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ABSTRACT

A methodology is developed for calculating the heat transfer coefficient on the surface of a vertical fluted condenser tube based upon condensate film thickness. A marine condenser sizing procedure is demonstrated using this methodology. A comparison is made between a fluted tube condenser and a comparable smooth tube horizontal condenser. Alternative materials are considered and applicability is assessed for submarine use.

The results of the analysis indicate that a smaller volume, lighter weight condenser can be designed for marine use. The marine engineer can specify tube length or condensate drainage rate and size the condenser accordingly. Approximately 20% can be saved in volume, and, by using Titanium in place of Copper-Nickel, approximately 30% can be saved in tube bundle weight, compared to a similar horizontal condenser.

Accession For

MARINE SURFACE CONDENSER DESIGN USING

VERTICAL TUBES WHICH ARE ENHANCED

by

CLIFFORD GERALD BARNES, JR.
LIEUTENANT, US NAVY

B.S., United States Military Academy
(1974)

Submitted in Partial Fulfillment
of the Requirements of the Degrees of

OCEAN ENGINEER

and

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

May 1981

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similar horizontal condenser.

Thesis Supervisor: Warren M. Rohsenow
            Title: Professor of Mechanical Engineering

To my brother, David

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CHAPTER I

INTRODUCTION

A. Background Information

In the design of a marine power plant, overall thermal efficiency, weight, and volume are the driving concerns [1]. This is especially true for the steam plant, which is the predominant form of propulsion for large naval vessels. While a large marine diesel may provide better specific fuel consumption than a fossil fired steam plant, the latter retains several intrinsic characteristics which make it attractive for naval propulsion. These advantages include:

1. Use of alternative heat sources.

    In the case of a submarine, aircraft carrier, or cruiser, a nuclear reactor provides the thermal energy. In smaller ships, a boiler is used as the steam generator.

2. Auxiliary steam generation.

    Auxiliary steam provides water, hotel services, and drives auxiliary equipment such as steam catapults on an aircraft carrier.

3. High speed.

    The steam plant has a much wider operating range than a large marine diesel. The design of a naval steam plant usually provides for

redundancy of steam generators which allows
for much higher speeds than a comparable sized
diesel plant.

4. Endurance.

The steam plant which has a nuclear reactor
as its heat source requires neither air nor
refueling during a specific mission.

There are advantages to other types of propulsion
plants, but the brief list above points out that for certain applications the steam plant is irreplaceable. The
high speed, large shaft horsepower, and endurance required
by an aircraft carrier, and the covert endurance required
by a submarine dictate the need for a steam propulsion
system. In the aircraft carrier, thermal efficiency may
be paramount to the design of the plant due to large
demands by auxiliary systems, but in the submarine,
weight and volume are critical.

In the submarine design, 25 per cent of all the use-
ful weight is devoted to propulsion. This term, for the
sum of all useful weight, is "normally submerged condi-
tion," (NSC), and it is the submerged displacement minus
the ballast water necessary to maintain neutral buoyancy
[2]. The design process for the submarine is to first
lay out the machinery section and determine its critical

length. This length is referred to as the "stack length." Once the stack length has been established along with its weight and volume, the remaining required weights are added, and the overall buoyant envelope can be determined [3]. This envelope is the volume necessary for neutral buoyancy. The stack length, therefore seriously impacts on the total volume of the submarine, and the weight of the machinery also becomes a constraint. With these two constraints merely removing weight from within the buoyant envelope, may be insufficient to allow the reduction of the envelope. Weight removal must often be countered with addition of lead ballast. Conversely, reducing the volume of the hull, but not the weight causes negative buoyancy. Also, the decrease in volume of specific items within the buoyant envelope allows greater freedom in arrangement, but unless accompanied by a commensurate weight reduction, a decrease in NSC will not be realized.

Consider the fact that a Los Angeles class attack submarine (SSN-688) costs 460 million FY-81 dollars [4]. This cost includes the cost of construction plus all government furnished equipment. The dimensions of this submarine are: length, 362 ft; diameter, 33 ft; and submerged displacement, 6900 tons [5]. A very crude

approximation would say that each foot of length costs
one million dollars and displaces 19 tons. So a moderate
reduction in the stack length which is accompanied by the
appropriate weight reduction should be worth several mil-
lion dollars in acquisition, with second order effects
being felt in horsepower requirements.

B.   Objectives of this Work

The objective of this thesis is to develop a design
methodology for an alternative surface condenser than
those presently used in marine steam plants. This alter-
native condenser is expected to be smaller and lighter
than existing units, and it will have a higher overall
heat transfer coefficient.

The average condensation heat transfer coefficient
on a smooth surface horizontal tube steam condenser is
approximately 2000 BTU/hr ft^2°F. Condensation inundation
of tubes which are lower in the condensing tube bundle
significantly reduces their heat transfer effectiveness
[6]. Vertical orientation of the smooth tube bundle
would not solve this inundation problem because a developing
boundary layer would cause similar adverse effects.
Figure I-1 shows the blocking effect of the fluid for
horizontal and vertical tubes.
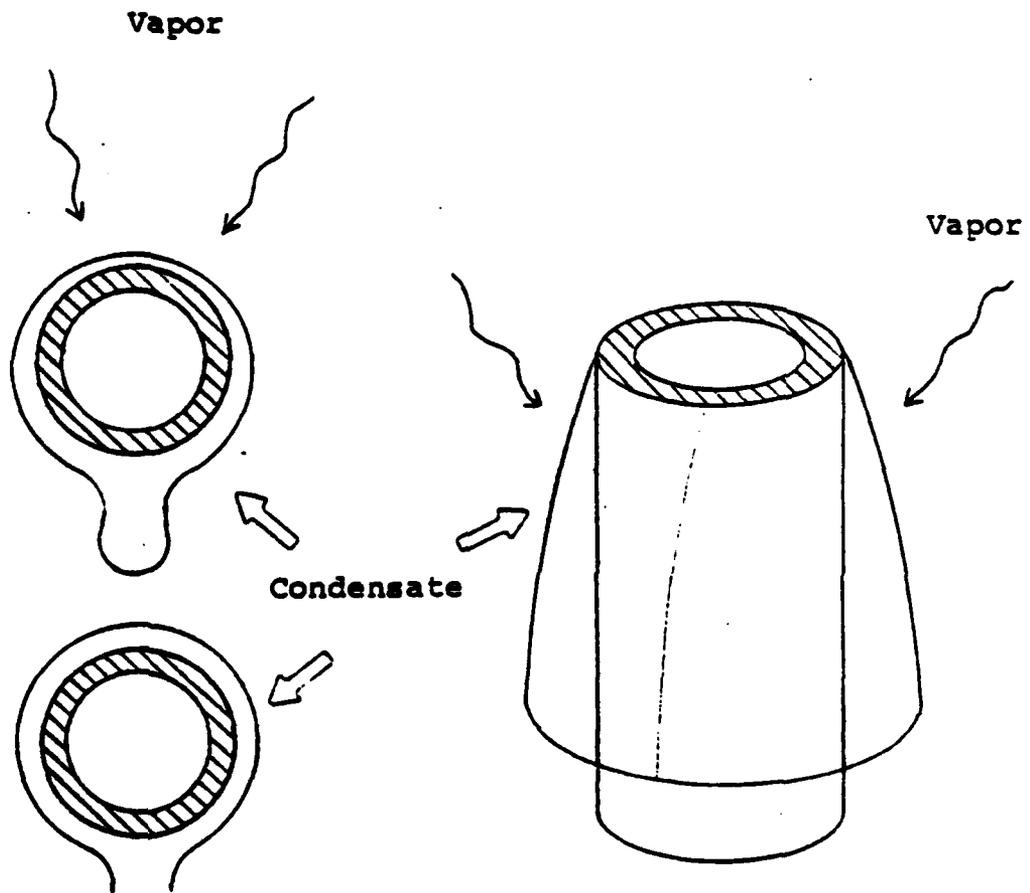
Figure I-1:Schematic showing condensation
         blockage effect on smooth horizontal
         and vertical tubes.

In 1954 Gregorig proposed condensing on vertically oriented tubes with fluted axial surfaces [7]. Surface tension of the condensate draws the liquid into the valleys of the flute where it then drains down, leaving a thinner liquid layer on the peaks. Figure I-2 demonstrates this concept. Heat transfer coefficients predicted for these condensing surfaces using Gregorig's fluid properties and temperature drops approach 8000 BTU/hr ft^2°F. In a procedure developed by Zener and Lavi [8] and refined by Webb [9] for the design of an optimized Gregorig condensing surface, heat transfer coefficients as high as 36000 BTU/hr ft^2°F are reported by experiment and as high as 55000 BTU/hr ft^2°F are predicted. These surfaces, however, may be difficult to manufacture, and for a generally curved repeating surface of such small dimensions as are required, the exact geometry may not be a significant influence on the condensing heat transfer coefficient [10]. The extrusion capabilities of industry are likely to set the limits on the geometry of the flute, and for these reasons a quite regular geometry such as a sine flute is considered. Longitudinal machining or axial welding to achieve a specific surface geometry is not acceptable for condenser tubes which are used in naval vessels [11]. By requiring seamless drawn tubes for naval condensers, residual stresses and machining flaws

Vapor

Tube wall   Condensate
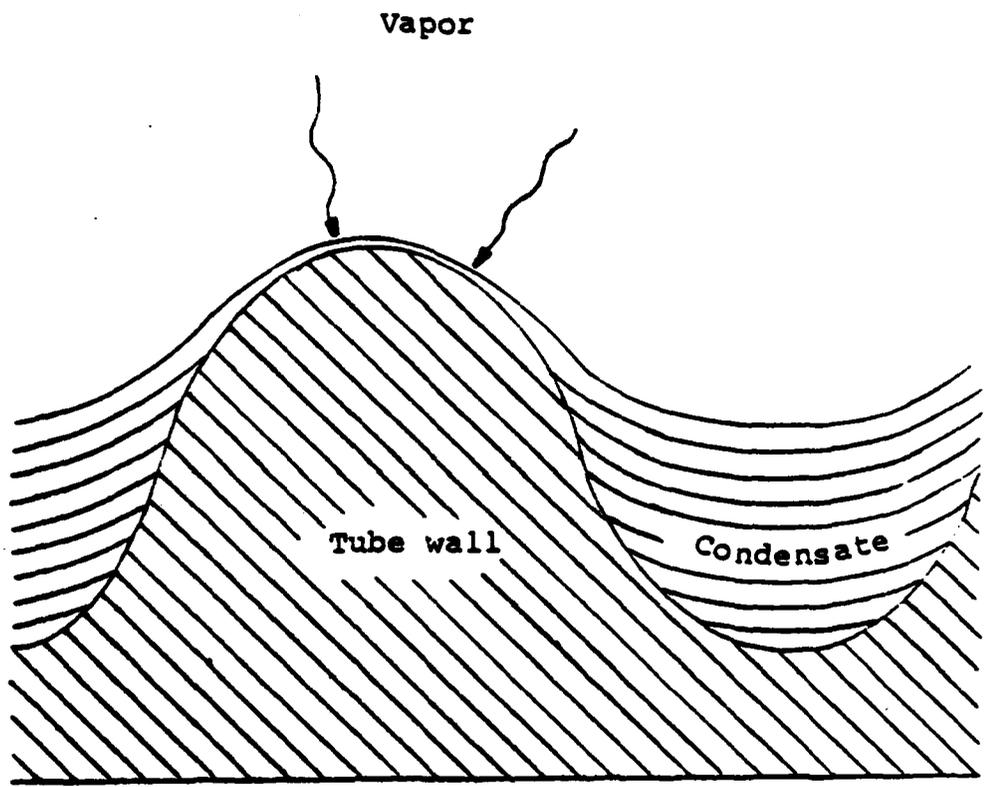
Figure I-2: Fluted surface concept

are avoided which could be sources of unexpected failure when the vessel is subjected to extremes of pressure, mechanical shock, or thermal shock.

The fluted surface having a sine geometry was analyzed by Yamamoto and Ishibachi using the Gregorig model [12]. Their analysis enables the calculation of an average condensate film thickness on the surface of the flute. The problem then reduces to one of balancing heat transfer and condensate mass flow. The phenomenon of the condensate filling the valley region of the flute is referred to as "flooding," and this concern for flooding presents the same problems as inundation of horizontal or vertical smooth tubes. Panchal and Bell considered the downstream effects of condensation as the valley of the flute begins to fill [13].

Proposing an alternative condenser design which simply re-orients the tubes and enhances their outer surfaces should accomplish the goals of being smaller and lighter. Condenser design considering alternative materials must be treated as part of the problem. Materials which warrant consideration are Copper-Nickel (Cu-Ni) 90-10, Cu-Ni 70-30, and Titanium, commercially pure, grade 2 (Ti, CP, Gr2). There are trade-offs for the use of these materials, the most important of which are strength and weight versus thermal conductivity.

In summary, the objective of this thesis is to present a methodology for a marine condenser design which has vertical fluted tubes and is possibly fabricated from different materials. This condenser must be lighter and smaller than existing condensers to warrant its manufacutre and installation.

C.   Design Inputs

Constraints for a condenser design can vary widely. For a marine condenser, these degrees of freedom are somewhat narrower, and the design process is governed by the concerns for weight, volume, and thermal efficiency. For naval condensers, reference [11] further limits the degrees of freedom in design. These limitations arise because of the requirements for the naval vessel to be able to operate effectively in varying environments.

The heat transfer requirements are for the conservation of energy from the condensation side of the condenser, through the tube wall, and into the cooling water. Thus the heat transfer characteristics can be analyzed separately for these three different regions.

1.   Cooling water.

Various correlations, such as McAdams, Colburn, or
Seider-Tate can be used to determine the heat transfer
coefficient for turbulently flowing cooling water [6].
Fundamental non-dimensional groups which determine the
cooling fluid heat transfer coefficient are the Reynolds
number and the Prandtl number. The design paprameters
which govern these numbers are temperature dependent fluid
properties, velocity, mass flow, and tube hydraulic
diameter.

2.   Tube Wall.

Heat transfer resistance through the tube wall is
dependent upon tube material, wall thickness, and a scaling
allowance.

3.   Condensing Fluid.

The heat transfer coefficient on the condensing
side of the tube will be shown to be a function of three
non-dimensional groups. These groups are specified by
surface geometry, condensing pressure (or temperature),
and condenstate drainage requirements [7, 12, 13].

Structural limitations are also defined in reference
[11]. These are the source of considerable conflict with
the heat transfer requirements. The desireable high

strength materials generally possess lower thermal conductivities and/or inferior metallurgical performance in salt water than lower strength pure alloys. Table I-1 gives a brief comparison for several common copper, nickel, and titanium based alloys. Stiffners are also necessary to prevent excessive tube vibrations. These stiffners may be used to provide condensate drainage strippers, but their spacing affects the condensation heat transfer coefficient [13, 14].

Power available for cooling water is a constraint which also affects the condenser design. Given a specific pump/motor limitation, by either size or power available, the velocity and mass flow of coolant are constrained. Heat losses through the condenser are determined by physical shape of the condenser, i.e., number of passes, baffles, etc., tube length, and the Reynolds number. As the tube length of the condenser decreases, a commensurate amount of pumping power becomes available for either increased coolant mass flow, or increased coolant velocity [15].

An itemized listing of design parameters for a marine condenser can be specified as follows [16]:

Table I-1

Thermal Conductivity and Yield Strength of

Several Common Metals and Select Alloys

[17,18]

Metal/Alloy*	Thermal Conductivity	Yield Stress**
	[BTU/hr ft °F]	[ksi]
Cu	230	∿ 4.8
Cu-NI 90-10	23	16
Cu-Ni 80-20	21	16
Cu-Ni 70-30	17	25
Ni (98.9% Pure)***	47	16
Monel 400 (66.5 Ni-31.5 Cu)	14	45-25
Monel 500 (66.5 Ni-29.5 Cu-2.7Al-0.6Ti)	10	111
Ti (Cp-Gr 2)	11.4-9.5	40
Ti-5Al-2.5Sn	4.5-4.3	120
Ti-6Al-4V	3.8	120

*Properties are for 50°F<T<100°F, annealed condition.

**Yield stress is defined for .2% deformation.

***Nickel and Ni-Cu alloys are unsuitable for marine tubing because they are subject to deep pitting in seawater at low velocities [23].

1. Tube material and material properties.

2. Tube geometry.

    a. Inside diameter, outside diameter, and wall thickness;

    b. Flute pitch and amplitude and nominal diameter.

3. Coolant characteristics.

    a. Inlet temperatures;

    b. Mass flow;

    c. Velocity;

    d. Physical and thermal properties.

4. Condenser operating conditions.

    a. Condenser pressure;

    b. Condensate temperature and mass flow;

    c. Heat transfer requirements.

5. External requirements.

    a. Tube length;

    b. Tube bundle lay-out;

    c. Number of passes;

    d. Operating depth and safety factors;

    e. Tube sheet and stiffner connections;

    f. Air ejector locations;

    g. Pumping power available or allowable head loss.

CHAPTER II

TUBE SELECTION

A.   Materials

The materials which will be considered for marine condenser tubing are the copper-nickel alloys, Cu-Ni 90-10 and Cu-Ni 70-30, and commercially pure titanium, grade 2 (Ti, Cp). These materials are of interest because the U.S. Navy uses both Cu-Ni alloys [11], and Ti is a possible alternative material which will provide considerable weight savings. Table II-1 lists the mechanical and thermal properties of these materials along with their compositions [17,18].

