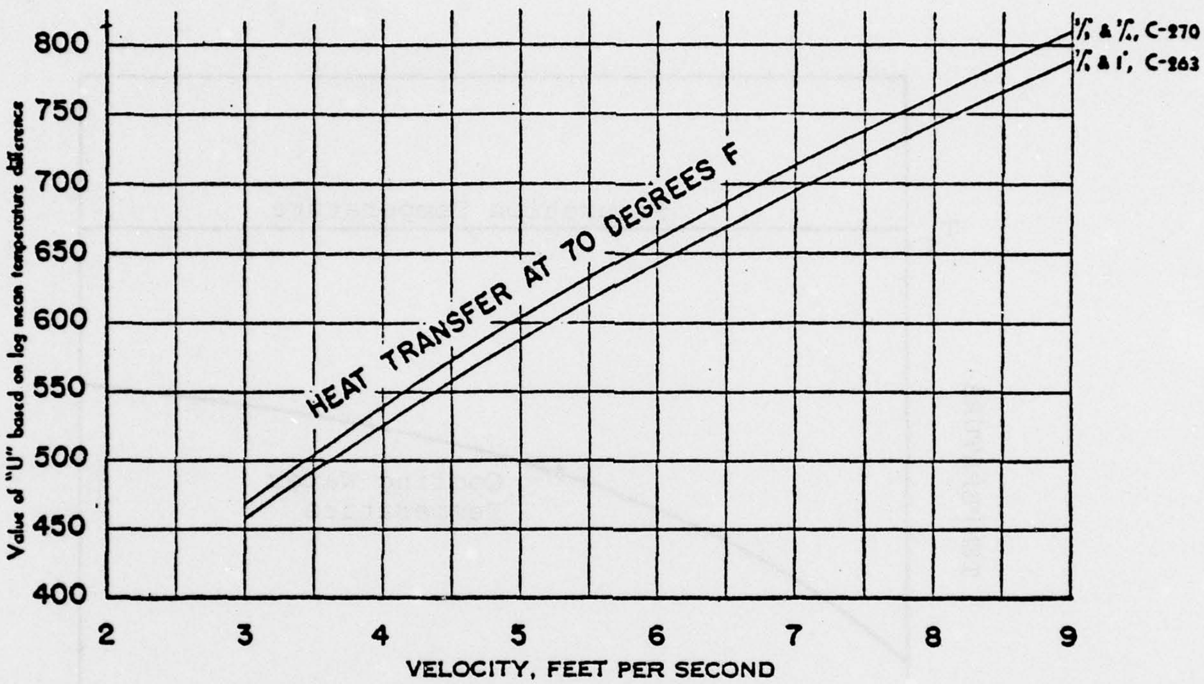


FIGURE 12. Dependence of Overall Heat Transfer Coefficient on Sea Water Velocity

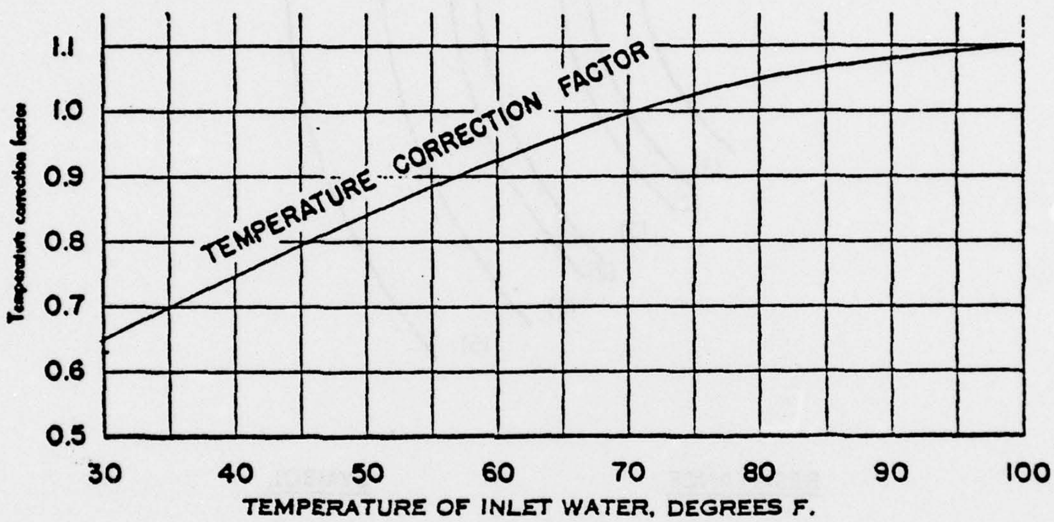
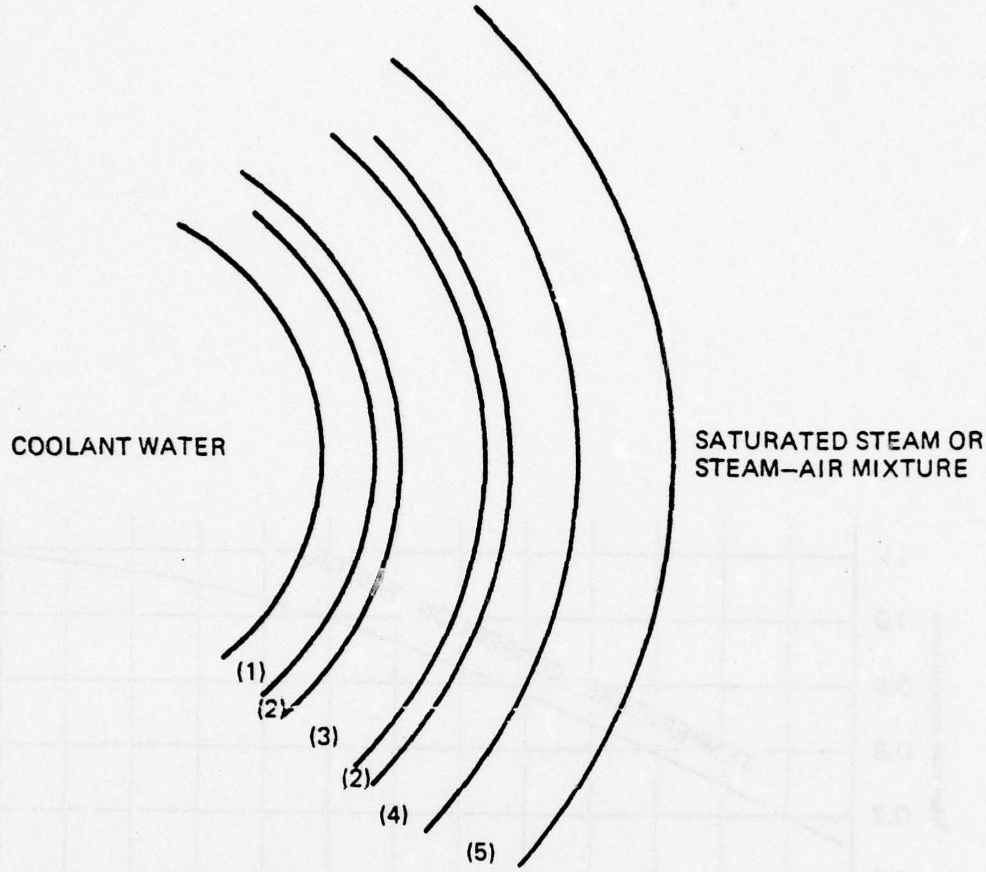
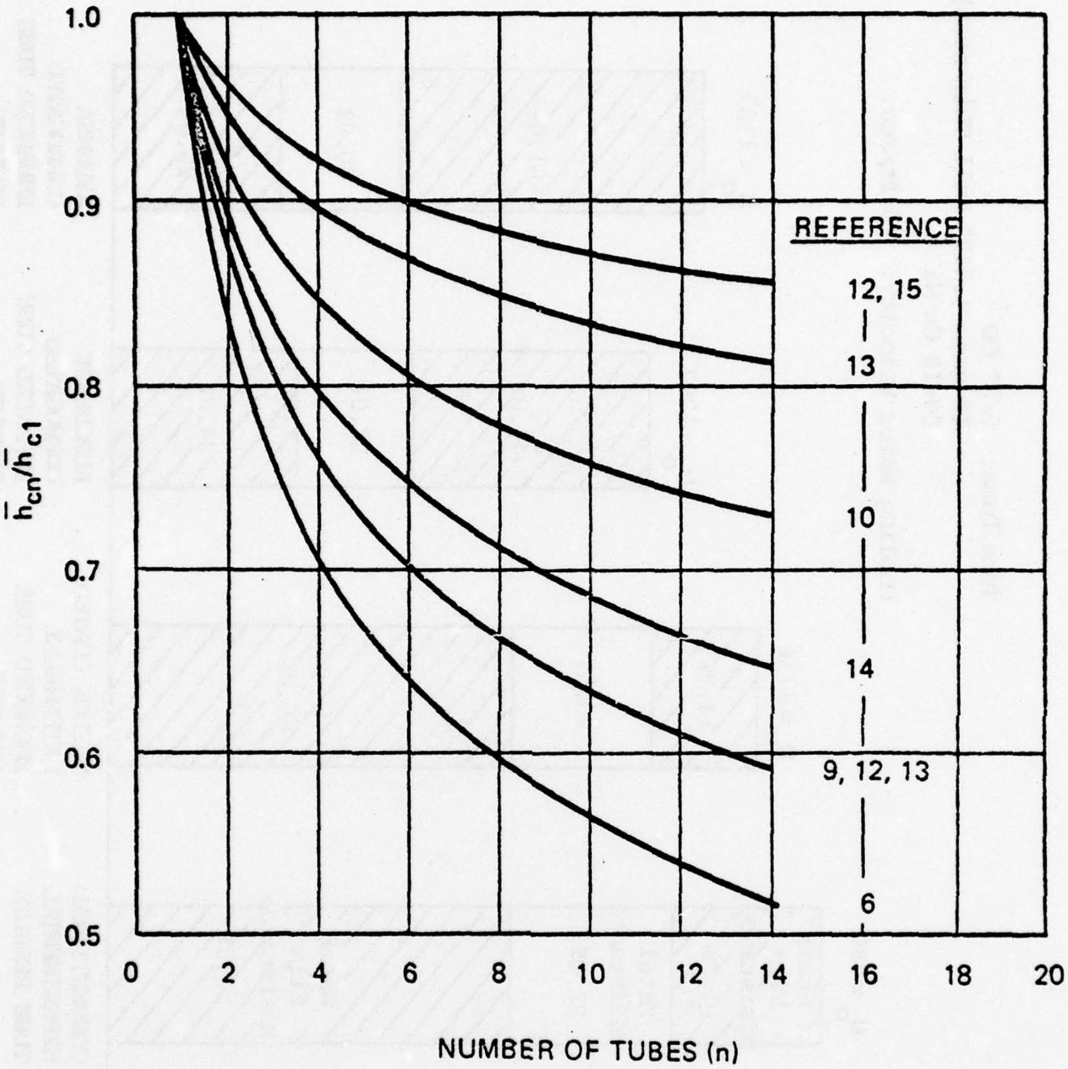