In selecting a material for use in a condenser, several environmental factors must be considered. These include corrosion characteristics and wear properties of the material in a salt water environment, effects of biological fouling, and galvanic coupling.

1.   Cooling water velocity effects.

The effects of sea water velocity on various metals is summarized in Figure II-1.

a.   Copper-Nickel Alloys.

Cooling water velocity is one of the more important factors that affect Cu-Ni condenser tube deterioration.

Table II-1

Mechanical Properties and Composition of

Marine Condenser Materials [17,18]

Alloy/Metal — **Designation**

Alloy/Metal	Designation
Cu-Ni 90-10	ASTM C-706, MIL-T-15005F
Cu-Ni 70-30	ASTM C-716, MIL-T-15005
Ti CP	ASTM B-265, Grade 2, Ti 50A

Composition

	Cu	Ni	Pb	Fe	Zn	Mn	Others
Cu-Ni 90-30	rem	9-11	0.05max	1-1.8	1.0max	1.0max	0.5max (total)
Cu-Ni 70-30	rem	29-33	0.05max	0.4-0.7	1.0max	1.0max	0.5max (total)

	Ti	N	C	H	Fe	O
Ti CP	rem	0.03max	0.10max	0.015max	0.30max	0.25max

Properties

	E (10^{+3}Ksi)	G (10^{+3}Ksi)	YS (Ksi)	P (lbm/in^3)	C_p (BTU/lbm°F)	K (BTU/hrft°F)	α (10^{-6}/°F)
Cu-Ni 90-10	20	7.5	16	.323	.09	23	9.5
Cu-Ni 79-30	22	8.3	25	.323	.09	17	9
Ti CP	15	6.0	40	.163	.125	9.5-11.4	4.8

Figure II-1 - The Effects of Sea Water Velocity on Corrosion
Resistance of Various Metals (23). (mpy = mils per year)

For these materials, a minimum velocity exists below which stagnation occurs and crevice corrosion is likely. Above a maximum critical velocity pitting and erosion occur. For Cu-Ni 70-30 this lower velocity limit is approximately 2-3 ft/sec, allowing for crevices on the order of 1.0 mil per year (mpy) [19,20,21]. Crevice corrosion character- istics for Cu-Ni 90-10 are similar to Cu-Ni 70-30 [22]. Various values are reported for maximum critical veloci- ties, above which impingement attack and erosion occur. The maximum sea water velocity of 11 ft/sec, specified by reference [11] is less than the generally accepted critical velocity for the Cu-Ni alloys which is 12-15 ft/sec [20,22].

b. Titanium

Ti is very immune to the effects of velocity. This behavior is attributed to a chemically stable, rapidly formed, oxide film. Passivation occurs rapidly, even in the presence of mechanical damage such as a scratch. The oxide film "repairs itself" virtually instananeously [21,22,23]. In sea water at velocities between 0-3 ft/ sec, reported general corrosion rates are nil [21,23]. Crevice corrosion occurs with Ti only at temperatures above 250°F [23], and no corrosion occurs at sea water velocities up to 25 ft/sec and possibly greater [21].

2.   Biological Fouling

In general, the more corrosion resistant metals and alloys are likely to experience biological fouling under slow moving conditions. Fouling by macro-organisms (mollusks, barnacles, etc.) has the obvious effect of reducing water flow, changing velocity profiles, and increasing pumping power requirements. Fouling by micro-organisms (algae, etc.) has the effect of adding another layer of thermal resistance to the inside of the tube wall [24].

a.   Copper-Nickel Alloys

Water velocities in excess of about 5 ft/sec, temperatures above 120°F and below 50°F, low oxygen content or low food supply (such as in stagnant or closed systems) in the water will effectively prevent fouling of copper alloys by macro-organisms [20]. The copper alloys most resistant to all bio-foulding, however, are those containing more than 85 percent copper. Thus, even in a high velocity flow, some micro-biological fouling (referred to as "slime") will occur on Cu-Ni 70-30 surfaces due to bacterial and algae growth and adherance within the boundary layer [20,25,26,27]. The irritant or toxic effect of Cu^{2+} ions is believed to be the cause of reduced biological fouling on Cu-Ni alloy tubes [28].

b. Titanium

Since Ti is such an immune metal, biological fouling is a serious problem. The threshold velocity to prevent macro-biological fouling on Ti tubes is approximately 5 ft/sec [23,29]. Design of a system for active production of toxic agents must ensure that these agents are not environmentally hazardous. A chlorination system is acceptable to the environment, but the production of chlorine ions (Cl^-) is accompanied by hydrogen ions (H^+). Hydrogen ions are a hazard to titanium alloys, and except in the case of very pure metal can cause stress corrosion cracking [30].

3. Galvanic Coupling

When considering condenser materials, in the presence of a strong flowing electrolyte, namely sea water, galvanic effects are significant. The material itself, with its propensity to passivate, and the velocity of the sea water dictate the kinetics of corrosion due to galvanic coupling. Titanium is a much more active metal than copper, but it is because of this activity that it rapidly forms on oxide and becomes passive. Hence it is more noble than copper, and its general corrosion rates are much less. Table II-2 shows the relative nobility of various metals in flowing sea water, and it implies that Cu-Ni piping (used for

Table II-2

Galvanic Series of Metals in Flowing Sea Water [33]

Anodic or Least Noble

      Magnesium and magnesium alloys
      Zinc
      Galvanized steel
      Aluminum
      Cadmium
      Mild steel
      Wrought iron
      Cast iron
      Stainess steel 304 (active), 316 (active)
      Lead
      Tin
      Naval brass (60% copper, 39% zinc, 1% tin)
      Copper
      Red brass (85% copper, 15% zinc)
      Copper-Nickel 90-10
      Copper-Nickel 70-30
      Nickel
      INCONEL alloy 600 (78% nickel, 13.5% chromium,
                          6% iron)
      Nickel aluminum bronze
      Silver
      Titanium
      Stainless steel 304 (passive)
      INCONEL alloy 625
      HASTELLOY alloy C
      MONEL alloy 400
      Stainless steel 316 (passive)
      INCOLOY alloy 825
      Graphite
      Platinum

Cathodic or Most Noble

most sea water piping) would deteriorate in the proximity and in contact with Ti [23,31]. Reference [11] requires zincs as cathodic protection in Cu-Ni tubed condensers since Cu is noble relative to steel. The steel is noble relative to the zincs, and they therefore corrode sacrificially. Such sacrificial anodes would definitely be required in a Ti tubed condenser [32]. In fact, the deterioration rate for such anodes might dictate an entire change in sea water piping systems to a more noble metal which would then exacerbate the biological fouling problem. Figure II-2 demonstrates this concern for cathodic protection requirements for a Ti tube condenser.

The significant variables for material use in condenser tubing, apart from strength requirements, are sea water velocity, galvanic effects, and susceptibility to biological fouling. The latter two of these are also functions of sea water velocity as shown in Figure II-1.

B.   Strength Requirements

Reference [11] specifies for surface ship condensers a minimum tube wall thickness of .049 in. with Cu-Ni 90-10 alloy. The tube outside diameter is also specified as 5/8 (.625) in. Appendix A shows the
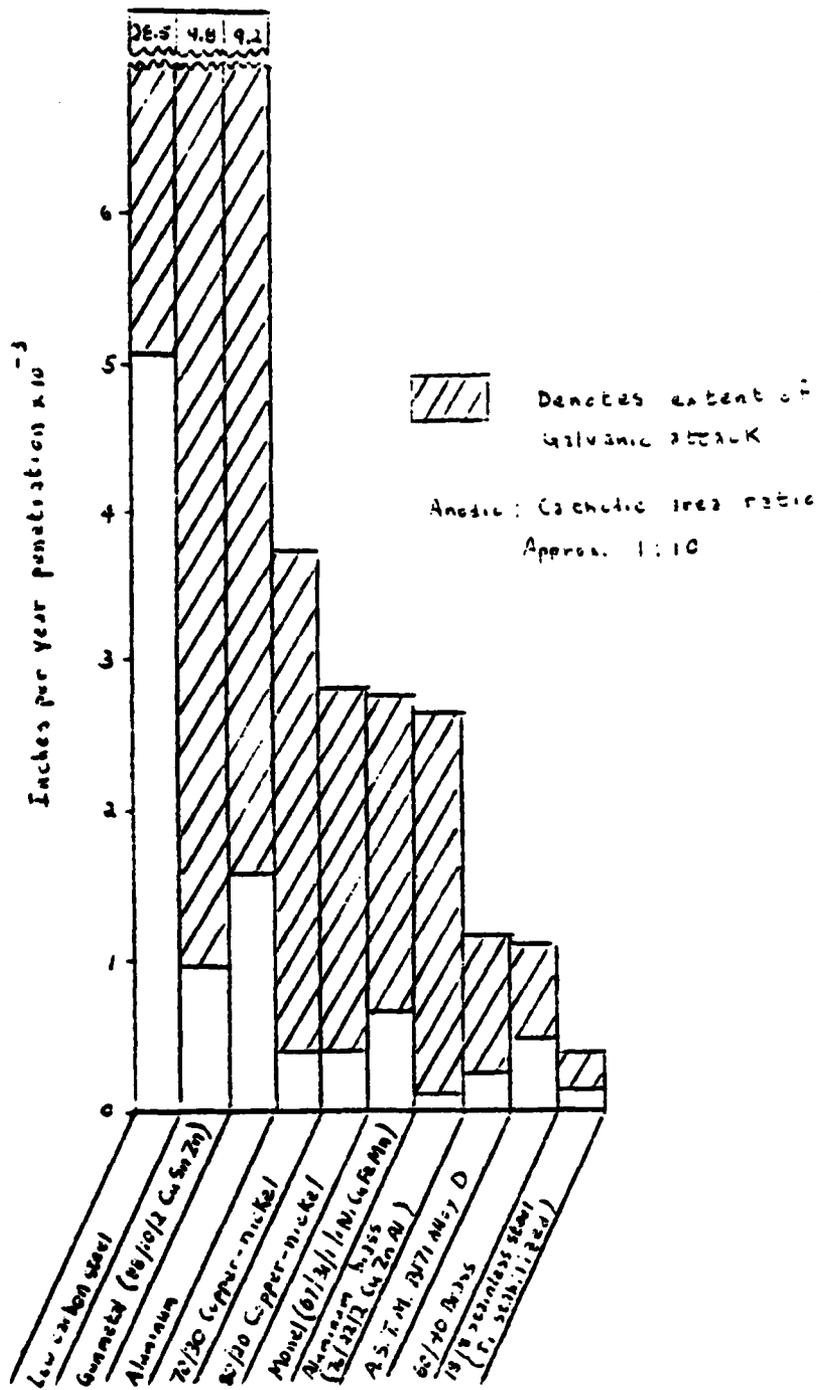
Figure II-2 - Behavior of Titanium metal couples immersed in aerated sea water for 2500 hrs. (32).

calculations required to determine condenser tube wall
thickness. These results are presented as Table II-3.
The minimum wall thickness allowed in practice for Cu-Ni
70-30 is 0.049 in. and for Ti is 0.035 in. [34]. Thus, from
Table II-3. replacing Cu-Ni 90-10 with Cu-Ni 70-30 Ti is
most practical strictly from a strength consideration for
a surface ship. Other considerations, such as increased
erosion protection in the face of a high velocity may
force the change in materials as shown in Figure II-1.

The question of tube wall thickness for a sub-
marine condenser is complicated by security requirements.
These can be treated academically by the following con-
servative assumptions:

1. The design depth of a submarine is not the
   depth at which it operates. A safety factor
   has been applied which may be either additive
   or multiplicative.

2. The yield criterion for tube wall material also
   incorporates a factor of safety.

Appendix A shows the calculations required to pro-
duce Table II-3. Table II-3 presents various condenser
tubes required for different design depths.

Table II-3

Tube Specifications for Various Design Depths*

Design Depth (ft)	Material	OD(in)	t(in)	Gauge
1050	Cu-Ni 90-10	.625	.049	18
1050	Cu-Ni 70-30	.625	.028	22
1050	Ti CP	.625	.019	--
1500	Cu-Ni 90-10	.625	.083	14
1500	Cu-Ni 70-30	.625	.049	18
1500	Ti CP	.625	.028	22
2000	Cu-Ni 90-10	.625	.109	12
2000	Cu-Ni 70-30	.625	.065	16
2000	Ti CP	.625	.035	20
2500	Cu-Ni 70-30	.625	.083	14
2500	Ti CP	.625	.047	18
3000	Cu-Ni 70-30	.625	.095	13+
3000	Ti CP	.625	.058	17+
1050	Cu-Ni 90-10	.500	.035	20
1050	Cu-Ni 70-30	.500	.028	22
1050	Ti CP	.500	--	--
1500	Cu-Ni 90-10	.500	.065	16
1500	Cu-Ni 70-30	.500	.035	20
1500	Ti CP	.500	.022	--
2000	Cu-Ni 90-10	.500	.083	14
2000	Cu-Ni 70-30	.500	.049	18
2000	Ti CP	.500	.028	22
2500	Cu-Ni 70-30	.500	.065	16
2500	Ti CP	.500	.035	20
3000	Cu-Ni 70-30	.500	.083	14
3000	Ti CP	.500	.049	18

*Thickness for tubes has been rounded to the nearest even gauge except as noted by +.

CHAPTER III

ELEMENTS OF HEAT TRANSFER

Nomenclature

a	amplitude of the flute (ft)
A	area (ft^2)
B	dimensional group defined in eq. (III-7) (ft/°F)
C, Cp	specific heat (BTU/lbm°F)
D	diameter (ft)
F	number of flutes on the tube surface
g_c	gravitational constant (lbm ft/lbf hr^2)
G	mass flux (lbm/ft^2 hr)
h	heat transfer coefficient (BTU/hr ft^2°F)
h_{fg}	latent heat (BTU/lbm)
K	thermal conductivity (BTU/hr ft°F)
L	length (ft)
M	non-dimensional group defined in eq. (III-16)
Nu	Nusselt number
p	pitch of the flute (ft)
P	pressure (1fb/ft^2) or (in-hg-abs)
Pr	Prandtl number
q	heat flow (BTU/hr)
r	radius of curvature of the fluted surface (ft)

R	thermal resistance (hr $ft^2 °F/BTU$)
Re	Reynolds number
S_v	flute half perimeter (ft)
S_c	flute arc length on which condensing occurs (ft)
t	tube wall thickness (ft)
T	temperature (°F)
ΔT	$T_S - T_W$ (°F)
U	overall heat transfer coefficient (BTU/ hr $ft^2 °F$)
V	velocity (ft/hr)
W	axial mass flow of liquid (lbm/hr)
s y z	tube surface coordinates

Greek letters

α	height of the condensate in the center of the flute (ft)
Γ	mass flow rate in S direction per unit length (lbm/hr ft)
δ	condensate film thickness (ft)
θ	angular representation of a point along the flute surface

μ dynamic viscosity (lbm/hr ft)

ρ density (lbm/ft^3)

σ surface tension (lbf/ft)

ϕ non-dimensional group defined in eq. (B-12)

Ω non-dimensional group defined in eq. (III-7)

Subscripts

b   at bulk temperature

c   coolant

D   referenced to hydraulic diameter

f   at flooding conditions

i   inside

n   nominal

o   outside

s   saturation

w   wall

sc   scale

Symbols

-      average

A. Heat Transfer Coefficient of Cooling Water

No simple analytical solution exists for heat transfer in turbulent pipe flow. The McAdams correlation (or Dittus-Boelter equation) is widely accepted for the determination of the heat transfer coefficient for the cooling water [6,39].

$$\left(\frac{h\,D}{K_b}\right) = 0.023 \left(\frac{G\,D}{\mu_b}\right)^{0.8} \left(\frac{\mu\,C_p}{K}\right)_b^n \qquad \text{(III-1)}$$

or

$$Nu_D = 0.023 \left(Re_D\right)^{0.8} (Pr)_b^n \qquad \text{(III-1a)}$$

where

1) $n = 0.3$ if the fluid is being cooled

  $= 0.4$ if the fluid is being heated

2) all fluid properties are evaluated at the average (or bulk) fluid temperature

3) $2300 < Re_D < 10^7$ where

  $Re_D \equiv$ Reynolds number based upon the hydraulic diameter

4) $0.5 < Pr_b < 120$ where

  $Pr_b \equiv$ Prandtl number based on bulk temperature

5) $L/D > 50$

For flow through a condenser tube, these conditions are usually met and equation (III-1) is used as a basic relationship.

B.   Heat Transfer Coefficient for a Tube Wall

For materials such as pure copper which have extremely high values for thermal conductivity, the thermal resistance of the condenser tube wall is almost negligible. From Table I-1, it is apparent that the strength requirements for marine condenser tubes mandate the use of materials with relatively low thermal conductivities.

The thermal resistance of the tube wall is the reciprocal of the heat transfer coefficient of the wall. For a smooth (non-enhanced) tube, this may be written as:

$$R_w = \frac{1}{h_w} = \frac{\ln(D_o/D_i) D_{ref}}{K} \frac{}{2} \qquad \text{(III-2)}$$

For a fluted tube, however, the nominal diameter is used in place of the outside diameter [40]. The nominal diameter is defined as:

$$D_n = D_o + 2a \qquad \text{(III-3)}$$

This is shown in Figure III-1. Use of the nominal dia-
meter will be shown to be reasonable for two reasons:

1. The size of the flute is small with respect
   to wall thickness. (This is exaggerated in
   Figure III-1).

2. Heat transfer is blocked by condensate at the
   bottom of the flute valley as shown in Figure
   III-2. [9].

Furthermore, it is convenient to work with the nominal
diameter vice the outside perimeter when dealing with
the heat flux and condensing heat transfer coefficient.
Equation (III-2) can be written as:

$$h_w = \frac{1}{\frac{\ln(D_n/D_i)}{K} \frac{D_n}{2}} \qquad \text{(III-2a)}$$

C.   Heat Transfer Coefficient for Condensation

The coordinate system for this analysis is schemati-
cally depicted in Figure III-2. The arc length, S, is
calculated by

$$S = \int_0^{\theta} \frac{p}{2\pi} \left\{ 1 + \left(\frac{2\pi a}{p} \right)^2 \sin^2 \phi \right\}^{1/2} d\phi \qquad \text{(III-3)}$$
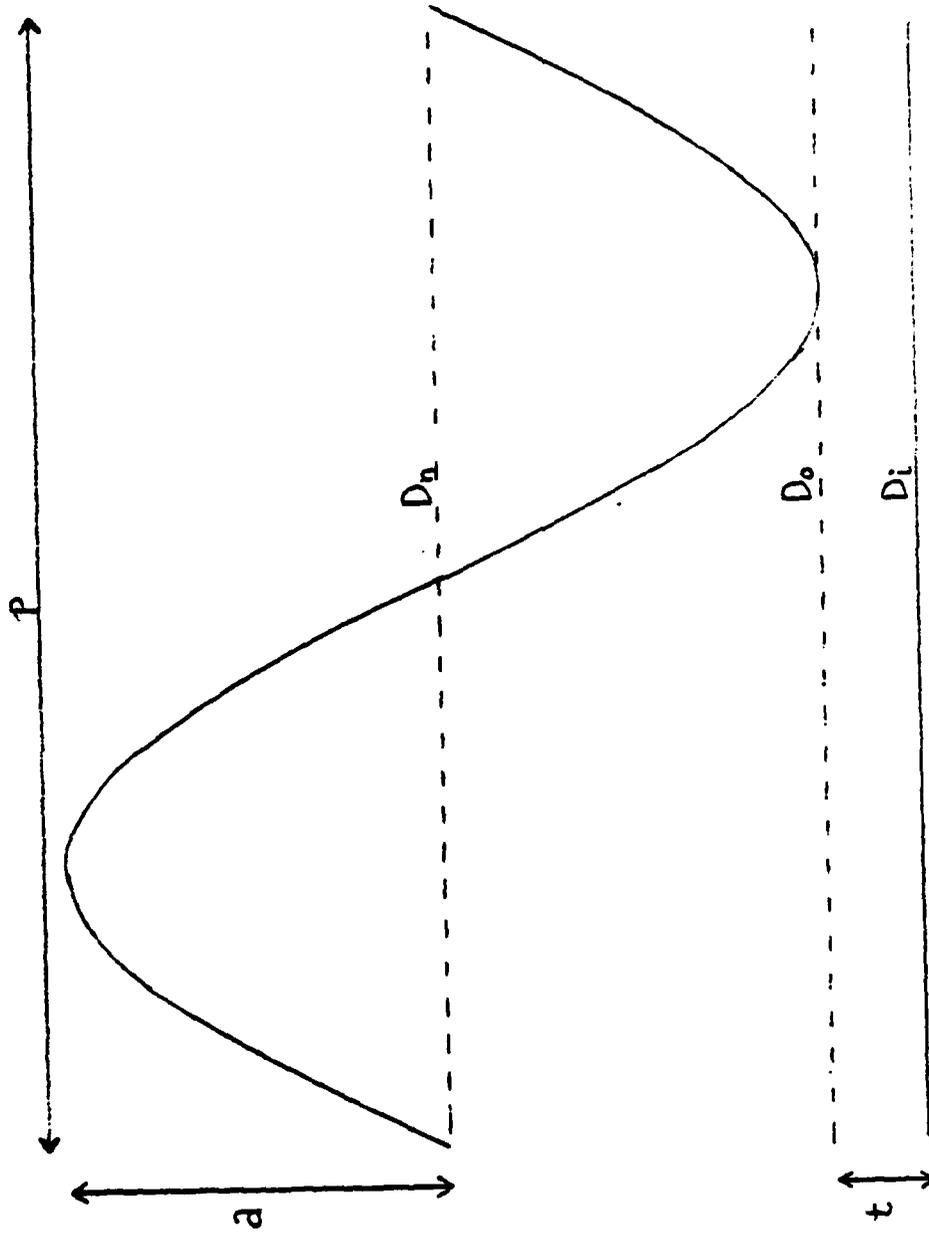
Figure III-1 - Schematic showing fluted tube dimensions
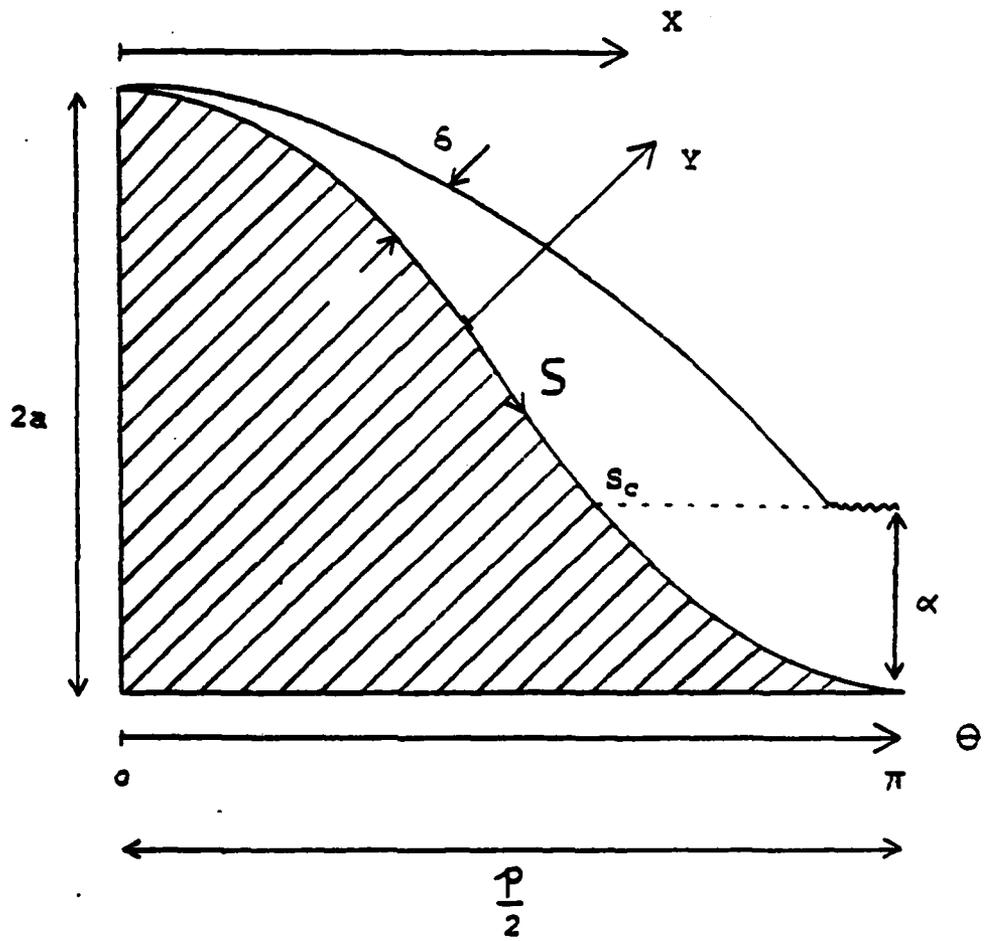
Figure III-2: Coordinates for flute
and condensate profile

The radius of curvature of the fluted surface which has a sine geometry is

$$r = \frac{a \left\{1 + \left(\frac{2\pi a}{p}\right)^2 \sin^2\theta\right\}^{3/2}}{\left(\frac{2\pi a}{p}\right)^2 \cos\theta}$$ (III-4)

From Figure III-3 it can be shown that

$$\frac{dP}{dS} = \frac{d}{dS}\left(\frac{\sigma}{r}\right)$$ (III-5)

Appendix B shows that the equation for the condensate film thickness, δ, for a curved surface with a surface tension induced pressure differential is

$$\Omega = \frac{-\delta}{3a} \frac{d}{dS}\left[\delta^3 \frac{d}{dS}\left(\frac{1}{r}\right)\right]$$ (III-6)

where Ω is defined as

$$\Omega \equiv \frac{\mu K \Delta T}{\rho \sigma h_{fg} g_c a} = \frac{B \Delta T}{a}$$ (III-7)

The steps for solving equation (III-6) are shown by Gregorig [7] and Yamamoto and Ishibachi [12]. The solution is presented in Appendix B. Sample calculations are shown in Table III-1.

$$P = P_{liquid} - P_{vapor}$$
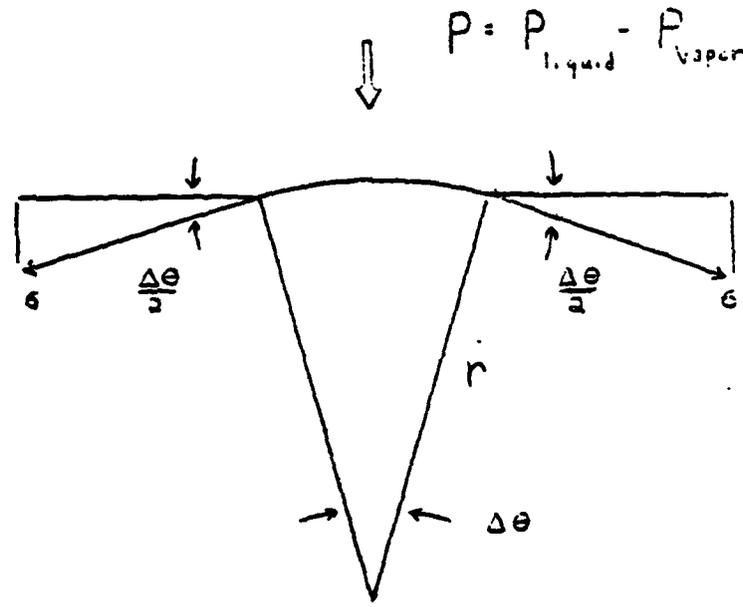
$$\sum F_r = 0 = Pr\Delta\theta - 6\sin\theta/2 \times 2$$
$$\sum F_\theta = 0 = 6\cos\theta/2 - 6\cos\theta/2$$

Figure III-3 - Schematic of fluid surface showing force balance between pressure and surface tension.

Table III-1

Sample Calculations for Condensate Film Thickness

a/p	.15	.35	.45	.55
n	$\delta_n \times 10^5$ ft			
0	1.5811	.4878	.3262	.2422
5	1.7219	.6673	.5436	.4802
10	2.0708	1.1451	1.0717	1.0522
15	2.5861	1.8466	1.8404	1.8785
20	3.1842	2.6712	2.7420	2.8451
25	3.7202	3.4183	3.5596	3.7215
30	4.0099	3.7606	3.9258	4.1083
35	3.9580	3.5078	3.6242	3.7693
40	3.6563	2.8574	2.8858	2.9616
45	3.2976	2.1107	2.0450	2.0468
50	3.0909	1.4664	1.3067	1.2374
55	3.4029	1.1087	.8361	.6900
60	6.7751	1.8444	1.2311	.8894

$$a = 2.00 \times 10^{-3} \text{ ft}$$

$$\Omega = 3.7649 \times 10^{-9}$$

$$\Delta\theta = \pi/60$$

$$\delta_n = f(n\Delta\theta)$$

From the results of equation (III-6), solved for the arc length $S_c = S_v$, Nu_o can be calculated as shown in Appendix B. A non-dimensional number, $Nu_o \Omega^{1/4}$, can be calculated, and from equation (III-6) and the definition of Nu_o,

$$Nu_o \Omega^{1/4} = f(a/p) \tag{III-8}$$

This result is presented in Table III-2 and plotted in Figure III-4.

At a distance Z down the tube, condensate W runs down the flute, and the thickness of this condensate, α (Figure III-2), increases with Z. Here the heat transfer is neglected in the valley of liquid thickness α, and the integration of equation (III-6) stops at $S = S_c$. S_c is determined by equation (III-3) and θ_c.

$$\theta_c = \pi \left(1 - \frac{\alpha}{2a} \right) \tag{III-9}$$

When $\alpha = 2a$ the valley is flooded [13], and $W = W_f$ where

$$W_f = 2g \frac{\rho^2}{\mu} (2a)^4 \left[36 \frac{a}{p} \exp \left\{ 3.33 \frac{a}{p} \right\} \right]^{-1} \tag{III-10}$$

Table III-2

Sample Calculations for $Nu_o \Omega^{1/4} = f(a/p)$

$a \times 10^3$ (ft)	a/p	Nu_o	$\Omega \times 10^9$	$Nu_o \Omega^{1/4}$
10.0	.1	532.31	3.7649	4.1697
5.0	.1	532.31	3.7649	4.1697
1.0	.1	532.31	3.7649	4.1697
2.0	.15	553.65	3.7649	4.3368
10.0	.25	326.54	37.649	4.5486
10.0	.25	580.68	3.7649	4.5486
10.0	.25	1032.62	.37649	4.5486
2.0	.25	580.69	3.7649	4.5486
2.0	.40	595.41	3.7649	4.6639
2.0	.45	596.89	3.7649	4.6755
2.0	.50	597.45	3.7649	4.6799
2.0	.55	597.36	3.7649	4.6792
2.0	.60	596.80	3.7649	4.6749
2.0	.65	595.92	3.7649	4.6680
2.0	.70	594.81	3.7649	4.6593
2.0	.75	593.55	3.7649	4.6494

Figure III-4: $Nu_o \cap^{\frac{1}{4}}$ vs. a/p
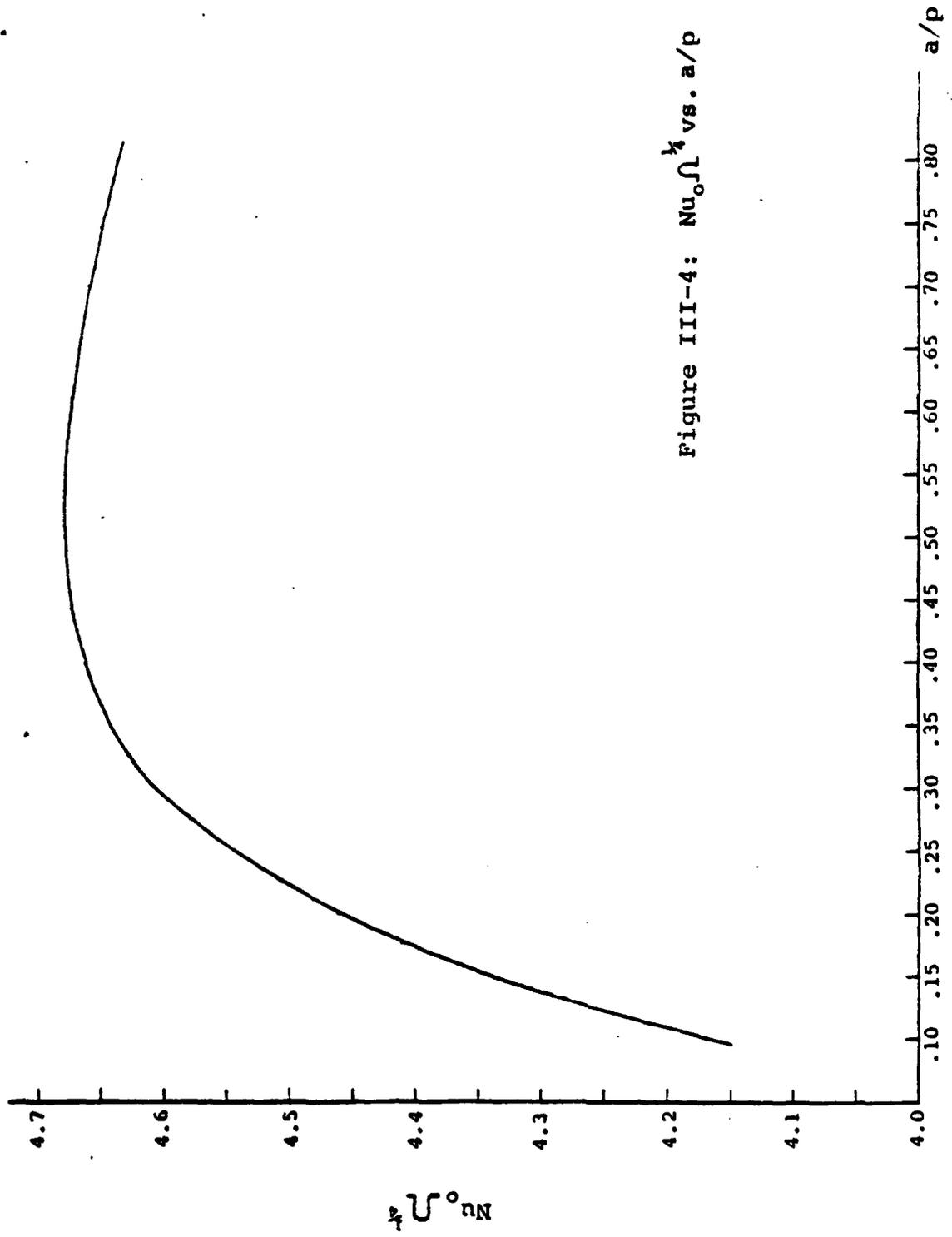
or

$$W_f = \frac{8}{9} \, g \, \frac{\rho^2}{\mu} \, (a)^3 \, p \, \exp\left[-3.33\left(\frac{a}{p}\right)\right] \qquad \text{(III.10a)}$$

A Reynolds number is defined [13] as

$$Re = \frac{4 \, W}{S_v \mu} \qquad \text{(III.11)}$$

Calculations by Panchal and Bell [13] show

$$\frac{W_z}{W_f} = \frac{Re_z}{Re_f} = \left(\frac{\alpha}{2a}\right)^{3.252} \qquad \text{(III.12)}$$

From Appendix B and equations (III-9) and (III-12)

$$\frac{h_z}{h_o} = \frac{Nu_z}{Nu_o} = f\left(\frac{Re_z}{Re_f} \, , \, \frac{a}{p}\right) = f\left(\frac{W_z}{W_f} \, , \, \frac{a}{p}\right) \qquad \text{(III.13)}$$

This is shown by example in Figure III-5.