FIGURE 13. Temperature Correction Factor for Overall Heat Transfer Coefficient



RESISTANCE	SYMBOL
(1) CONVECTIVE FILM	R_i
(2) SOLIDS FOULING	R_{fo} & R_{fi}
(3) TUBE WALL	R_w
(4) CONDENSATE FILM	R_o
(5) NON CONDENSING GAS FILM	R_{nc}

FIGURE 14. Schematic Representation of Heat Transfer Resistances in a Condenser



NOTE: Reference numbers refer to the original Bibliography of Eissenberg [15].

FIGURE 15. Comparison of Experimental Data Showing the Effect of Condensate Rain on the Mean Condensate Film Heat Transfer Coefficient [15]

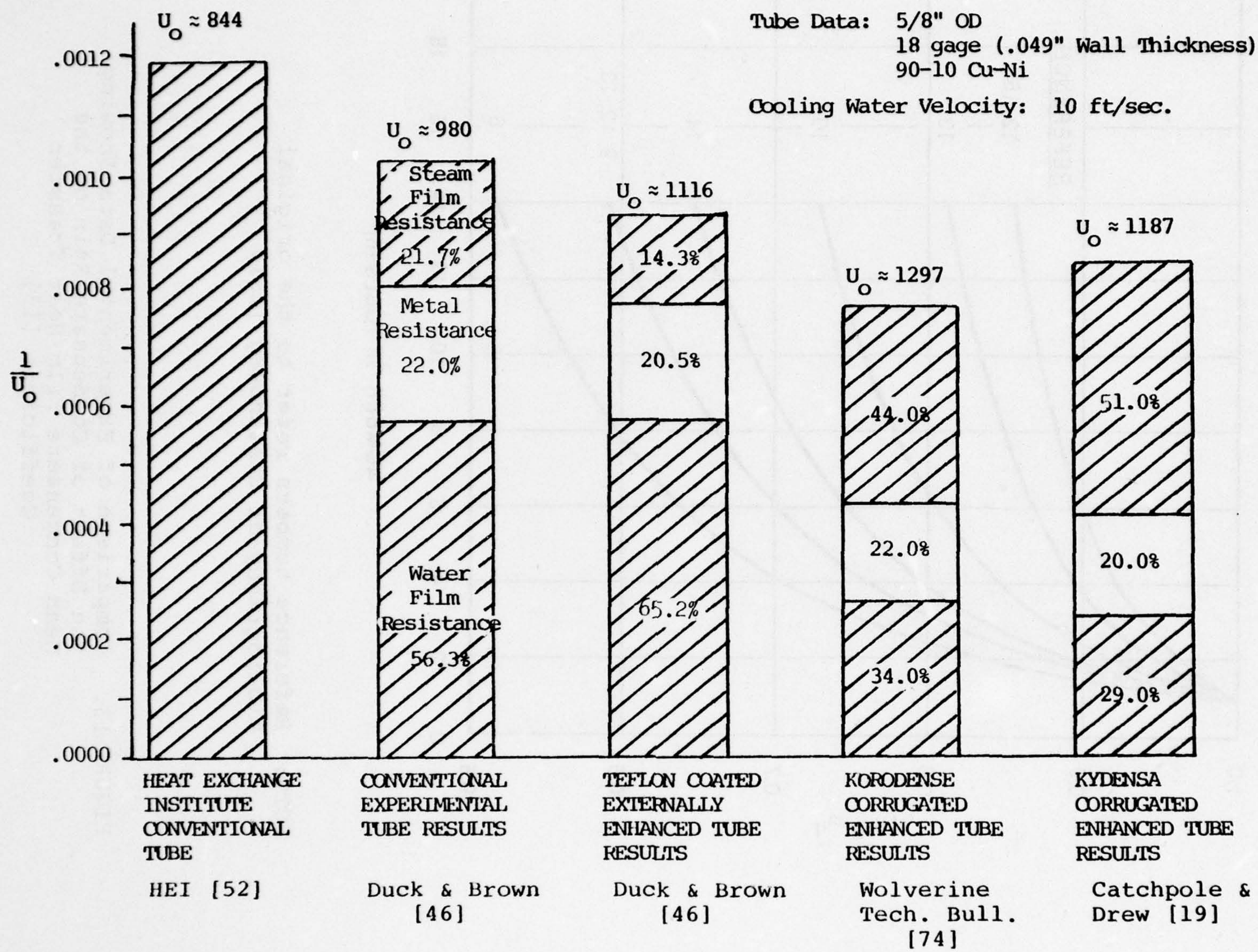


FIGURE 16. Comparison of Thermal Resistances for Various Tube Enhancement Techniques

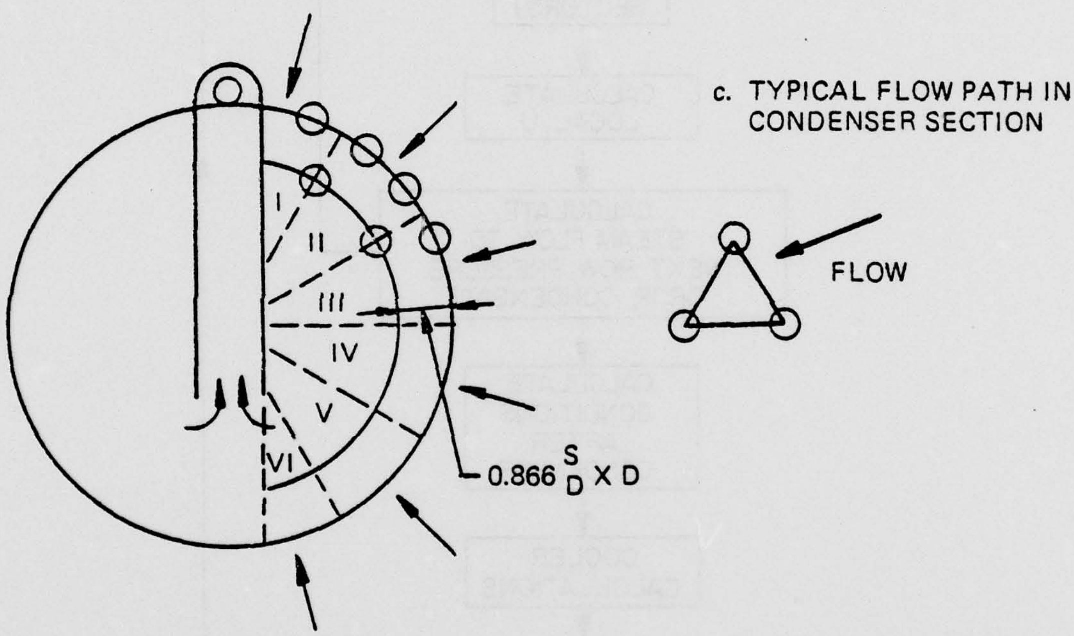


FIGURE 17. Schematic Representation of Tube Bundle Geometry Used in ORCON 1

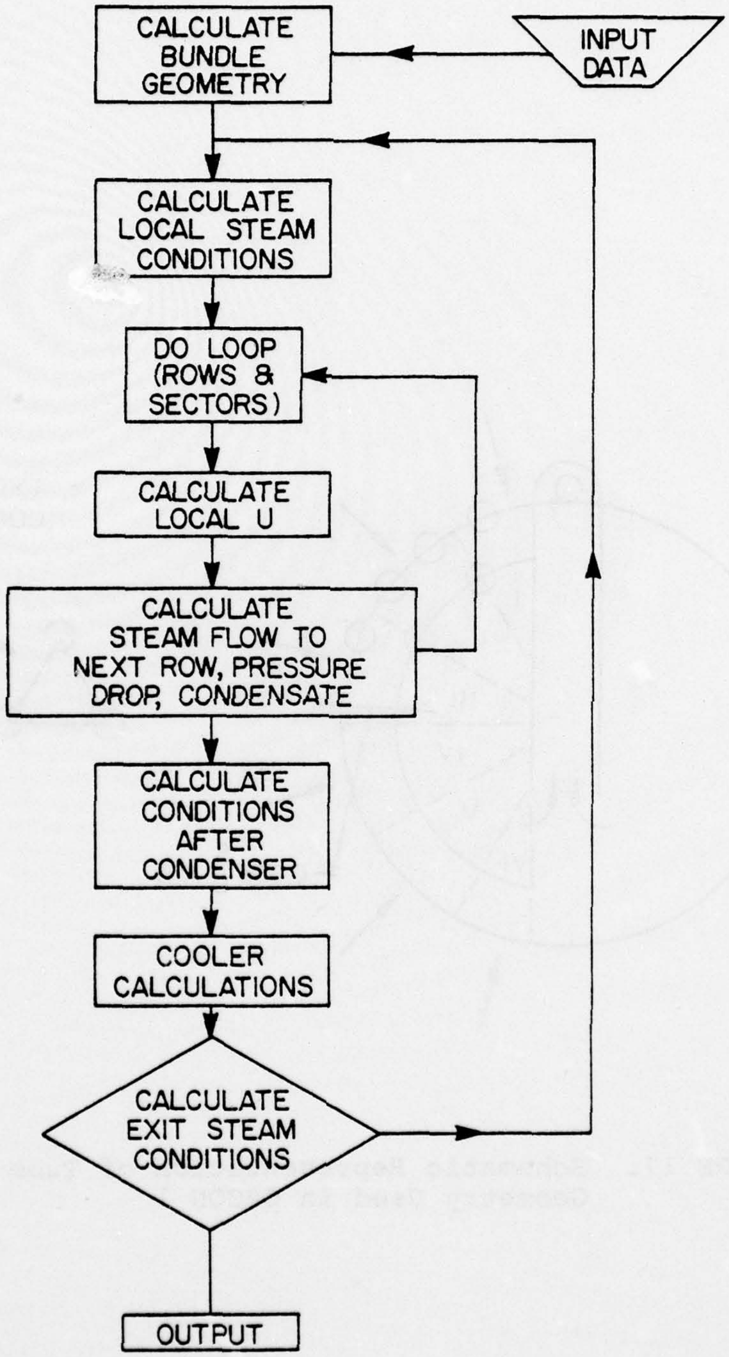
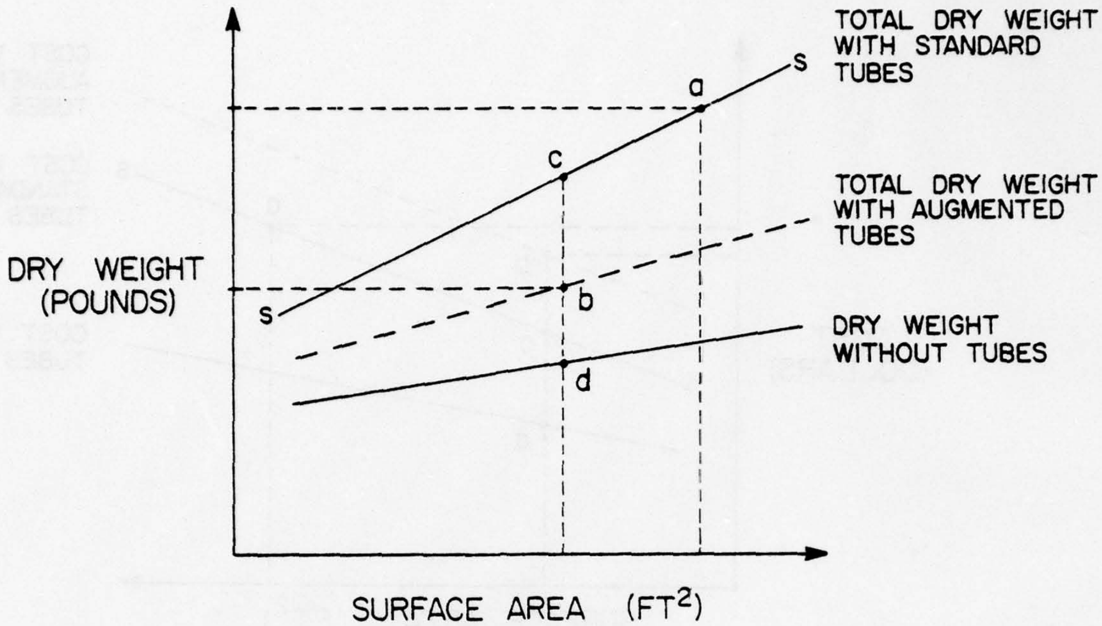