An energy balance on the tube surface requires

$$h_z dZ \, \pi \, D_n \, \Delta T = h_{fg} \, dW \, \frac{\pi D_n}{p} \qquad \text{(III.14)}$$

This equation is re-arranged and integrated between $Z = 0$ and $Z = L$.
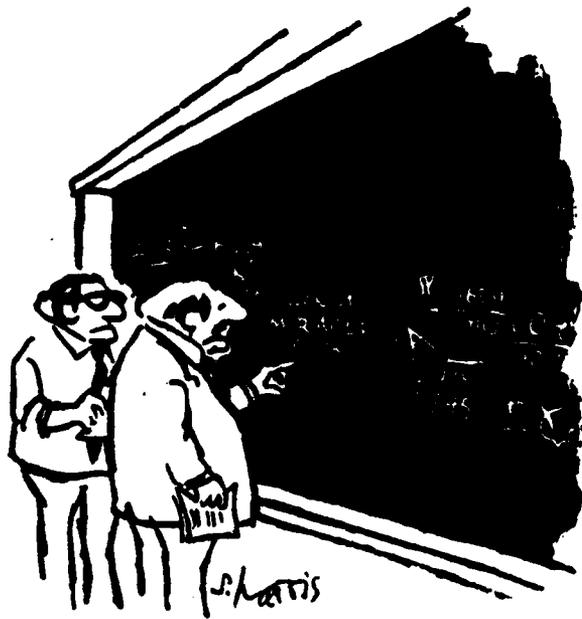
"I THINK YOU SHOULD BE
MORE EXPLICIT HERE IN STEP TWO."

Figure III-5: Nu/Nu_o vs. W/W_f

$\dfrac{a/p}{}$	
——	.15
—·—	.35
– –	.55
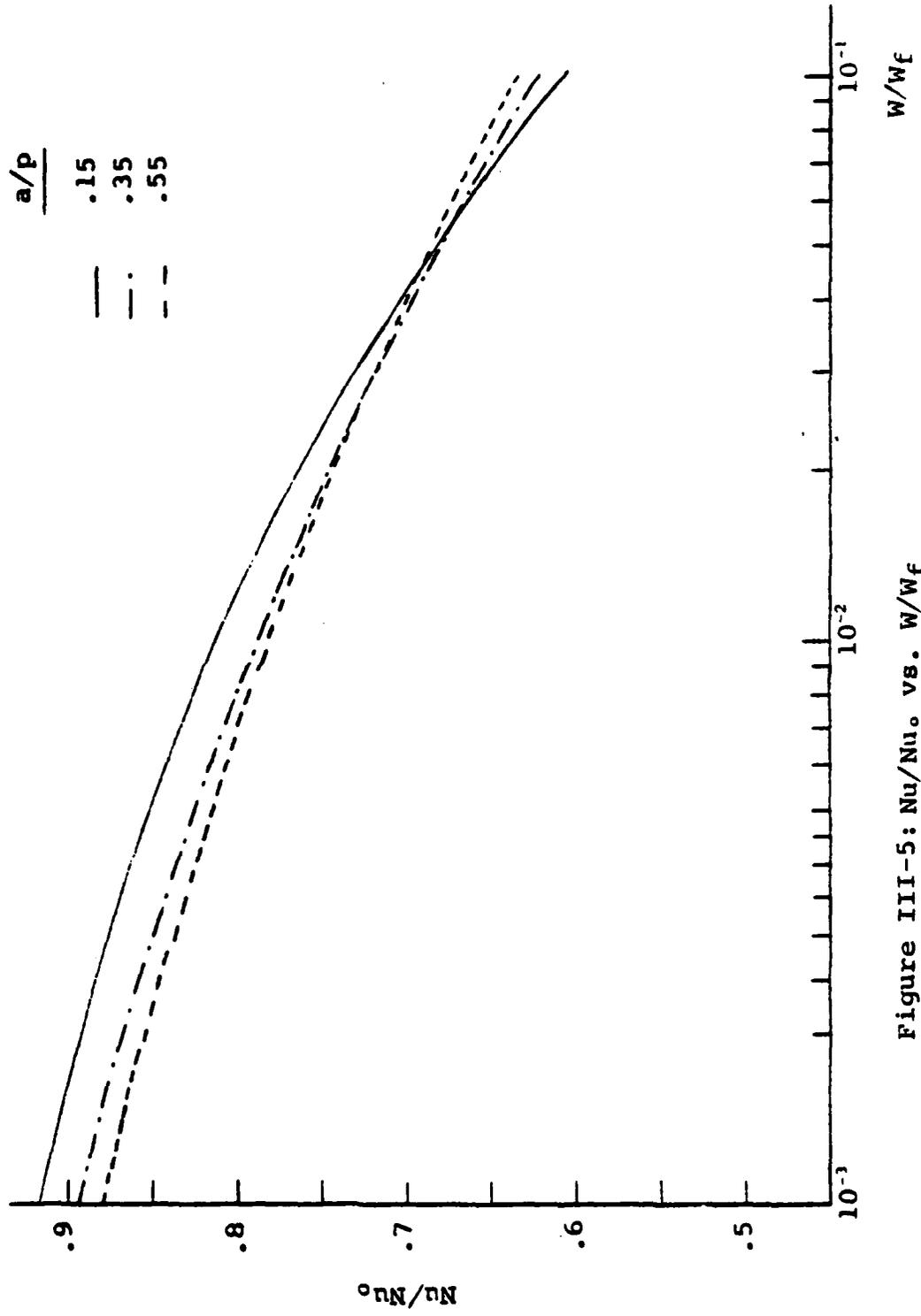
$$\frac{L \, p \, \Delta T \left[\frac{h_z p}{K}\right] \frac{K}{p} \left(Nu_o \Omega^{1/4}\right)}{h_{fg} \, W_f \, Nu_z \left[\frac{B\Delta T}{a}\right]^{1/4}} = \frac{Nu_o}{W_f} \int_0^{W_L} \frac{dW}{Nu_z} \qquad \text{(III.15)}$$

By grouping the constants in the right-hand expression, M can be defined as

$$M \equiv \frac{K(a)^{1/4} \left[Nu_o \Omega^{1/4}\right] \left[L\Delta T^{3/4}\right]}{h_{fg} \, (B)^{1/4} \, W_f} = \int_0^{(W/W_f)_L} \frac{Nu_o}{Nu_z} \, d\left(\frac{W}{W_f}\right)$$

$$\text{(III.16)}$$

Figure III-5 is replotted as

$$\frac{Nu_o}{Nu_z} = f\left(\frac{W_z}{W_f}\right) \qquad \text{(III.17)}$$

and presented as Figure III.6. The integral in equation (III.16) is evaluated from Figure III-6, and two of the cases are presented in Figures III-7a and III-7b. These results are also tabulated and presented as Table III-3. Note that the integration smooths out most of the dependency on (a/p) so that

Figure III-6 :Nu_o/Nu vs. W/W_f
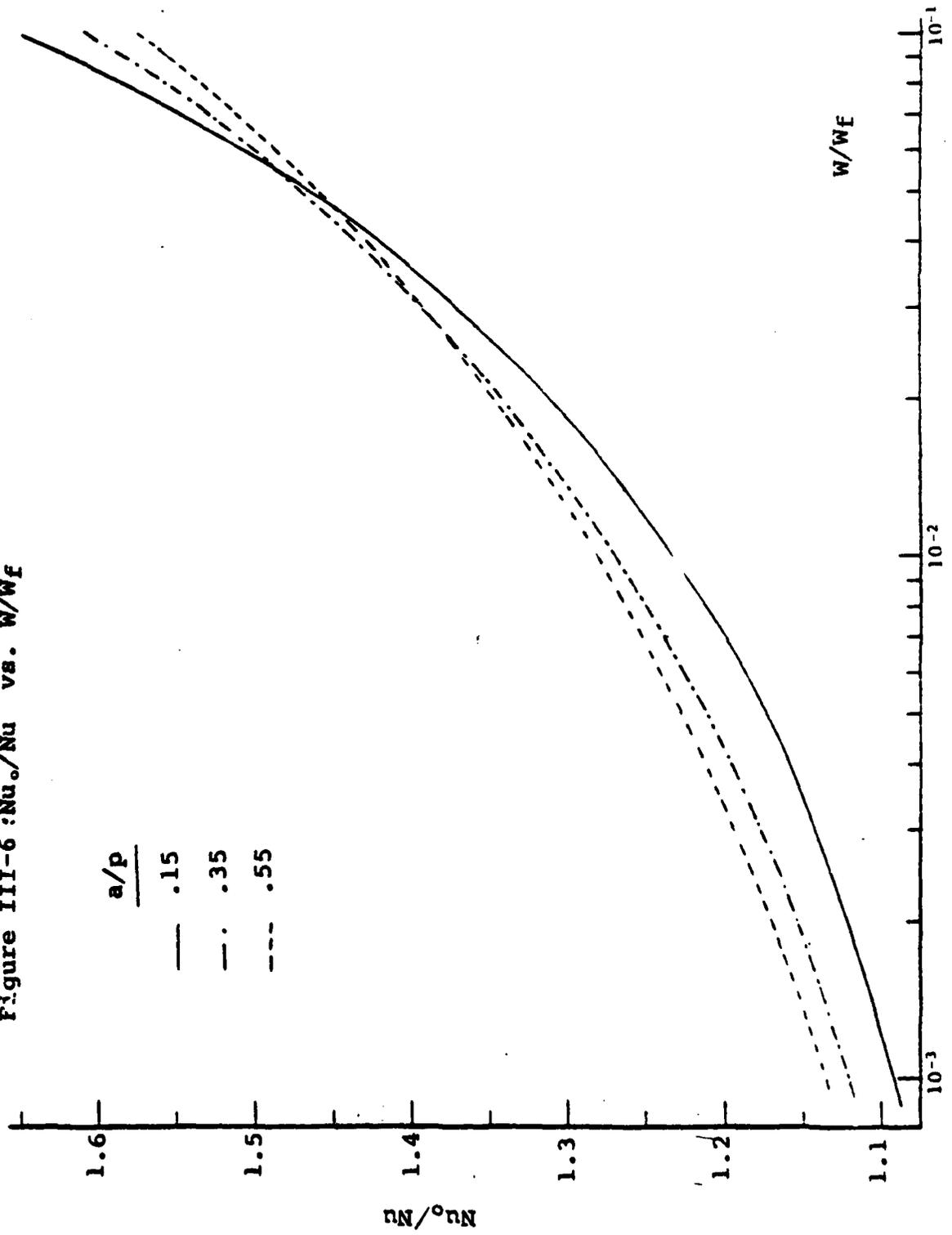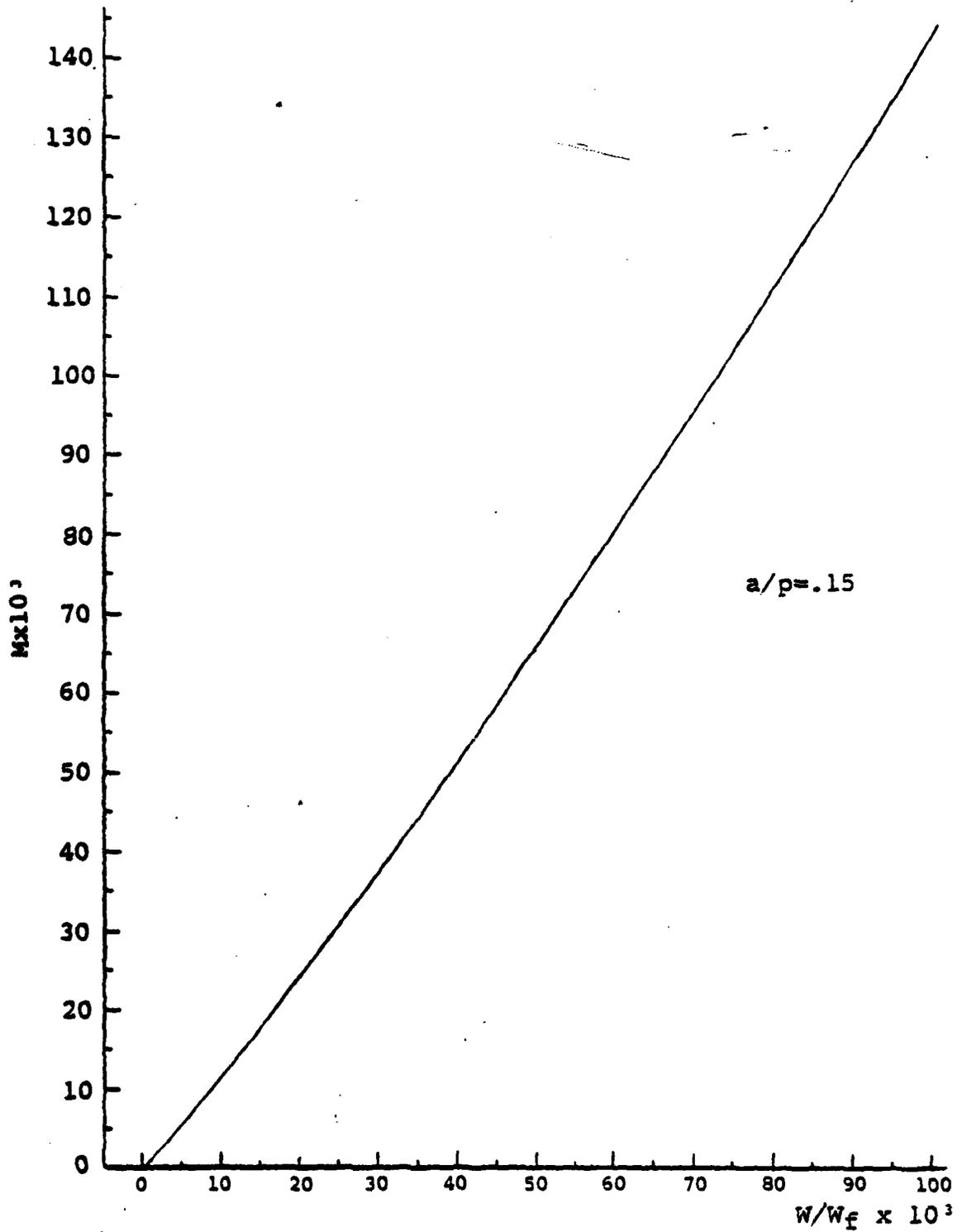
Figure III-7a: M vs. W/W$_f$
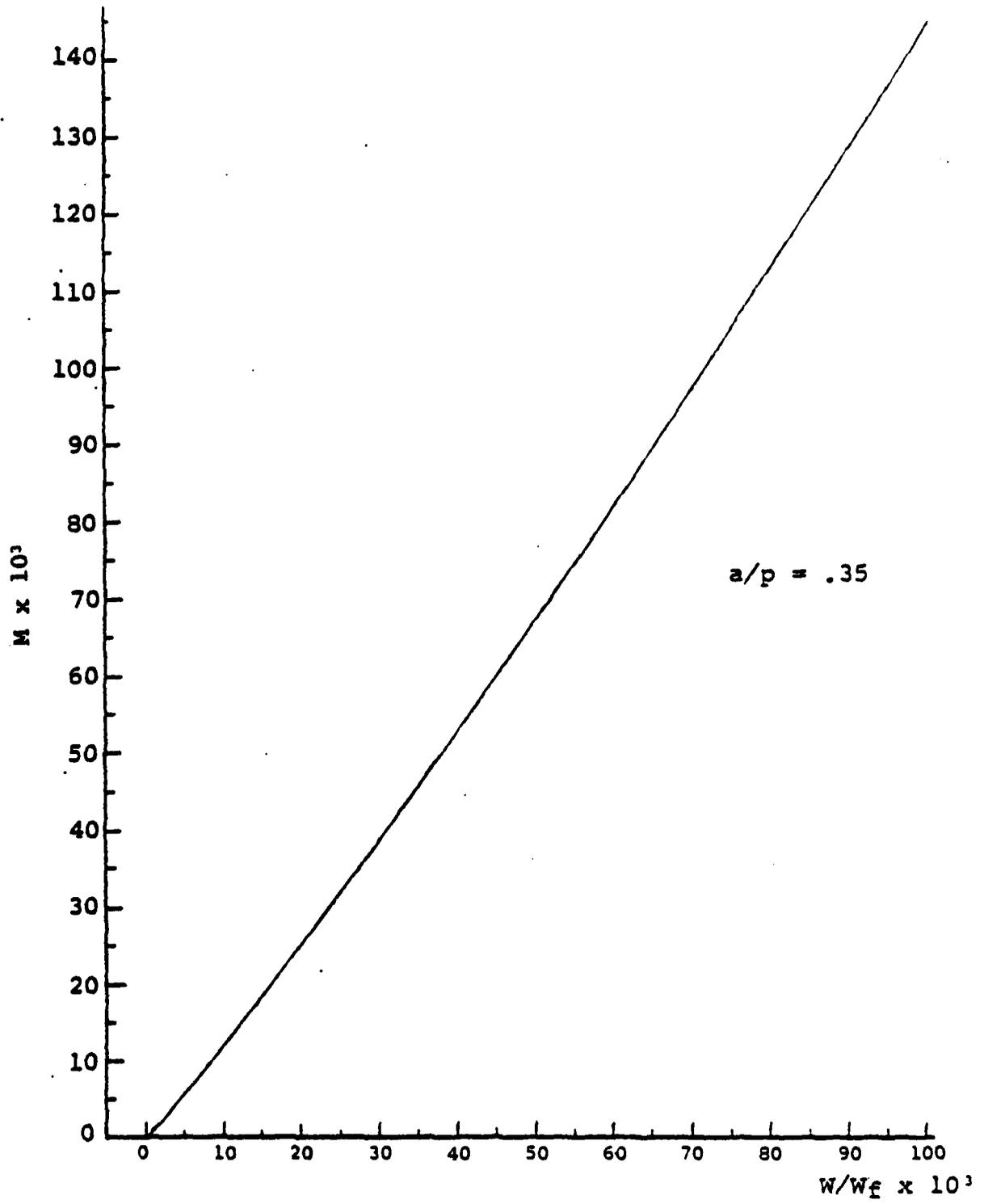
Figure III-7b: M vs. W/W$_f$

Table III-3

Calculated Values for M and $\frac{W}{W_f}$

for Several Different Values of a/p

M x 10^3

$\frac{W}{W_f}$ x 10^3	a/p = .15	a/p = .35	a/p = .55
0	0	0	0
1	1.06	1.07	1.07
2	2.18	2.21	2.24
3	3.35	3.38	3.42
4	4.44	4.56	4.62
5	5.61	5.77	5.85
6	6.79	6.97	7.08
7	7.98	8.22	8.32
8	9.19	9.46	9.58
9	10.40	10.72	10.85
10	11.63	11.98	12.12
15	17.87	18.43	18.62
20	24.36	25.04	25.26
30	37.61	37.75	38.97
40	51.48	52.56	52.76
50	65.97	67.14	67.20
60	80.75	81.95	81.88
70	96.07	97.15	96.90
80	111.63	112.55	112.08
90	127.67	128.30	127.55
100	143.96	144.21	143.16
110	160.70	160.46	159.05

$$M = f \ (W/W_f) \qquad \text{(III.18)}$$

$$M \neq f \ (W/W_f, \ a/p) \qquad \text{(III.18a)}$$

The data in Table III-3 suggests a relationship for M such that

$$M = 1.7312 \ (W/W_f)^{1.0839} \qquad \text{(III.19)}$$

for $W/W_f \geq 0.003$

The average heat transfer coefficient, $\bar{h}$, is defined as

$$\bar{h} \equiv \frac{(q/A_n)}{\Delta T} \qquad \text{(III.20)}$$

which may be written as

$$\bar{h} = \frac{h_{fg} \ W_f \ \left(\frac{W}{W_f}\right)}{p \quad L \quad \Delta T} = \frac{h_{fg} \ W_f \ .6027(M)^{.9226}}{p \quad L \quad \Delta T} \qquad \text{(III.21)}$$

Substituting for M from equation (III.16) gives

$$\bar{h} = .6027 \left[\frac{h_{fg}w_f}{L\Delta T}\right]^{.0774} a^{.2307} \frac{\left[\overline{f(a/p)}\right]^{.9226}}{p} \left[\frac{K^3 \rho c h_{fg} g_c}{\mu \Delta T}\right]^{.2307}$$

(III.22)

where $f(a/p) = Nu_o \Omega^{1/4}$ from Figure III-4. At $L = 0$ this should show that $\bar{h} = \bar{h}_o$. The relationship for w/w_f and M fails for $w/w_f < 0.003$, and this equation for $\bar{h}$ goes to ∞.

D.   Application to Condenser Design

In the design of a fluted condenser, it may be desireable to place stripper plates along the length to remove the condensate well before it reaches the flooding flow rate - perhaps keeping $w/w_f \leq 0.1$. Figure III-8 shows such stripper plates.

To determine L and w/w_f, a system of 4 equations must be solved. Other unknowns to this system of equations are: (q/A_n), Tc_2 and ΔT. Therefore, one of these five variables must be specified. In practice, either L or w/w_f would generally be chosen. The system of equations is:
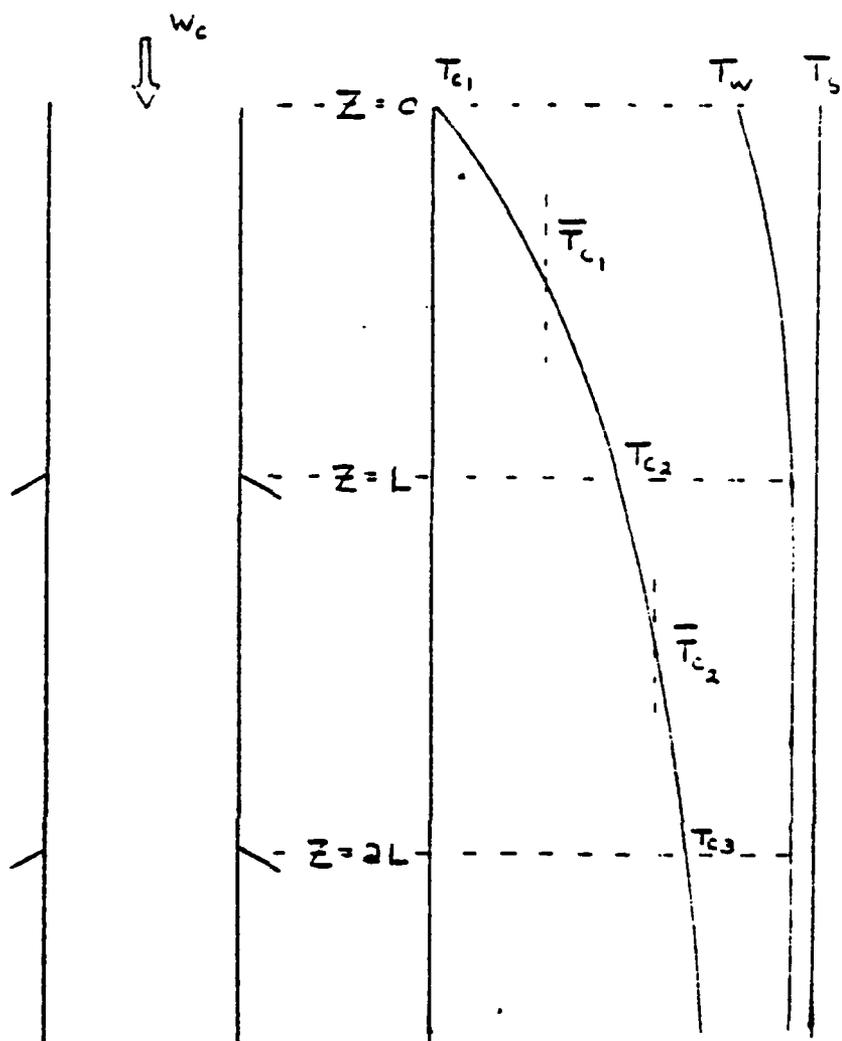
Figure III-8- Schematic diagram showing stripper plate spacing and axial temperature distribution.

1)
$$(q/A_n) = \frac{h_{fg} \ w_f \ (w/w_f) \ (\Delta T)^{3/4}}{p \ [L \ (\Delta T)^{3/4}]} \qquad \text{(III.23)}$$

which comes from the conservation of energy on one flute of the tube surface.

2)
$$(q/A_n) = \frac{W_c \ C_c \ (Tc_2 - Tc_1)}{\pi \ D_n \ L} \qquad \text{(III.24)}$$

which comes from the conservation of energy in the cooling water.

3)
$$(q/A_n) = U_{n,w} \ \frac{(Tc_2 - Tc_1)}{\ln\left[\frac{(T_s - \Delta T - Tc_1)}{(T_s - \Delta T - Tc_2)}\right]} \qquad \text{(III.25)}$$

which is the conservation of energy through the tube wall using an overall heat transfer coefficient and log mean temperature difference for the coolant.

$$\frac{1}{U_{n,w}} = \frac{D_n}{D_i} \left[\frac{1}{h_c} + \frac{1}{h_{sc}}\right] + \frac{\ln \ (D_n/D_i) \ D_n}{K \ \ \ \ 2} \qquad \text{(III.26)}$$

For short lengths or small temperature rises in the cool-
ant, the log mean temperature difference may be replaced
by

$$\frac{(Tc_2 - Tc_1)}{\ln\left[\frac{(T_s-\Delta T-Tc_1)}{(T_s-\Delta T-Tc_2)}\right]} \approx T_s - \Delta T - \overline{T}_c \qquad (III.27)$$

where $\overline{T}_c = \frac{1}{2}(Tc_1 + Tc_2)$ \qquad (III.28)

The fourth equation of the system comes from the defini-
tion of M in equation (III.16)

4)
$$\left[L(\Delta T)^{3/4}\right] = \frac{M\, w_f(B)^{1/4}\, h_{fg}}{(Nu_o\Omega^{1/4})\,(a)^{1/4}\,K} \qquad (III.29)$$

where M is approximated by equation (III.19).

With L specified and w/w_f allowed to float, the
solution to these equations is as follows:

    1) Assume Tc_2

    2) Assume ΔT

    3) Solve for (q/A_n) in equation (III.25)

    4) Solve for (w/w_f) in equation (III.23)

    5) Solve for M in equation (III.19)

6) Solve for ΔT in equation (III.29)

7) Repeat steps 3-6 until

$$|\Delta T_i - \Delta T_{i+1}| \leq \varepsilon_{\Delta T}$$

8) Solve for Tc_2 in equation (III.24)

9) Repeat steps 2-8 until

$$|Tc_{2j} - Tc_{2j+1}| \leq \varepsilon_{Tc_2}.$$

With w/w_f specified and L allowed to float, the solution to these equations is as follows:

1) Assume Tc_2

2) Assume L

3) Solve for ΔT in equation (III.29)

4) Solve for (q/A_n) in equation (III.25)

5) Solve for L in equation (III.23)

6) Repeat steps 3-5 until

$$|L_i - L_{i+1}| \leq \varepsilon_L$$

7) Solve for Tc_2 in equation (III.24)

8) Repeat steps 2-7 until

$$|Tc_{2j} - Tc_{2j+1}| \leq \varepsilon_{Tc_2}.$$

Table III-4 shows the results of these calculations for a typical condenser, neglecting thermal resistance in the wall. Appendix C shows the calculations for the solution to this system of equations as applied to a condenser design.

Table III-4

Sample Results for a Fluted Condenser*

Given:
D_o = .625 in

t = .049 in

Tc_1 = 75.0°F

v_c = 7.5 ft/sec

w_c = 12000 GPM (6.1513×10^6 lbm/hr)

a = 1.50×10^{-3} ft

a/p = .31205

P_c = 2.5 in-Hg-abs

$Nu_o \Omega^{1/4}$ = 4.612

B = 4.3777×10^{-12} ft/°F

D_n = 5.5083×10^{-2} ft

P = 4.8069×10^{-3} ft

F = 36

No. of tubes = 2354

w_f = 5.4020 lbm/hr

Section	1	2	3	4
ΔT(°F)	1.084	0.764	0.523	0.350
w/w_f	0.10	0.079	0.060	0.046
L (ft)	3.252	3.252	3.252	3.252
(q/A_n) (BTU/hrft^2)	35650	28000	21550	16300
Tc_2	83.17	89.59	94.53	98.26

q_{total} = 1.345×10^8 BTU/hr

*wall resistance neglected; cooling fluid is sea water.

E.   Heat Transfer Resistance Due to Scaling

Allowance for dirt or scale can be made by either including separate thermal resistances for the scale or fouling associated with the tube material and fluid or by allowing a percentage reduction of the overall heat transfer coefficient, U. Reference [11] requires the latter method using a 15 percent reduction. Reference [6] recommends a straight allowance, regardless of material but dependent upon cooling medium. The value recommended for sea water is:

$$R_{sc} = \frac{1}{h_{sc}} = .0005 \text{ (hr } ft^2 {}^\circ F/BTU)$$

Reference [41] recommends including a separate resistance for metal oxide on each side of the tube and a separate fouling resistance. This gives a value for U of the form,

$$\frac{1}{U_n} = \frac{1}{h_{condensing}} + \frac{D_n}{D_i} \frac{1}{h_{sea\ water}} + R_W + R_{ox} + R_f$$

(III-31)

Values for R_{oxide} are listed in Table III-5. Since $R_{fouling}$ is a virtual unknown and the quantity $2R_{ox} + R_f$ can be approximated for all cases in sea water as

$$2R_{ox} + R_f = R_{sc} \sim .0005 \text{ (hr ft}^2\text{°F)/BTU)}$$

This gives a value for U_n of the form

$$\frac{1}{U_n} = \frac{1}{h_{condensing}} + \frac{D_n}{D_i}\left[\frac{1}{h_{sea\ water}} + \frac{1}{h_{scale}}\right] +$$

$$+ \frac{\ln\left(\frac{D_n}{D_i}\right)D_n}{K_{tube}\ 2} \qquad \text{(III.32)}$$

where $h_{sc} = 2000$ BTU/hr ft^2°F.