FIGURE 18. Computer Flow Chart of ORCON 1



1. Point a illustrates the square feet of surface area and total dry weight of a standard condenser for a specified heat load.
2. Point b illustrates the square feet of surface area and total dry weight of an augmented condenser for an equivalent heat load.
3. Augmented condenser dry weight (b) is obtained from a parametric estimate of standard condenser dry weight illustrated by the line S-S. For a condenser with condensing surface area given by point b, the total dry weight with standard tubes would be at point c; the dry weight without tubes would be at point d; and the total dry weight with augmented tubes would be at point b.
4. In this particular illustration, the dry weight (b) for the augmented condenser is less than the dry weight (a) for a standard condenser to handle the same heat load.

FIGURE 19. Schematic Illustration of Condenser Dry Weight Calculations

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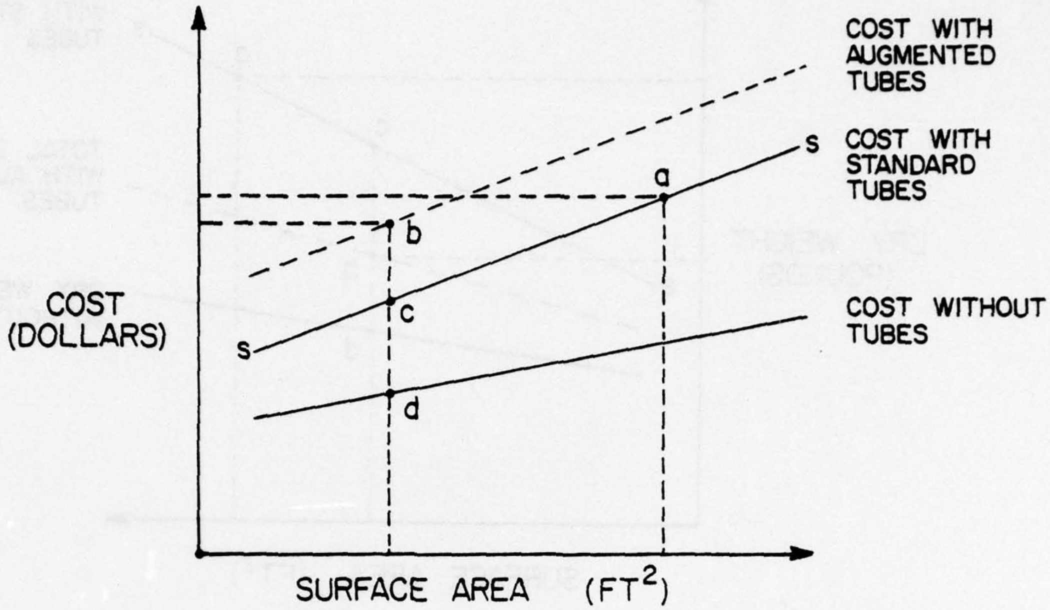


1. Point a Illustrates the square feet of surface area and cost of a standard condenser for a specified heat load.
2. Point b illustrates the square feet of surface area and cost of an augmented condenser for an equivalent heat load.
3. Augmented condenser cost (b) is obtained from a parametric estimate of standard condenser cost, illustrated by the line s-s. For a condenser with condensing surface area given by point b, the cost with standard tubes would be at point c, The cost without tubes would be at point d, and the cost with augmented tubes would be at point b.
4. In this particular illustration, the cost (b) for the augmented condenser is less than the cost (a) for a standard condenser to handle the same heat load.