TABLE III-5

Values of R_{ox}, the Resistance of the

Oxide Film on the Inside and Outside

Surfaces of Clean Tubes [41]

Tube Material	$R_{oxide} \left(\dfrac{hrft^2{}^\circ F}{BTU} \right)$
Admiralty Metal	0.000136
Aluminum Bronze	0.000153
Aluminum Brass	0.000167
Cu-Ni 90-10	0.000178
Cu-Ni 80-20	0.000193
Cu-Ni 70-30	0.000243
Titanium	0.000195

CHAPTER IV

DESIGN PROPOSALS

A.   Condenser Geometry

A primary objective of this work is to provide a
methodology for a condenser design which allows the naval
architect to specify condenser length. The naval archi-
tect is constrained by weight and volume, but the dimen-
sion which is most critical in marine plant layout is the
tube length. The transverse area of the tube bundle can
grow with significantly less impact on the other en-
gineering systems than axial growth. By specifying the
tube length, the maximum tube deflections and vibration
characteristics can be predetermined. This allows the
placement of tube and shell stiffners which may also act
as condensate stripper plates as suggested in Figure
III-8. For typical spacing requirements of 1-8 feet,
flutes with amplitude of 12-24 mils ($1.0 - 2.0 \times 10^{-3}$ ft)
are predicted by this work. Appendix C demonstrates how
a length may be specified for a condenser design. A
typical double-pass fluted condenser drawing is shown
in Figure IV-1.
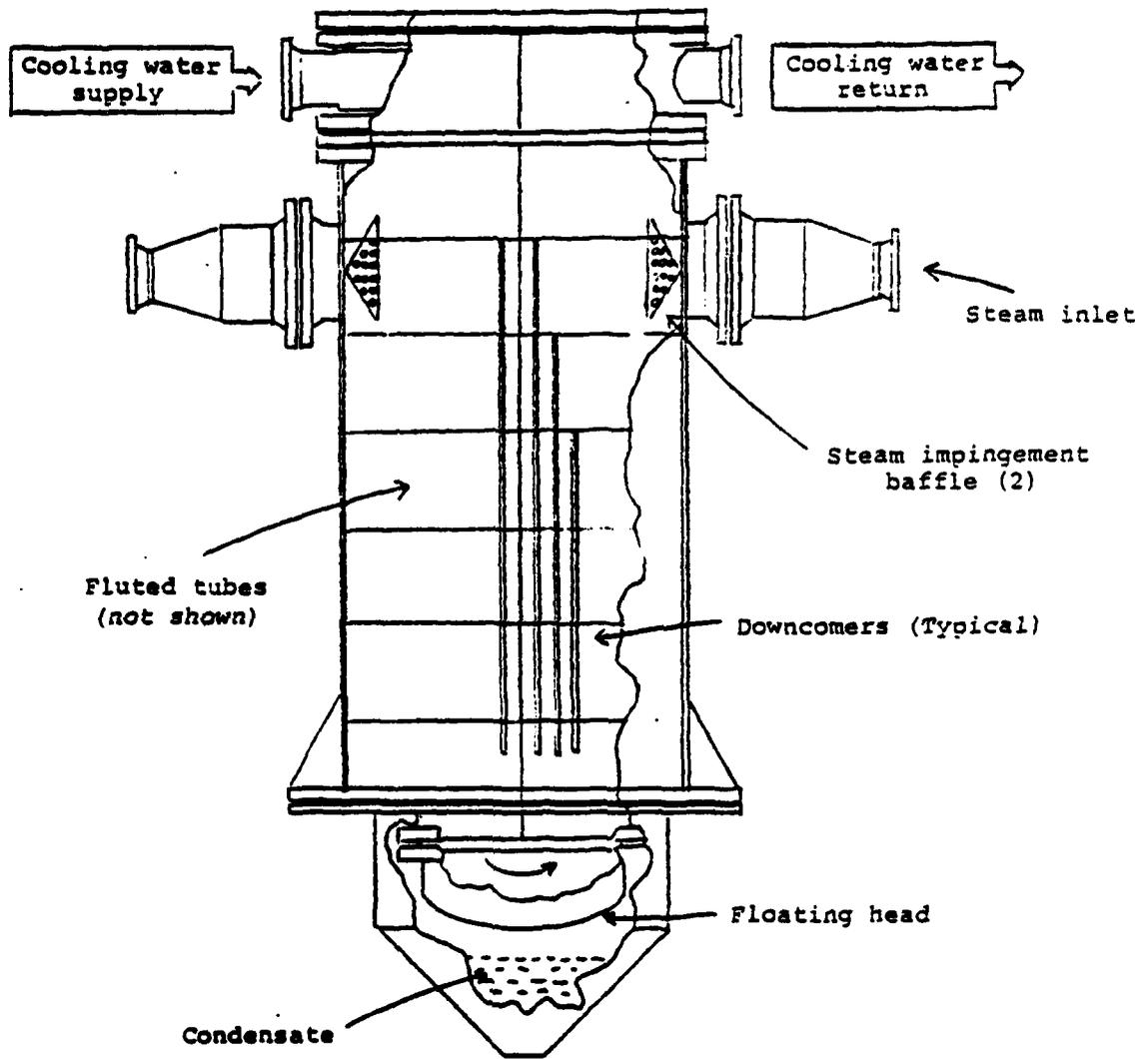
Figure IV-1: Possible configuration for
a 2 Pass Vertical Condenser

B.   Tube Attachment

A serious consideration for a naval condenser is the ability to pull and replace or plug tubes without having to remove the entire condenser or tube sheet. Tubes in naval condensers are usually attached to the tube sheets by rolling their ends and actually pressing them into the tube sheet [41,42]. This procedure is well established for Cu-Ni alloys, and rolled Ti tube-to-tube sheet joining procedures are being established to meet naval heat exchanger requirements [43].

Figure IV-2 is a drawing of a proposal to facilitate fluted tube attachment to the tube sheet while maintaining the ability to pull a tube if required. As shown in Figure IV-2, a forged collar can be placed around the fluted surface at points where attachment is desired. This collar could be snugged into place by heating it, prior to attachment, and allowing it to cool around the tube. Reference [41] shows a similar attachment for ferrules to allow longitudinal expansion. Reference [42] reports successful rolling of 5/8 inch diameter Ti tubes with 0.049 inch wall thickness. The composite wall thickness for a Ti tube with flute amplitude of 18 mils and tube wall thickness of 0.035 inches is 0.710 inches. This may pose a problem, and further research is warranted.
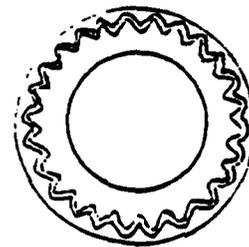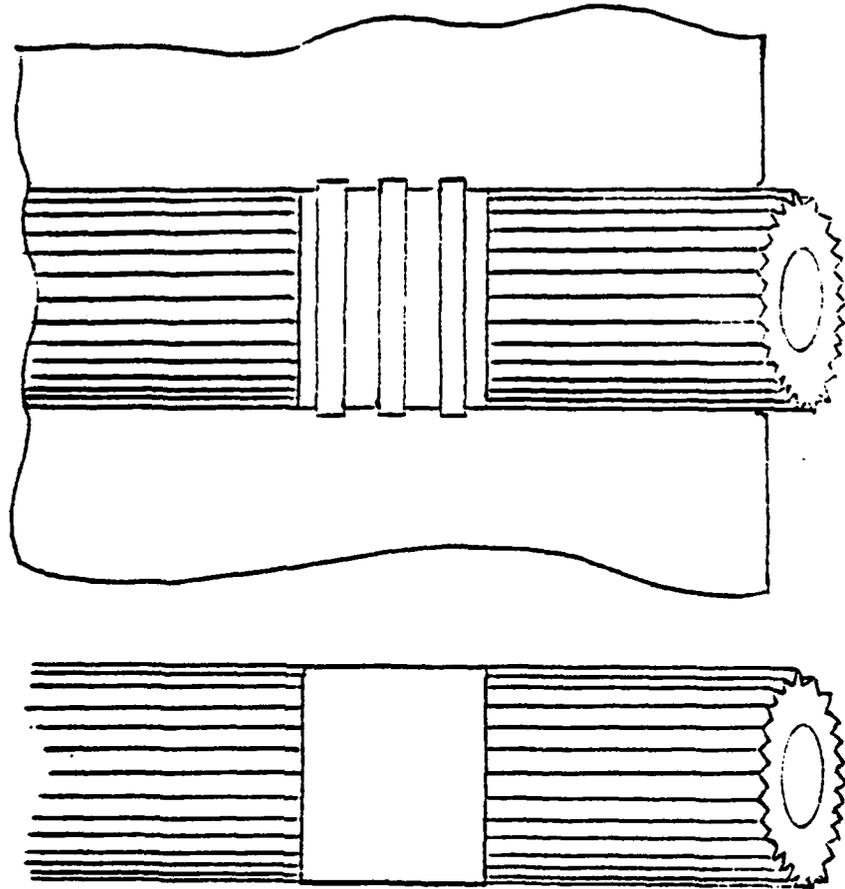
Figure IV-2 - A Proposed Method for Rolling Fluted Tubes into
the Tube Sheet Showing a Shaped Band Around the Tube.

CHAPTER V

RESULTS AND CONCLUSIONS

A.   Comparison with Nusselt analysis

The average heat transfer coefficient for conden-

sation on a fluted surface is given by equation (III.22).

$$\bar{h}_{fl} = .6027 \left[\frac{K^3 \rho h_{fg}}{\mu \Delta T} \right]^{.2307} \frac{(\sigma g_c)^{.2307}}{p}$$

$$x \; a^{.2037} \; [f(a/p)]^{.9226} \left[\frac{h_{fg} w_f}{L \Delta T} \right]^{.0774}$$

(III.22)

From the Nusselt analysis for condensation on a

smooth vertical surface,

$$\bar{h}_{Nu} = .943 \left[\frac{K^3 \rho h_{fg}}{\mu \Delta T} \right]^{.25} \left[\frac{(\rho - \rho_v) g}{L} \right]^{.25} \qquad (V.1)$$

These two equations show a much different dependence

on the axial length L, but $\bar{h}_{fl}$ is strongly influenced by

pitch. ΔT also affects $\bar{h}_{fl}$ more than $\bar{h}_{Nu}$. All other

things being of a similar magnitude or extremely weak

functions, $\bar{h}_{fl}$ is dominated by p, ΔT, and a. The Nusselt

analysis shows that $\bar{h}_{Nu}$ is strongly affected by L and ΔT.

Figure V-1 is a graph relating the average heat transfer coefficient with length. Since M as defined in equation (III.16) is linear with L, the relationship between (w/w_f), M, and L for a given $\bar{h}_L$ (or $\bar{h}_L/h_o = \overline{Nu}_L/Nu_o$) can be seen to be unique.

B.   Conden·· ·r Comparison Using Titanium

To cc ..,are a Ti condenser with the Cu-Ni 70-30 condenser cited in Appendix C, another constraint must be included. Since the Ti tubes have thinner walls, less of them will be required for the sea water mass flow. This implies that the Ti condenser will either be longer or transfer less heat. An alternative is to allow the mass flow of the coolant to increase, i.e., increase the number of tubes. This will be the approach taken here so that the condenser length and overall heat transfer remain the same. Table V-1 shows the differences between these two condensers.

No attempt has been made in Table V-1 to assess the overall weight change between Cu-Ni 70-30 and Ti as condenser materials for the example condenser. This overall weight change must include tube sheets, structural elements, waterbox heads, affiliated piping, foundations, etc. Assessment of these weights is beyond the scope of

Figure V-1

LEGEND

1 $\mathfrak{h}u_z/\mathfrak{h}u_o = f(u/u_f)$

2 $\mathfrak{h}u_z/\mathfrak{h}u_o = f(\mathfrak{h})$

3 $\overline{\mathfrak{h}u}_{1,z}/\mathfrak{h}u_o = f(u/u_f)$

4 $\overline{\mathfrak{h}u}_{1,z}/\mathfrak{h}u_o = f(\mathfrak{h})$
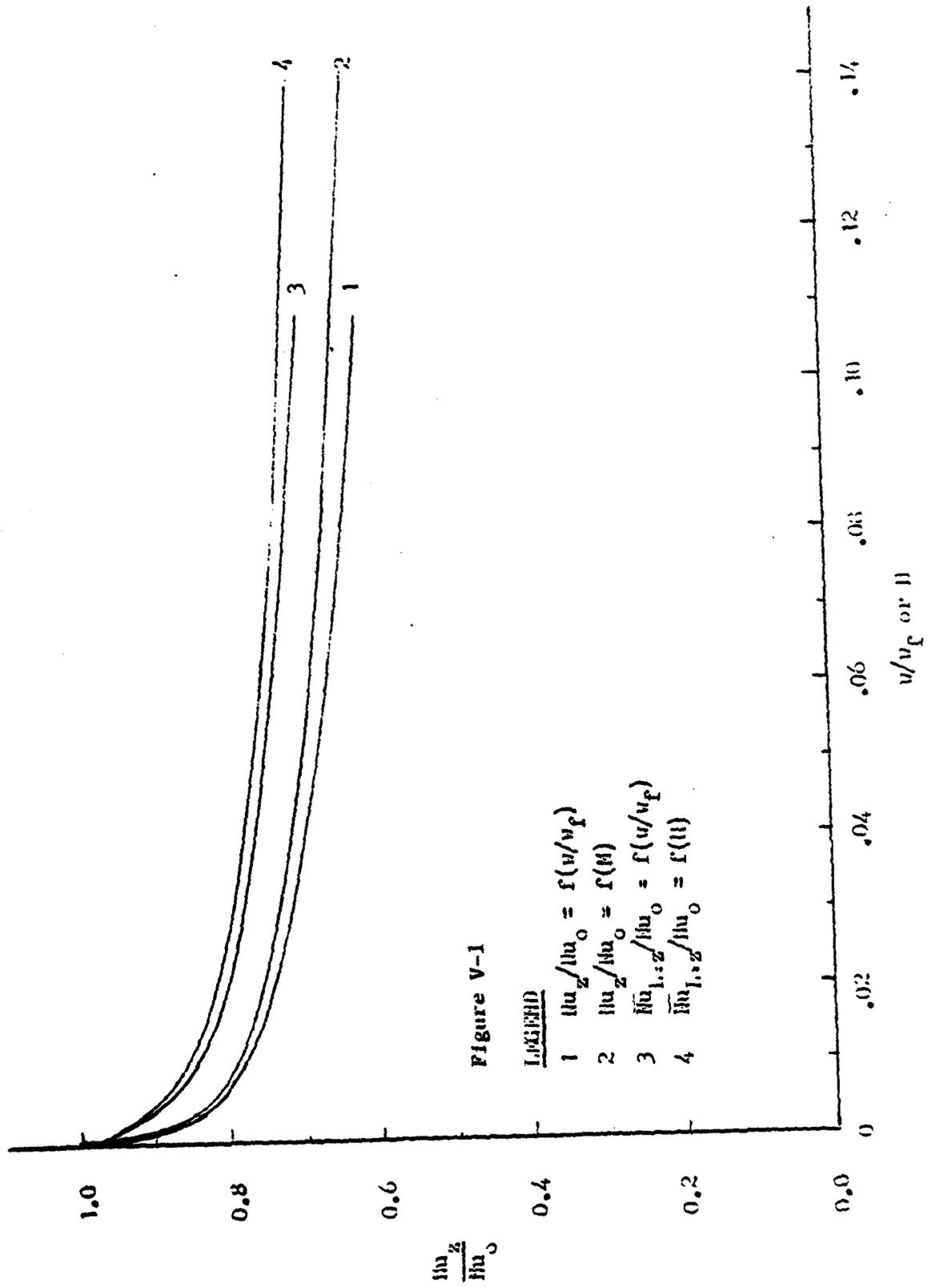
Table V-1

Comparison Between Condensers Using Cu-Ni and Ti

	Cu-Ni 70-30	Ti
D_o (in)	.625	.625
t (in)	.065	.035
a (mils)	18	18
a/p	.13869	.13869
Tc_1 (°F)	65.0	65.0
V_c (ft/sec)	8.0	8.0
W_c (GPM)	7900	8600
N_T	3294	2852
P_c (in-Hg-abs)	5.0	5.0
h_{sc} (BTU/hr ft^2°F)	2000	2000
K (BTU/hr ft°F)	17.0	11.4
F	16	16
No. of passes	2	2
$Nu_o \Omega^{1/4}$	4.305	4.305

Section 1

	Cu-Ni 70-30	Ti
$\bar{h}_{cond}$ (BTU/hr ft^2°F)	12312	11855
U_n (BTU/hr ft^2°F)	487.7	535.7
ΔT(°F)	2.489	2.895
w/w_f	0.060	0.060
L (ft)	4.957	4.426

Table V-1 (Continued)

	Cu-Ni 70-30	Ti
q/A_n (BTU/hrft^2)	30649	34325
Tc_2	76.36	74.03

Section 2

$\bar{h}_{cond}$ (BTU/hrft^2°F)	13197	12506
U_n (BTU/hrft^2°F)	501.3	547.9
ΔT (°F)	1.988	2.435
w/w_f	0.051	0.053
L (ft)	4.957	4.426
q/A_n (BTU/hrft^2)	26230	30451
Tc_2 (°F)	86.08	82.05

Section 3

$\bar{h}_{cond}$ (BTU/hrft^2°F)	15302	13628
U_n (BTU/hrft^2°F)	510.8	557.6
ΔT (°F)	1.528	2.007
w/w_f	0.019	0.034
L (ft)	2.086	3.148
q/A_n (BTU/hrft^2)	23383	27351
Tc_2 (°F)	89.72	87.16

Section 4

$\bar{h}_{cond}$ (BTU/hrft^2°F)	15800	14205
U_n (BTU/hrft^2°F)	514.9	564.2
ΔT (°F)	1.377	1.754
w/w_f	0.018	0.031

Table V-I (Continued)

	Cu-Ni 70-30	Ti
L (ft)	2.086	3.148
q/A_n (BTU/hrft^2)	21759	24919
Tc_2 (°F)	93.12	91.83

Section 5

$\bar{h}_{cond}$ (BTU/nuft^2°F)	15178	14405
U_n (BTU/hr ft^2°F)	519.0	569.9
ΔT (°F)	1.262	1.539
w/w_f	0.038	0.039
L (ft)	4.957	4.426
q/A_n (BTU/hrft^2)	19160	22166
Tc_2 (°F)	100.22	97.66

Section 6

$\bar{h}_{cond}$ (BTU/hrft^2°F)	16441	15326
U_n (BTU/hrft^2°F)	525.9	576.7
ΔT (°F)	0.974	1.258
w/w_f	0.031	0.034
L (ft)	4.957	4.426
q/A_n (BTU/ft^2)	16011	19287
Tc_2 (°F)	106.15	102.7

	Cu-Ni 70-30	Ti
q_{total} (BTU/hr)	1.57 x 10^8	1.57 x 10^8
Bundle Diameter (in)	61.8	57.0
Tube Bundle Weight (tons)	10.31	3.78
Weight of Added Water (tons)	--	2.67

this work. Table V-I shows that a smaller lighter con-
denser can be designed using Ti, but the penalty is a
requirement for increased coolant flow. Table V-2 is
an estimate of a comparable horizontal condenser using
Cu-Ni 70-30 and designed to meet the same requirements
as the condensers presented in Table V-1.

C. Conclusions

This thesis has presented a design methodology for
a fluted condenser. It has shown by example in Table
V-2 that a fluted condenser can be designed which is
more compact than a horizontal condenser. Overall den-
sity of the fluted condenser tube bundle increases, how-
ever, due to the tube enhancement. This may be off-set
by foundation and structural benefits, but that is not
addressed here. With a change in material, Table V-1
shows that both volume and weight can be saved at the
cost of increased pumping requirements.