FIGURE 20. Schematic Illustration of Condenser Cost Calculations

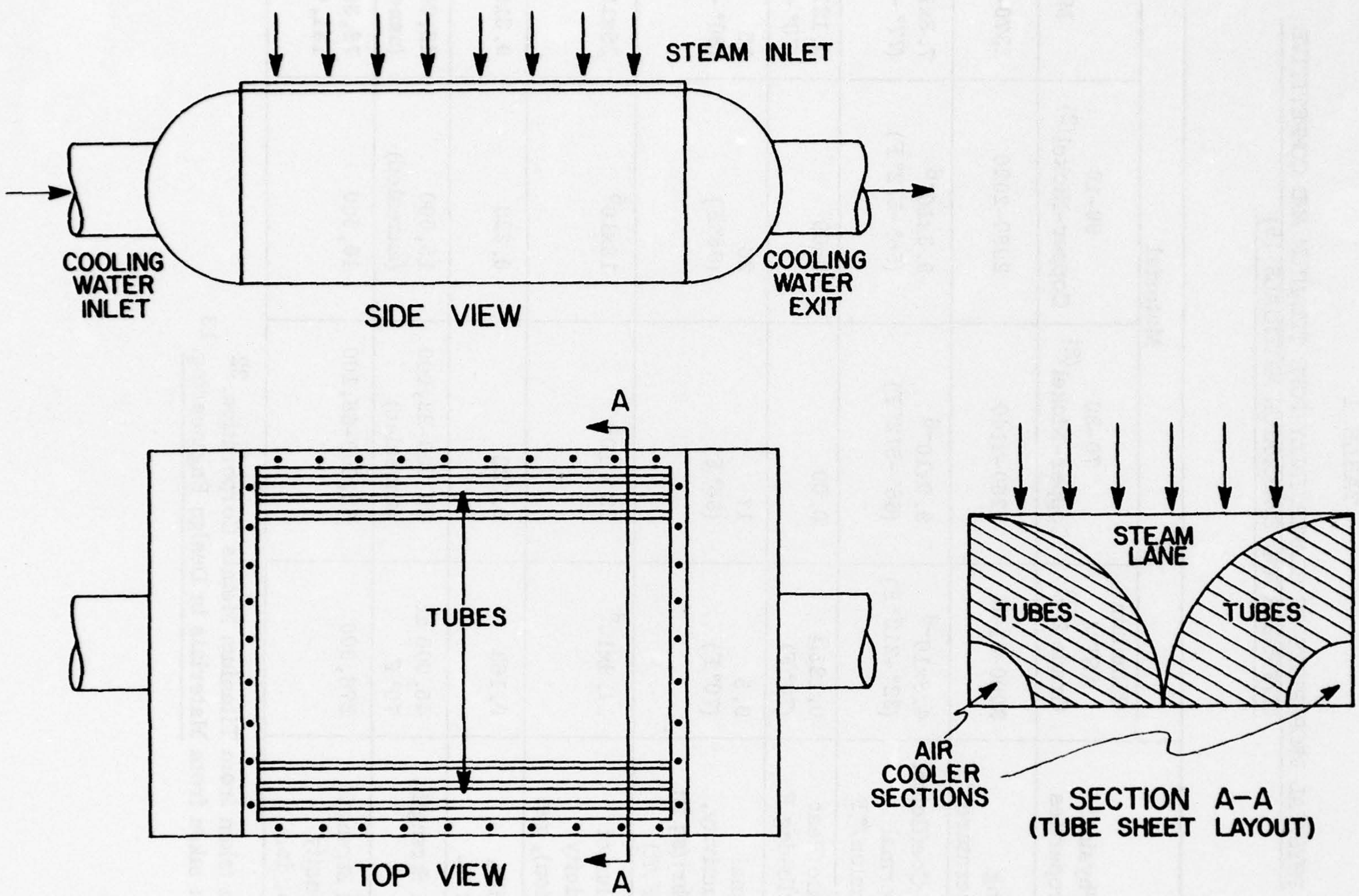


FIGURE 21. Schematic Configuration of Condenser in ORCON 1 Program

TABLE 1

PHYSICAL PROPERTIES OF COMMERCIALLY PURE TITANIUM AND COMPETITIVE CONDENSER AND EVAPORATOR MATERIALS [5]

Physical Properties	Material			
	Titanium ASTM Grade 2(1)	70-30 Copper-Nickel(2)	90-10 Copper-Nickel(2)	Monel(2)
Melting Temperature, °F	3000-3100	2260-2140	2080-2020	2370-2460
Mean Coefficient of Thermal Expansion/°F	4.8×10^{-6} (32°-212°F)	9.0×10^{-6} (68°-572°F)	9.3×10^{-6} (68°-572°F)	7.8×10^{-6} (77°-212°F)
Specific Heat BTU/lb-deg F	0.125@ (70°F)	0.09	0.09	0.127 (70°-750°F)
Thermal Conductivity, Btu/(hr)(sq ft)(deg F/ft)	9.5 (70°F)	17 (68°F)	26 (68°F)	15 (80°-212°F)
Modulus of Elasticity (tension), psi	14.9×10^6	22×10^6	18×10^6	26×10^6
Density, lb/cu in.	0.163	0.323	0.323	0.319
Yield Strength, psi	45,000 at 70°F	20,000-22,000 (annealed)	15,000 (annealed)	25,000-45,000 (annealed)
Yield Strength to Density Ratio, in.	276,000	62,000-68,100	46,500	78,300-141,000

[1]Data taken from Titanium Metals Corporation.[22]
[2]Data taken from Materials in Design Engineering.[13]

TABLE 2

PERFORMANCE COMPARISON OF TITANIUM TUBES VERSUS COPPER-NICKEL TUBES IN AN AIRCRAFT CARRIER CONDENSER [5]

Steam Conditions: 4 inches of Hg Absolute; 396,000,000 Btu per Hr Single Water Pass

	Condenser Material						
	90-10 Cu-Ni	Titanium	Titanium	Titanium	Titanium	Titanium	Titanium
Water Velocity in Tubes, fps	9.0	9.0	10.0	12.0	9.0(1)	9.0(1)	9.0(2)
No. of Tubes	6,540	6,640	6,220	5,600	5,400	5,930	9,030
Overall Tube Length, ft	15.0	15.0	15.0	15.0	19.25	17.25	15.0
Tube OD	5/8	5/8	5/8	5/8	5/8	5/8	1/2
Tube Wall Thickness, in.	0.049	0.022	0.022	0.022	0.022	0.022	0.020
Inlet Water Temperature, °F	75	75	75	75	75	75	75
Outlet Water Temperature, °F	94.9	91.0	90.4	89.3	94.7	93.0	95.2
Water Flow, gpm	39,900	49,400	51,400	55,500	40,200	44,000	39,200
Pressure Drop, ft of water	12.5	10.8	14.2	19.7	14.3	13.1	14.5
Condensing Surface, sq ft	15,800	16,100	15,100	13,550	16,800	16,400	16,300
Overall Heat-Transfer Coefficient Btu/(hr)(sq ft) (deg F)	624	590	623	682	590	590	613
Condenser Dry Weight, lb	85,000	58,340	57,800	57,050	58,670	58,540	58,310
Total Condenser Weight/Ship, lb	340,000	233,360	231,200	228,200	234,680	234,160	233,240
Weight Saved/Ship, lb	0	106,640	108,800	111,800	105,320	105,840	106,760
Weight Saved, %	0	31.3	32.0	32.9	31.0	31.2	31.4
Overall Tube Length, ft	15.0	15.0	15.0	15.0	19.25	17.25	15.0

1 Different tube length used.
2 Tube OD = 1/2 inch.

TABLE 3

CORRECTION FACTORS FOR TUBE MATERIAL AND WALL THICKNESS IN HEI METHOD OF CALCULATING THE OVERALL HEAT TRANSFER COEFFICIENT

Tube materials	Tube wall gage (BWG)		
	No. 18	No. 17	No. 16
Admiralty metal	1.0	0.98	0.96
Arsenical copper	1.0	.98	.96
Aluminum brass	.96	.94	.91
Muntz metal	.96	.94	.91
Aluminum bronze	.90	.87	.84
90–10 copper nickel	.90	.87	.84
70–30 copper nickel	.83	.80	.76