Heat transfer coefficients for condensation are
predicted by the method described in Chapter III. For
flutes which are of the size 12-24 mils in amplitude,
increases in the heat transfer coefficients for con-
densation are greater than eight times that predicted

Table V-2

Comparison Between a Horizontal and a Fluted Condenser*

	Horizontal	Vertical
D_o (in)	.625	.625
t (in)	.065	.065
a (mils)	-	18
a/p	-	.13869
Tc_1 (°F)	65.0	65.0
v_c (ft/sec)	8.0	8.0
w_c (GPM)	7900	7900
No. of passes	2	2
N_T	3294	3294
P_c (in-Hg-abs)	5.0	5.0
h_{sc} (BTU/hr^2°F)	2000	2000
K (BTU/hrft°F)	17.0	17.0
L (ft)	14.85	12.0
$\bar{U}_o$ (BTU/hrft^2°F)	435.8	538.6 ($\bar{U}_n$=509.3)
$\overline{\Delta T}$ (°F)	12.07	1.639
$\overline{(q/A_o)}$ (BTU/hr ft^2)	19627	24257 (q/A_n= 22936
Tc_2 (°F)	106.15	106.15
q_{total}	1.57×10^8	1.57×10^8
Bundle Dia. (in)	53.4	61.8
Bundle Wt (tons)	9.68	10.31

*Material for both is Cu-Ni 70-30

by the Nusselt analysis for horizontal tubes. Stress

concentrations may be a concern for submarine use, but

by limiting the amplitude-to-pitch ratio, stresses can

be kept below an acceptable yield criterion. Great

gains in weight and volume savings could be realized

with enhancement on the cooling water side of the tube

in addition to vertical flutes.

D.   Recommendations

Because thermal resistance is treated in series,

the gains made on the condensation side of a tube may

not be fully realized because of the insensitivity of

cooling water flow and resistance due to scaling or

fouling.

Further investigation is required to consider the

following:

1.   Internal enhancement: Tremendous gains may be

realized by increasing the heat transfer co-

efficient of the coolant. An internal fin

arrangement could accomplish this. Drawbacks

to such arrangements are susceptibility to

fouling and tube inspectability.

2. Biofouling: The compactness of a titanium marine condenser, or possibly an aluminum stationary condenser, may be to no avail if they are rapidly blocked or fouled by marine organisms. Active and passive protection systems must be investigated which retard or prevent biofouling, and if these are not feasible, an in-service cleaning method must be considered.

3. Scaling: This allowance is a bona fide guess. With new materials, such as titanium, investigation into scaling allowances and mechanisms would allow a less conservative approach to condenser design.

4. Experimental Data: This is an absolute necessity. None has been provided with this thesis because very little is available in the literature concerning fluted tubes. A criticism of the literature cited is that no common terminology is used, nor is design or zero point data usually provided. Furthermore, most work on fluted tubes has been performed with ammonia and various regrigerants.

E.   Summary

A proposal has been made in this thesis to design a lighter more compact marine condenser. Calculations have been presented for selecting different condenser tubes for submarine use. Alternative materials have been considered and a comparison condenser design has been presented. A methodology for calculating the heat transer coefficient for fluted tubes has been presented, and this is incorporated into the condenser sizing procedure.

APPENDIX A

Condenser Tube Stress Calculations

Nomenclature

a	amplitude of the flute (in)
d	tube diameter (in)
h	design depth (ft)
p	pitch of the flute (in)
P	pressure (psi)
t	tube wall thickness (in)
Y	yield criterion (ksi)
YS	yield stress (ksi)
α	stress intensity factor
σ	stress (ksi)
r	
θ	cylindrical coordinates
z	

Subscripts

i	inner diameter
o	outer diameter
x	extreme diameter

The general equations for stress in a thick walled non-fluted tube are [35]:

$$\sigma_r = \frac{-P_i\left[(d_o/d)^2 - 1\right] + P_o\left[(d_o/d_i)^2 - (d_o/d)^2\right]}{\left(d_o/d_i\right)^2 - 1} \qquad (A.1)$$

$$\sigma_\theta = \frac{P_i\left[(d_o/d)^2 + 1\right] - P_o\left[(d_o/d_i)^2 + (d_o/d)^2\right]}{\left(d_o/d_i\right)^2 - 1} \qquad (A.2)$$

$$\sigma_z = -P_i \qquad \text{(this is because the tube sheet} \qquad (A.3)$$
distributes external pressure on

the ends of the tubes).

These stresses are shown in Figure A-1. For a condenser, $P_o \ll P_i$ and equations (A.1) and (A.2) simplify to:

$$\sigma_r = \frac{-P_i\left[(d_o/d)^2 - 1\right]}{\left(d_o/d_i\right)^2 - 1} \qquad (A.4)$$

$$\sigma_\theta = \frac{P_i\left[(d_o/d)^2 + 1\right]}{\left(d_o/d_i\right)^2 - 1} \qquad (A.5)$$
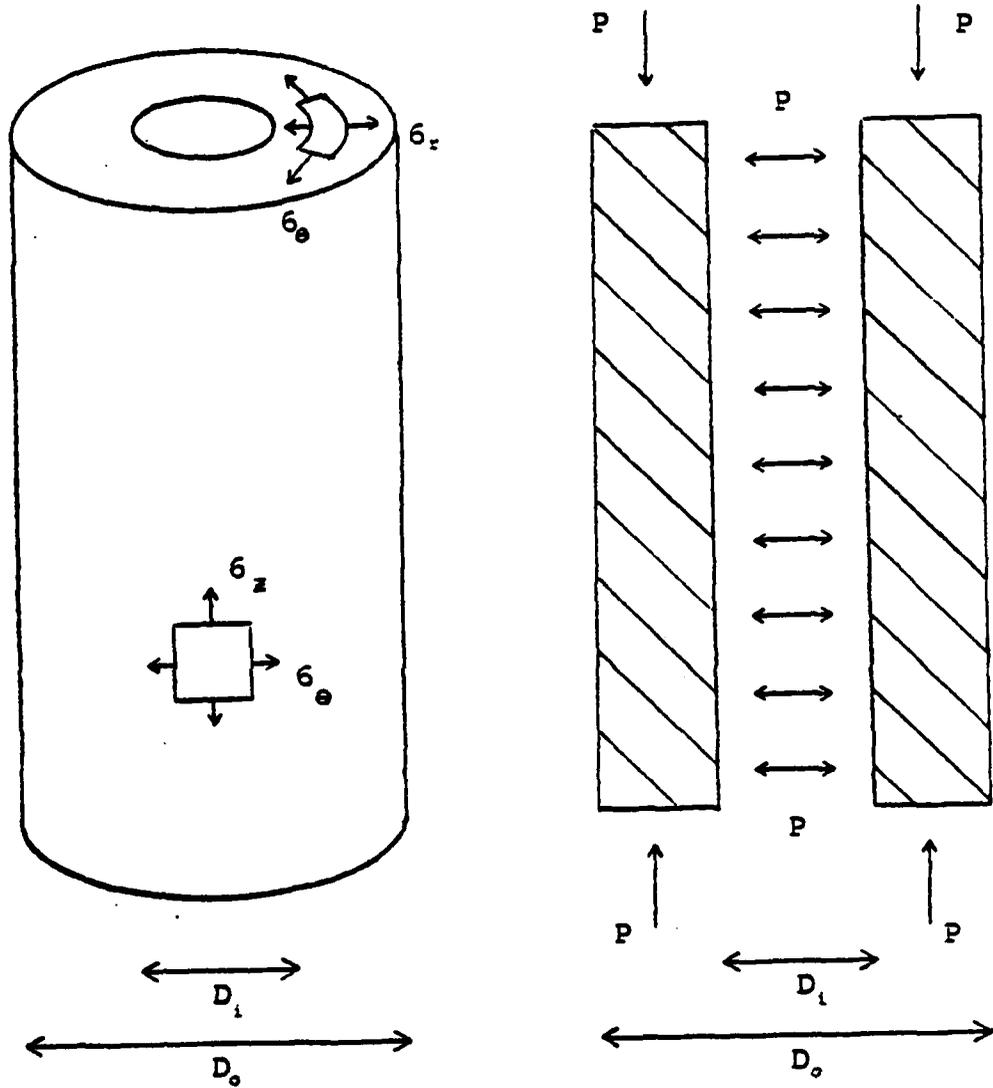
Figure A-1 - Schematic of a condenser tube

Maximum stresses occur at the inner diameter when $d = d_i$

$$\sigma_{r_{max}} = - P_i \qquad (A.6)$$

$$\sigma_{\theta_{max}} = P_i \left[1 + \frac{2}{(d_o/d_i)^2 - 1}\right] \qquad (A.7)$$

$$\sigma_z = - P_i \qquad (A.8)$$

The failure criterion used is the Mises yield criterion, Y [35,36] where

$$Y = \left\{\frac{1}{2}\left[(\sigma_r - \sigma_\theta)^2 + (\sigma_\theta - \sigma_z)^2 + (\sigma_z - \sigma_r)^2\right]\right\}^{1/2} \qquad (A.9)$$

For a smooth surface tube this reduces to

$$Y = 2 P_i \left[1 + \frac{1}{(d_o/d_i)^2 - 1}\right] \qquad (A.10)$$

or

$$Y = P_i + \sigma_{\theta_{max}}$$

For a fluted tube, a stress concentration occurs at the bottom of the flute valley. The mechanical restriction to selecting a fluted tube geometry is to have maximum stresses in the fluted tube which do not exceed a yield criterion which is no greater than the yield criterion for a smooth tube. Since $\sigma_r = 0$ at the surface of the tube, the Tresca yield criterion is appropriate in the form:

$$\frac{Y}{2} = \frac{\sigma_{\theta_{flute}} - \sigma_z}{2} \qquad (A.11)$$

or

$$Y = P_i + \sigma_{\theta_{flute}}$$

The tangential stress for the flute tube at the bottom of the flute, where $d = d_o$, is

$$\sigma_\theta = \frac{\alpha \, P_i \left[(d_x/d_o)^2 + 1 \right]}{(d_x/d_i)^2 - 1} \qquad (A.12)$$

and $d_x \equiv d_o + 4a$ \qquad (A.13)

α is a stress intensity or stress concentration factor. By requiring the yield criterion in equation (A.11) to be less than or equal to the yield criterion in equation A-2, a relationship between maximum stress for the two tubes is

$$\frac{\sigma_{\theta_{flute}}}{\sigma_{\theta_{max}}} \leq 1 \qquad (A.14)$$

For a given fluted tube, compared to a smooth tube of the same inner and outer diameters, a maximum value for α can be determined. Neuber [37] shows a method to relate the amplitude-to-pitch ratio, a/p, with the stress intensity factor. Thus a maximum value for a/p can be determined. This concentration factor for a sine flute can be shown to be

$$\alpha = 1 + \sqrt{\frac{\gamma \; 2a}{r_0}} \qquad (A.15)$$

and

$$r_0 = \frac{a}{(2\pi a/p)^2} \qquad (A.16)$$

Therefore,

$$\alpha = 1 + \frac{a}{p} \; 2\pi \sqrt{2\gamma} \qquad (A.17)$$

γ is the load relieving factor which is described in reference [37] and accounts for the presence of many flutes on the surface of the tube. It is presented in Figure (A-2).

Y can be related to some mechanical property of the tube material such as yield stress or fatigue limit. For a submarine, a very conservative safety factor of 5 can be assumed [38]. Therefore,

$$Y \leq .2YS \qquad (A.10)$$

From reference [2]

$$P_i = h/2248 \text{ Ksi} \qquad (A.19)$$

with h, the design depth in feet.

Then for a smooth tube

$$.2YS = 2\left(\frac{h}{2248}\right) \left[1 + \frac{1}{(d_o/d_i)^2 - 1}\right] \qquad (A.20)$$

and for a fluted tube

$$.2YS = \left(\frac{h}{2248}\right) \left[\frac{\alpha\left\{(d_x/d_o)^2 + 1\right\}}{(d_x/d_i)^2 - 1} + 1\right] \qquad (A.21)$$
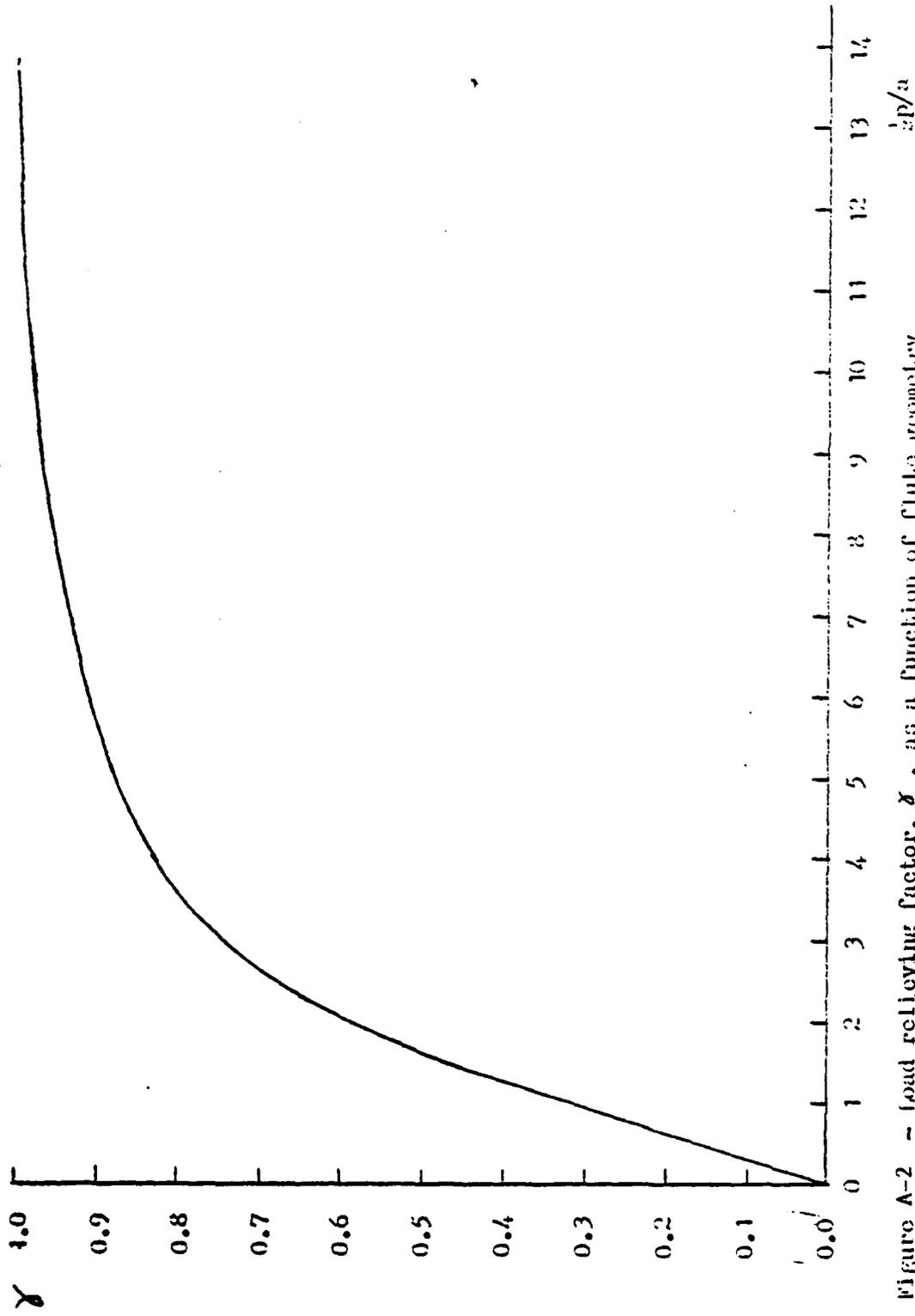
Figure A-2 — load relieving factor, γ, as a function of flute geometry.

Both of these equations contain the constraints due to material strength, design depth, and tube geometry. The tube wall thickness for a smooth tube can be readily calculated from equation (A.20), and the flute limitations can then be determined from equations (A.19) and (A.17) which will ensure that the yield criterion is not violated due to stress concentrations.

An example calculation is provided.

EXAMPLE

Two different materials are considered for a 2000 ft design depth condenser. These are Cu-Ni 70-30 and Ti. Other specifications are:

$$d_o = .625 \text{ in}$$
$$a = .018 \text{ in}$$
$$d_x = .697 \text{ in}$$
$$YS(Cu-Ni) = 25 \text{ Ksi}$$
$$YS(Ti) = 40 \text{ Ksi}$$

From equation (A.20), the inner diameter, d_i, can be calculated and hence the wall thickness, t. Wall thickness, however, is rounded off to the nearest even gauge.

	Cu-Ni	Ti
d_i (in)	.5016	.5511
t (in)	.0617	.0369
d_i (rounded off) (in)	.495	.555
t (rounded off) (in)	.065	.035

Using the rounded values in equation (A.21) to calculate, α, and then using α to determine maximum a/p gives

	Cu-Ni	Ti
α	2.0235	2.0559
γ	.82	.81
a/p	.13	.13

In light of the conservative allowance for yield criterion select a/p in both cases as a/p = .15.