TABLE 4

CONDENSER DESIGN CONSIDERATIONS

<u>Tubing Considerations</u>

Tube diameter
Wall thickness
Tube material
Weldability
Strength
Coatings
Vibration (need for support plates)
Flexural rigidity
Tube geometry
Tube length

<u>Steam Side Considerations</u>

Steam velocity
Noncondensible gasses
Pressure drop
Condensate inundation effects

<u>Baffling</u>

Steam impingement on tubes
Tube spacing
Steam flow lane design
Fouling due to contaminates
Steam flow relative to condensate flow direction
Air vent location

<u>Sea Water Side Considerations</u>

Sea water velocity
Turbulence
Pressure drop
Erosion
Corrosion
Mechanical fouling
Biological fouling
Tube cleaning
External pressure
Tube sheet strength and corrosion resistance

<u>System Considerations</u>

Turbine design characteristics
Scoop injection design
Pressure environment
Machinery layout
Circulating pumps
Condensate pumps
Air removal equipment

<u>Heat Transfer Considerations</u>

Condensing coefficient
Convective coefficient
Tube thermal conductivity
Noncondensible gas resistance
Fouling resistances
Internal tube enhancement factors
External tube enhancement factors

<u>Manufacturing Considerations</u>

Condenser fabrication considerations (Producibility and cost)
Tubing fabrication considerations (Producibility and cost)
Material (Availability and cost)
Tube coating cost (Drop wise)

<u>Management Considerations</u>

Life cycle cost
Reliability
Maintainability
Flexibility

TABLE 5

INPUT DATA FOR ORCON 1

1. Case identification

2. Tube factors; number, spacing, length, wall thickness, material, etc.

3. Coolant factors; flow rate, velocity, inlet temperature, etc.

4. Steam factors; flow rate, inlet saturation temperature, etc.

5. Type of non-condensable gas present; flow rate.

6. Geometry control factors; number of sectors, etc.

7. Exit steam fraction desired.

8. Various control flags needed to control flow of program.

TABLE 6

STRUCTURE OF A COMPUTER DESIGN METHOD

1. Rating program
2. Input program
3. Physical/thermodynamics properties program
4. Configuration (design) modification program
5. Mechanical design program
6. Costing program
7. Optimization program
8. Systems performance program
9. Monitor program

TABLE 7

DESCRIPTION OF TEST CASES

CASE	TUBING DESCRIPTION	TUBING MATERIAL	O.D. (in)	WALL THICKNESS (in)	MODE OF CONDENSATION
Std	Plain	Cu/Ni	0.625	0.049	Film
1	Corrugated (a)	Cu/Ni	0.624	0.049	Film
2	Corrugated (a)	Cu/Ni	0.749	0.049	Film
3	Corrugated (a)	Cu/Ni	0.998	0.049	Film
4	Plain	Cu/Ni	0.625	0.049	Dropwise
5	Plain	Titanium	0.625	0.022	Film
6	Plain	Titanium	0.625	0.022	Film
7	Plain	Titanium	0.625	0.022	Film
8	Plain	Titanium	0.625	0.022	Dropwise
9	Corrugated (a)	Titanium	0.624	0.022	Film
10	Corrugated (a)	Titanium	0.749	0.022	Film
11	Corrugated (a)	Titanium	0.998	0.022	Film
12	Spiralled Internal Fins (b)	Cu/Ni	0.749	0.046(c)	Film
13	Straight Internal Fins (b)	Cu/Ni	0.750	0.033(c)	Film
14	Spiralled Internal Fins (b)	Cu/Ni	0.749	0.046(c)	Dropwise
15	Straight Internal Fins (b)	Cu/Ni	0.750	0.033(c)	Dropwise

(a) Wolverine Tube Div., Univ. Oil Products [74]

(b) Noranda Metals Industries [25]

(c) Wall thickness for finned tubes is based on equivalent coolant flow area

TABLE 8

TEST CASE INPUT DATA FOR ORCON 1

Case	Number of Tubes	Tube Length (ft)	Tube Thermal Conductivity (BTU/hr ft F)	Sea Water Velocity (ft/sec)	Inside Enhancement Factor	Outside Enhancement Factor	ΔP Ratio	$\dot{m}$ Ratio
Std	6,612	14.80	26.0	9.0	1.0	1.14	1.0	1.0
1	11,362	8.54	26.0	5.27	2.83(a)	1.11(a)	1.0	1.0
2	7,414	10.97	26.0	5.27	2.83(a)	1.22(a)	1.0	1.0
3	3,856	15.91	26.0	5.29	2.83(a)	1.45(a)	1.0	1.0
4	6,612	14.80	26.0	9.0	1.0	11.4(c)	1.0	1.0
5	5,440	16.66	9.5	9.0	1.0	1.14	1.0	1.0
6	6,120	14.87	9.5	12.0	1.0	1.14	1.5	1.5
7	6,528	13.25	9.5	15.0	1.0	1.14	2.0	2.0
8	6,528	13.25	9.5	15.0	1.0	2.85(c)	2.0	2.0
9	9,608	10.14	9.5	5.11	2.83(a)	1.11(a)	1.0	1.0
10	6,468	12.60	9.5	5.15	2.83(a)	1.22(a)	1.0	1.0
11	3,488	17.60	9.5	5.20	2.83(a)	1.45(a)	1.0	1.0
12	5,780	14.13	26.0	6.63	1.60(b)	1.14	1.0	1.0
13	4,932	16.56	26.0	7.17	1.50(b)	1.14	1.0	1.0
14	5,780	14.13	26.0	6.63	1.60(b)	11.4(c)	1.0	1.0
15	4,932	16.56	26.0	7.17	1.50(b)	11.4(c)	1.0	1.0

(a) Wolverine Tube Div., Univ. Oil Products [74]

(b) Noranda Metals Industries [25]

(c) Hannemann and Mikic [45]

TABLE 9

ORCON 1 CALCULATED VALUES OF STEAM FLOW RATE

Sensitivity to Non-condensable Gas

NC Gas Flow Rate (lb/hr)	Steam Flow Rate (lbm/hr)
11.25	303,300
22.50	303,200 (Std. design)
45.00	303,000
67.50	302,800
225.00	150,800

Sensitivity to Condensate Flooding (No Baffles)

Flooding Equation	Steam Flow Rate (lbm/hr)
Eq (A-3)	303,200 (Std. design)
Eq (A-2)	355,500

Sensitivity to Baffles and Condenser Height

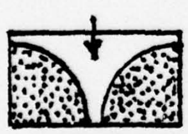
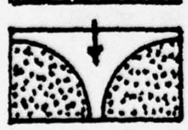
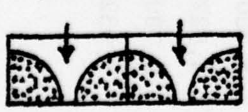
Condenser Geometry	Baffles/Bundle	Steam Flow Rate (lbm/hr)
	0	327,132
	1	330,634
	1	345,564
	1	356,548

TABLE 10

HEAT LOAD COMPARISON AT CONSTANT PUMPING POWER

	$\Delta\dot{Q}$ (%)	U_o (BTU/hr ft^2 °F)	ΔT_{LM} (°F)	A (ft^2)
Std	----	507	41	16,009
1	+11.0	584	40	15,857
2	+22.4	658	39	15,937
3	+33.7	737	38	16,032
4	+30.1	712	38	16,009
5	- 3.6	523	42	14,832
6	+ 6.3	543	44	14,892
7	+ 7.2	556	46	14,151
8	+35.5	724	44	14,151
9	+14.9	608	40	15,916
10	+23.3	663	39	15,979
11	+33.4	734	38	16,046
12	+ 5.7	543	41	16,001
13	+ 9.9	570	40	16,033
14	+42.7	809	37	16,001
15	+50.7	874	36	16,033

NOTE: Baseline heat load equals 335.33×10^{-6} $\frac{BTU}{HR}$

TABLE 11

VOLUME, WEIGHT, AND COST COMPARISON AT EQUIVALENT HEAT LOAD

Percentage Deviation From Standard Design

Case	Tube Bundle Volume	Condenser Volume	Condenser Weight Dry	Condenser Weight Wet	Condenser Cost	Sea Water Mass Flow Rate
1	-10.7	+10.4	- 6.1	+ 5.5	+15.4	- 9.9
2	- 0.3	+11.3	-11.2	- 1.5	- 0.6	-18.2
3	+18.6	+18.1	-15.3	- 6.7	- 9.9	-25.1
4	-22.5	-24.5	-15.8	-18.5	-13.0 **	-23.1
5	- 2.4	- 5.9	-32.4	-26.3	+21.4	+ 3.7
6	-12.8	-14.2	-35.6	-29.7	+16.4	+41.2
7	-16.4	-15.1	-37.5	-30.6	+14.3	+86.6
8	-35.8	-36.7	-44.0	-41.3	- 0.5 **	+47.6
9	-13.9	- 8.9	-35.7	-26.3	+35.5	-13.0
10	- 1.3	+ 4.0	-37.8	-24.7	+17.5	-18.7
11	+17.8	+13.8	-40.1	-26.8	+ 5.6	-25.0
12	+14.6	+18.1	- 5.2	+ 1.9	+30.6	- 5.3
13	+10.5	+ 7.9	-17.1	-11.1	+ 4.1	- 9.0
14	-17.2	-17.6	-21.6	-20.2	+ 8.0 **	-29.9
15	-19.1	-23.2	-31.0	-29.8	-12.3 **	-33.6

Standard Design

Heat Load ------------	335.3×10^6	BTU/hr
Sea Water Mass Flow Rate	20.7×10^6	lbm/hr
Tube Bundle Volume ----	617	ft^3
Condenser Volume ------	996	ft^3
Dry Weight ------------	83,900	lbm
Wet Weight ------------	107,000	lbm

** The cost of the dropwise promoter has been neglected.

TABLE 12

REQUIRED SURFACE AREA, COOLANT FLOW RATE, NUMBER OF TUBES OF LENGTH L, AND BUNDLE RADIUS R_B TO HANDLE SPECIFIED HEAT LOAD

Case	$U\Delta T_{LM}$	A	$\dot{m}_C [x10^6]$	N	L	R_B
Std	20,946	16,009	20.695	6,612	14.797	5.117
1	23,482	14,280	18.639	10,232	8.543	6.332
2	25,706	13,045	16.937	6,069	10.965	5.921
3	27,937	12,003	15.499	2,887	15.907	5.378
4	27,233	12,313	15.918	5,085	14.797	4.504
5	21,792	15,388	21.470	5,644	16.663	4.767
6	23,924	14,016	29.217	5,760	14.871	4.767
7	25,401	13,201	38.613	6,090	13.248	4.942
8	32,111	10,443	30.544	4,817	13.248	4.329
9	24,208	13,852	18.014	8,362	10.140	5.720
10	25,871	12,962	16.820	5,247	12.602	5.501
11	27,852	12,040	15.528	2,617	17.600	5.098
12	22,119	15,160	19.606	5,476	14.127	5.604
13	22,975	14,595	18.838	4,490	16.556	5.090
14	29,875	11,224	14.517	4,054	14.127	4.765
15	31,517	10,640	13.732	3,273	16.556	4.355

APPENDIX A: HEAT TRANSFER AND PRESSURE DROP CORRELATIONS UTILIZED FOR THERMAL ANALYSIS

1. Condensate Film Coefficient

For a single horizontal tube, the mean heat transfer coefficient during laminar film condensation is usually expressed by the Nusselt relationship:

$$h_{oN} = 0.725 \left\{ \frac{k_f^3 \rho_f^2 g h_{fg}}{\mu_f D_o \Delta T_f} \right\}^{1/4} . \qquad \text{(A-1)}$$

This value can be adjusted to take into account condensate flooding from other tubes. Several alternative methods are available as given below.

(a) Method of Withers and Young [12,13]

The mean effective coefficient for n tubes in a vertical row is calculated by the expression:

$$\overline{h}_{oN} = 0.725 \; C_n \left\{ \frac{k_f^3 \rho_f^2 g h_{fg}}{n \mu_f D_o \Delta T_f} \right\}^{1/4} \qquad \text{(A-2)}$$

where

$$C_n = Z[n]^{PW} .$$

Values of the tube enhancement factor Z, and the exponent PW for plain and corrugated tubes are provided in Reference 74.

(b) Method of Oak Ridge National Laboratory [53,54]

The mean effective coefficient for n tubes in a vertical row is calculated by:

$$\bar{h}_{on} = E_o \, h_{oN} \, F(n) \qquad \text{(A-3)}$$

where

h_{oN} = film coefficient for a single tube, equation (A-1)

E_o = enhancement factor on the outside, and

$F(n)$ = correction factor to account for condensate inundation

In the above expression,

$$F(n) = 0.6 \, F_d + (1 - 0.5647 \, F_d) \, n^{-0.20} \ . \qquad \text{(A-4)}$$

F_d is an input value which varies between 0 and 1, and is related to tube spacing and orientation as shown below.

Tube Lattice	F_d
In line	0
Staggered, S/D > 1.40	0
S/D = 1.33	0.5
S/D < 1.25	1.0 .

2. Convective Film Coefficient on Tubeside

On the cooling water side of the condenser tubes, the well-known Colburn analogy is utilized:

$$\frac{h_i D_i}{k} = E_i \, (0.023) \, Re_i^{0.8} \, Pr^{1/3} \tag{A-5}$$

where

E_i = internal enhancement factor.

3. Effective Heat Transfer Coefficient Due to Non-Condensable Gas

An effective gas coefficient is calculated using the Colburn analogy applied to mass transfer of steam across an air barrier. An excellent presentation of this theory with equations is provided by Eissenberg [15].

4. Shellside Pressure Drop

The following equations were utilized as recommended by Eissenberg and Noritake [54]:

$$\Delta p_o = \frac{2 \, f_o \, G^2 \, N}{\rho_v \, g_c}, \text{ and} \tag{A-6}$$

$$f_o = 0.102 + \frac{52.2}{Re_v} \tag{A-7}$$

5. Tubeside Pressure Drop

On the water side of the tube, the following equation was used:

$$\Delta p_i = 2 f_i \frac{L}{D_i} \frac{\rho v_c^2}{g_c} . \qquad \text{(A-8)}$$

In the above equation, the friction factor, f_i, varies with the type of tube as shown below.

Tube Type	Friction Factor	
Plain [76]:	$f_i = \dfrac{0.046}{Re_i^{0.2}}$	(A-9)
Corrugated [74]:	$f_i = \dfrac{0.129}{Re_i^{0.154}}$	(A-10)
Internally Finned:		
(a) Straight [25]	$f_i = \dfrac{0.406 (b/De)^{0.16}}{Re_e^{0.39}}$	(A-11)
(b) Spiral [25]	$f_i = \dfrac{0.614}{Re_e^{0.39} (P/De)^{0.20}}$	(A-12)

APPENDIX B: CALCULATIONAL PROCEDURE TO OBTAIN CONDENSER VOLUME

Each condenser consists of two tube bundles, and the condenser configuration is based on a tube bundle radius (R_B) as shown in Figure 21. The tube length (L_B) for each condenser results from the equations utilized for the determination of the desired pressure drop. The volume calculation is as follows:

1. Tube bundle dimensions

   Width $W_B = 2R_B$

   Height $H_B = R_B$

   Length $L_B = L_B$

   Volume $V_B = 2R_B^2 L_B$ (includes steam lane)

2. Header dimensions (each header)

   Width $W_H = 2R_B$

   Height $H_H = R_B$

   Depth $D_H = 1/2\ R_B$ (side area: $1/2\ \pi \frac{R_B^2}{4}$)

   Volume $V_H = 1/4\ \pi\ R_B^3$ (side area times width)

3. Condenser volume (tube bundle and two headers)

$$V_c = V_B + 2V_H$$

$$V_c = 2R_B^2 L_B + 1/2\ \pi\ R_B^3$$

The tube bundle radius (R_B) for each augmentation design is hand calculated using the condenser bundle geometry algorithm from ORCON 1. The number of tubes for this calculation is based on the results of Table 12 which shows the required surface area in order to handle the specified heat load.

APPENDIX C: CALCULATIONAL PROCEDURE TO OBTAIN CONDENSER WEIGHT

1. Assume the weight of salt water at 64.043 lb/ft^3.
2. Obtain standard tube surface area (ft^2) per lineal foot (ft) from manufacturers data.
3. Obtain augmented tube surface area (ft^2) per lineal foot (ft) from manufacturers data.
4. Obtain augmented tube cross-sectional flow area from manufacturers data.
5. Obtain augmented tube weight (lb.) per foot (ft) from manufacturers data.
6. Obtain standard tube weight (lb.) per foot (ft) from manufacturers data.
7. Obtain header volume from the volume calculations.
8. Obtain number of augmented tubes from the augmented condenser design.
9. Obtain length of augmented tubes from the augmented condenser design.
10. Calculate number of square feet (ft^2) of condensing surface (A).

$$\text{Step } 10 \ (ft^2) = 3 \left(\frac{ft^2}{ft}\right) \cdot 8 \cdot 9 \ (ft)$$

11. Graphically obtain the dry weight of a standard condenser from the parametric curve of weight versus square feet (ft^2) of condensing surface (A).

12. Calculate total tube weight in the standard condenser.

$$\text{Step } (12) \ (\text{lb}) = \frac{(10) \ (\text{ft}^2)}{(2) \ (\frac{\text{ft}^2}{\text{ft}})} \cdot (6) \ (\frac{\text{lb}}{\text{ft}})$$

13. Calculate dry weight of the standard condenser without tubes.

$$\text{Step } (13) \ (\text{lb}) = (11) \ (\text{lb}) - (12) \ (\text{lb})$$

14. Calculate total tube weight in the augmented condenser.

$$\text{Step } (14) \ (\text{lb}) = \frac{(10) \ (\text{ft}^2)}{(3) \ (\frac{\text{ft}^2}{\text{ft}})} \cdot (5) \ (\frac{\text{lb}}{\text{ft}})$$

Alternately:

$$\text{Step } (14) \ (\text{lb}) = (8) \cdot (9) \ (\text{ft}) \cdot (5) \ (\frac{\text{lb}}{\text{ft}})$$

15. Calculate dry weight of the augmented condenser.

$$\text{Step } (15) \ (\text{lb}) = (13) \ (\text{lb}) + (14) \ (\text{lb})$$

16. Calculate weight of salt water in the augmented condenser tubes.

$$\text{Step } (16) \ (\text{lb}) = (1) \ (\frac{\text{lb}}{\text{ft}^3}) \cdot (4) \ (\text{ft}^2) \cdot (8) \cdot (9) \ (\text{ft})$$

17. Calculate weight of salt water in the augmented condenser headers.

Step 17 (lb) = 2.0 1 $\left(\frac{lb}{ft^3}\right)$ · 7 (ft^3)

18. Calculate weight of the augmented condenser.

Step 18 (lb) = 15 (lb) + 16 (lb) + 17 (lb)

APPENDIX D: CALCULATIONAL PROCEDURE TO OBTAIN CONDENSER COST

1. Obtain augmented tube cost ($$) per lineal foot (ft) from manufacturers data.
2. Obtain augmented tube surface area (ft^2) per lineal foot (ft) from manufacturers data.
3. Obtain number of augmented tubes from the augmented condenser design.
4. Obtain length (ft) of augmented tubes from the augmented condenser design.
5. Obtain standard tube surface area (ft^2) per lineal foot (ft) from manufacturers data.
6. Obtain standard tube cost ($$) per lineal foot (ft) from manufacturers data.
7. Calculate number of square feet (ft^2) of condensing surface (as) required.

$$\text{Step } 7 \ (ft^2) = 2 \left(\frac{ft^2}{ft}\right) \cdot 3 \cdot 4 \ (ft)$$

8. Graphically obtain the cost of a standard condenser from the parametric curve of cost ($$) versus square feet (ft^2) of condensing surface (A).
9. Calculate total tube cost ($$) in the standard condenser.

$$\text{Step } 9 \ (\$\$) = \frac{7 \ (ft^2)}{5 \left(\frac{ft^2}{ft}\right)} \cdot 6 \left(\frac{\$\$}{ft}\right)$$

10. Calculate cost of the standard condenser without tubes.

$$\text{Step } (10)\ (\$\$) = (8)\ (\$\$) - (9)\ (\$\$)$$

11. Calculate tube cost in the augmented condenser.

$$\text{Step } (11)\ (\$\$) = \frac{(7)\ (\text{ft}^2)}{(2)\ (\frac{\text{ft}^2}{\text{ft}})} \cdot (1)\ (\frac{\$\$}{\text{ft}})$$

Alternately:

$$\text{Step } (11)\ (\$\$) = (3) \cdot (4)\ (\text{ft}) \cdot (1)\ (\frac{\$\$}{\text{ft}})$$

12. Calculate cost of the augmented condenser.

$$\text{Step } (12)\ (\$\$) = (10)\ (\$\$) + (11)\ (\$\$)$$

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