APPENDIX B

Calculations for Condensate Film Thickness

Nomenclature

a	amplitude of the flute (ft)
A	area (ft^2)
B	dimensional group defined in equation (B.6)
g_c	gravitational constent (lbm ft/lbf hr^2)
h	heat transfer coefficient (BTU/hrft^2°F)
h_{fg}	latent heat (BTU/lbm)
K	thermal conductivity (BTU/hrft°F)
N_u	Nusselt number
p	pitch of the flute (ft)
P	pressure (lbf/ft^2)
q	heat flow (BTU/hr)
r	radius of curvature of the fluted sur-face (ft)
S_v	flute half perimeter (ft)
S_c	flute arc length on which condensing occurs (ft)
T	temperature (°F)
ΔT	$T_s - T_w$ (°F)
V	velocity (ft/hr)

$$\left.\begin{array}{c} s \\ y \\ z \end{array}\right\}$$ tube surface coordinates

Greek letters

α height of the condensate in the center
of the flute (ft)

Γ mass flow in S direction per unit length
(lbm/hrft)

δ condensate film thickness (ft)

θ angular representation of a point
along the flute surface

μ dynamic viscosity (lbm/hrft)

ρ density (lbm/ft^3)

σ surface tension (lbf/ft)

ϕ non-dimensional group defined in eqn (B.12)

Ω non-dimensional group defined in eqn (B.6)

Subscripts

c   coolant

f   at flooding conditions

i   inside

n   nominal

o   outside/initial conditions

s   saturation

w   wall

Symbols

−   average

The surface considered is shown in Figure III-2. The following assumptions are stated explicitly and apply to Figure III-2 [7,9,12,13].

1. The flow of the condensate from peak to valley on the flute surface is thin laminar flow.

2. Because the film is thin, intertia forces are negligible.

3. There is no interfacial shear stress between the liquid and the vapor.

4. There is no flow in the y direction. Flow in the z direction occurs only within the trough of height α.

From a momentum balance on the surface specified,

$$\mu \; \frac{d^2 v_s}{d y^2} \; = \; \frac{dP}{dS} \; = \; \frac{d}{dS} \left(\frac{\sigma}{r} \right) \tag{B.1}$$

Integrate this with boundary conditions

$$\frac{dV_s}{dy} = 0 \text{ at } y = \delta$$

and

$$V_s = 0 \text{ at } y = 0$$

The equation for velocity is

$$V_S = \frac{1}{\mu} \frac{dP}{dS} \left(\frac{y^2}{2} - \delta y \right)$$

(B.2a)

and

$$\overline{V}_S = \frac{1}{\delta} \int_0^\delta V_S d_y = - \frac{1}{\mu} \frac{dP}{dS} \frac{\delta^2}{3}$$

(B.2b)

Define the mass flow rate per unit length, Γ.

$$\Gamma = \rho \overline{V}_S \delta = - \frac{\rho}{\mu} \frac{dP}{dS} \frac{\delta^3}{3}$$

(B.3a)

and

$$\frac{d\Gamma}{dS} = - \frac{d}{dS} \left[\frac{\rho}{\mu} \frac{dP}{dS} \frac{\delta^3}{3} \right]$$

(B.3b)

From an energy balance

$$q/A_S = \frac{K \Delta T}{\delta} = h_{fg} \frac{d\Gamma}{dS}$$

(B.4)

Solve eqn (B.4) for $\frac{d\Gamma}{dS}$ and set equal to eqn (B.3b).

END

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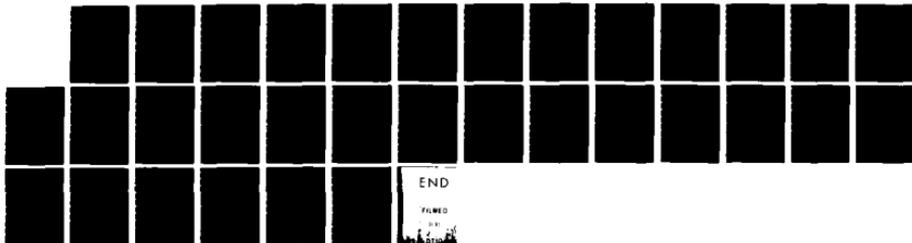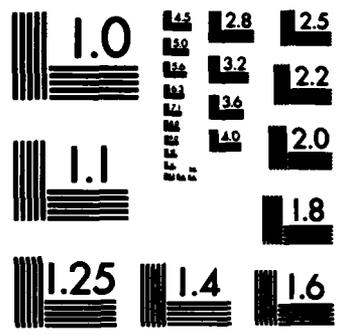
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$$\frac{K\Delta T}{\sigma h_{fg}} = - \frac{d}{dS} \left[\frac{\rho}{\mu} \frac{\delta^3}{3} \frac{d}{dS} \left(\frac{\sigma}{r} \right) \right] \tag{B.5a}$$

which, for constant physical properties becomes

$$\frac{\mu K\Delta T}{\rho \sigma h_{fg} g_c} = \frac{\delta}{3} \frac{d}{dS} \left[\delta^3 \frac{d}{dS} \left(\frac{1}{r} \right) \right] \tag{B.5b}$$

Define Ω,

$$\Omega \equiv \frac{\mu K\Delta T}{\rho \sigma h_{fg} g_c a} = \frac{B\Delta T}{a} \tag{B.6}$$

So eqn. (B.5b) may be written as

$$\Omega = - \frac{\delta}{3a} \frac{d}{dS} \left[\delta^3 \frac{d}{dS} \left(\frac{1}{r} \right) \right] \tag{B.5c}$$

For a sine function flute the radius of curvature, r, is

$$r = \frac{a \left\{ 1 + \left(\frac{2\pi a}{p} \right)^2 \sin^2 \theta \right\}^{3/2}}{\left(\frac{2\pi a}{p} \right)^2 \cos\theta} \tag{B.7}$$

The arc length, S, is

$$S = \int_0^\theta \frac{p}{2\pi} \left\{ 1 + \left(\frac{2\pi a}{p} \right)^2 \sin^2 \phi \right\}^{1/2} d\phi \tag{B.8a}$$

as shown in Figure III-2.

$$\theta_c = \pi\left(1 - \frac{\alpha}{2a}\right) \quad \text{at } S_c \qquad (B.8b)$$

For given values of a, p, and Ω, eqn. (B.5c) can be solved numerically [7,12]. By symmetry

$$\frac{d\delta}{dS} = 0 \quad \text{at} \quad \theta = 0$$

Eqn. (B.5c) becomes

$$-\frac{d}{dS}\left[\frac{d}{dS}\left(\frac{1}{r}\right)\right]\Bigg|_{\substack{S=0 \\ \theta=0}} = \frac{3a\Omega}{\delta_o^4} \qquad (B.9)$$

which gives the initial value for the boundary layer thickness, δ_o.

$$\delta_o = \frac{p}{2\pi}\left[\frac{3\Omega}{1 + 3\left(\frac{2\pi a}{p}\right)^2}\right]^{1/4} \qquad (B.10)$$

Integrating equation (B.5C) gives

$$- \int_{r_o}^{r_s} d\left(\frac{1}{r}\right) = 3a\Omega \int_0^S \frac{1}{\delta^3} \left[\int_0^S \frac{dS}{\delta} \right] dS \qquad (B.11)$$

Define ϕ such that

$$\phi \equiv \int \frac{dS}{\delta} \qquad (B.12a)$$

and by finite difference

$$\phi_{n+1} = \frac{\Delta S_n}{\delta_n} + \phi_n \qquad (B.12b)$$

where

$$\Delta S_n = S_{n+1} - S_n \qquad (B.13)$$

Eqn. (B.11) may be written in the form of a trapezoid rule integration to give

$$\frac{1}{r_n} - \frac{1}{r_{n+1}} = \frac{3a\Omega}{\delta_n^3} \left[\phi_{n+1} + \phi_n \right] \frac{\Delta S_n}{2} \qquad (B.14a)$$

which becomes

$$\frac{1}{r_n} - \frac{1}{r_{n+1}} = \frac{3a\Omega}{\delta_n^3} \left[2\phi_n + \frac{\Delta S_n}{\delta_n} \right] \frac{\Delta S_n}{2} \tag{B.14b}$$

δ_n can now be determined for $\Delta\theta$ increments, with the initial conditions

$\quad \delta_0 \quad$ from eqn (B.10)

$\quad \phi_0 = 0$

$\quad S_0 = 0$

Addendum 1 to this appendix is a program written for a programmable calculator to solve eqn. (B.14b)

The average film thickness of the condensate can be calculated

$$\frac{1}{\bar{\delta}} = \frac{1}{S} \int_0^S \frac{dS}{\delta} = \frac{1}{S} \phi(S) \tag{B.15}$$

Define the heat transfer coefficient, h

$$h \equiv \frac{1}{\delta} K \tag{B.16}$$

The average heat transfer coefficient is

$$\bar{h}_s = \frac{1}{\bar{\delta}} K = K \frac{1}{S_c} \int_0^{S_c} \frac{dS}{\delta} = \frac{K}{S_c} \phi(S_c) \tag{B.17a}$$

where S_c is shown in Figure III-2 and is the arc length over which condensation occurs. By definition

$$\bar{h}_s = \frac{q'}{2S_c \Delta T} \qquad \text{(B.17b)}$$

where q' is the heat transfer per unit length over the entire flute perimeter. The heat transfer coefficient on the nominal diameter, h_n, may be obtained from the equation for heat transfer

$$q = h A \Delta T \qquad \text{(B.18a)}$$

From energy conservation,

$$q_s = q_n \qquad \text{(B.18b)}$$

and therefore

$$\bar{h}_n A_n = \bar{h}_s A_s \qquad \text{(B.18c)}$$

Then

$$\bar{h}_n = \frac{2S_c}{p} \bar{h}_s = 2K\phi(S_c) \qquad \text{(B.19)}$$

A Nusselt number may be defined as

$$Nu \equiv \frac{\bar{h}_n p}{K} = 2\phi(S_c) \qquad \text{(B.20a)}$$

and for $S_c = S_v$

$$Nu = Nu_o(S_v) = 2\phi(S_v) \qquad \text{(B.20b)}$$

From equations (B.6), (B.8a), (B.8b), and (III-10)

$$\bar{h}_n = f(\Omega, a/p, w/w_f) \qquad \text{(B.21)}$$

APPENDIX B - ADDENDUM 1

Calculator Program for the

Solution of Equation (B.14b)

This program was written for a Hewlett Packard
HP-41C programmable calculator. It requires two addi-
tional memory modules, and a printer is highly desireable.
Twenty six (26) memories must be allocated. The program
is run by simply executing the program label, "DELTA".
The calculator will respond by inter-actively asking for
inputs. The required inputs are:

a, p, and Ω.

The outputs are for $n = 0$ to $n = 60$,

r_n, S_n, ΔS_n, δ_n, and ϕ_n.

The method of solution is exactly as explained in
Appendix B. Equation (B.8a) is solved by a 10 point
Gauss-Legendre quadrature method for a sine geometry flute.

```
01   LBL  "DELTA"
02   "COMPUTER BOUNDA"
03   "DRY LAYER"
04   AVIEW
05   "AS A FCT. OF AR"
06   "C LENGTH."
07   AVIEW
08   ADV
09   "************"
10   AVIEW
11   "S<N>/PHI<N>=DEL"
12   "TA<N>,AVG"
13   AVIEW
14   "************"
15   AVIEW
16   ADV
17   CLRG                    Clear all memories
18   SCI 4
19   1.488743390 E-1
20   STO 10
21   2.955252247 E-1
22   STO 11
23   4.333953941 E-1
24   STO 12
25   2.692667193 E-1
26   STO 13
27   6.794095683 E-1         Loading into memory
28   STO 14                  locations weighting
29   2.190863625 E-1         values necessary for
30   STO 15                  integration by
31   8.650633667 E-1         quadrature
32   STO 16
33   1.494513492 E-1
34   STO 17
35   9.739065285 E-1
36   STO 18
37   6.667134431 E-2
38   STO 19
39   GTO 10
```

```
40   LBL 11
41   "INTEGRATION"
42   STO 21
43   X<>Y
44   STO 22
45   -
46   2
47   /
48   STO 22
49   RCL 22
50   RCL 21
51   +
52   2
53   /
54   STO 21
55   0
56   STO 00
57   10.021
58   STO 25
59   XEQ 12
60   XEQ 12
61   XEQ 12
62   XEQ 12
63   XEQ 12
64   RCL 00
65   RCL 20
66   *
67   RTN
68   LBL 12
69   "QUADRATURE SUB-"
70   "ROUTINE"
71   RCL IND 25
72   ISG 25
73   STO 22
74   CHS
75   RCL 20
76   *
77   RCL 21
78   +
79   XEQ 14
80   RCL IND 25
81   *
82   ST+ 00
83   RCL 22
84   RCL 20
85   *
86   RCL 21
87   +
88   XEQ 14
89   RCL IND 25
90   *
91   ST+ 00
92   ISG 25
93   RTN
```

Integration by Gauss-Legendre Quadrature

$$\int_{a}^{b} f(x)\,d_x =$$

$$\frac{b-a}{2} \sum_{i=1}^{10} W_i f\left(\frac{Z_i(b-a)+b+a}{2}\right)$$

```
 94 LBL 14
 95 "INTEGRAND"
 96 SIN
 97 RCL 23
 98 *
 99 2
100 *
101 PI
102 *
103 RCL 24
104 /
105 X↑2
106 1
107 +
108 SQRT
109 RCL 24
110 *
111 2
112 /
113 PI
114 /
115 RTN
```

Integrand:

$$\frac{p}{2\pi} \left\{ 1 + \left[\frac{2\pi a}{p}\right]^2 \sin^2\theta \right\}^{1/2}$$

```
116 LBL 10
117 "INITIALIZATION"
118 "ROUTINE"
119 RAD
120 "A<FT>?"
121 PROMPT
122 STO 23
123 " P<FT>?"
124 PROMPT
125 STO 24
126 "OMEGA?"
127 PROMPT
```

Data input: a, p, Ω

128	3	Calculation of initial
129	*	values (continued)
130	RCL 23	
131	*	
132	STO 09	
133	PI	$\Delta\theta = \pi/60$
134	60	
135	/	
136	STO 08	
137	0	
138	STO 02	
139	STO 05	
140	STO 06	
141	STO 26	
142	RCL 23	
143	RCL 24	r_0 from equation (B.7),
144	/	
145	2	$\theta = 0$
146	*	
174	PI	$s_0 = 0$
148	*	
149	X↑2	$\phi_0 = 0$
150	1/X	
151	RCL 23	
152	*	
153	STO 04	
154	"RO<FT>="	
155	ARCL X	
156	AVIEW	
157	0	
158	"SO<FT>="	
159	ARCL X	
160	AVIEW	
161	FIX 3	
162	"PHIO="	
163	ARCL X	
164	AVIEW	

165	RCL 23	Calculation of initial
166	RCL 24	values (continued)
167	/	
168	2	
169	*	
170	PI	
171	*	
172	X↑2	
173	3	
174	*	
175	1	
176	+	
177	RCL 23	
178	*	
179	1/X	
180	RCL 09	
181	*	
182	SQRT	
183	SQRT	
184	RCL 24	
185	*	
186	2	
187	/	
188	PI	
189	/	
190	STO 03	
191	SCI 4	
192	"DELTAO<FT>="	δ_o from equation (B.10)
193	ARCL X	
194	AVIEW	
195	2	
196	STO 25	
197	SF 01	

198	LBL 01	Solution to equation (B.14b)
199	ADV	
200	DSE 25	
201	RCL 25	
202	STO 07	
203	1	
204	-	
205	SF 12	
206	FIX 0	
207	"N="	
208	ARCL X	n
209	AVIEW	
210	CF 12	
211	SCI 4	
212	XEQ 03	Subroutine for equation (B.7)
213	STO 26	
214	STO 04	
215	"R<N> <FT>="	
216	ARCL X	r_n from equation (B.7)
217	AVIEW	
218	RCL 26	
219	RCL 04	
220	X<>Y	
221	STO 04	
222	1/X	
223	CHS	
224	X<>Y	
225	1/X	
226	+	
227	STO 01	
228	0	
229	ENTER↑	
230	RCL 07	
231	ENTER↑	
232	RCL 25	
233	STO 07	
234	RDN	
235	XEQ 11	Integration subroutine
236	STO 27	
237	RCL 05	$S_n = \int_0^{n\Delta\theta} f(\theta)\,d\theta$ from equation (B.8a)
238	"S<N><FT>="	
239	ARCL X	
240	AVIEW	
241	RCL 27	
242	RCL 05	
243	X<>Y	
244	STO 05	
245	-	
246	CHS	
247	"DEL,S<N> <FT>="	ΔS_n from equation (B.13)
248	ARCL X	
249	AVIEW	

```
250     STO 06
251     RCL 07
252     STO 25
253     RCL 03
254     STO 07
255     SCI 4
256     FC?C 01
257     XEQ 02          Subroutine to iterate for
258     "DELTA<N> <FT>="  $\delta_n$ from equation (B.14b)
259     ARCL x
260     AVIEW
261     RCL 06
262     RCL 07
263     /
264     RCL 02          $\phi_n$ from equation (B.12b)
265     FIX 3
266     "PHI<N>="
267     ARCL X
268     AVIEW
269     +
270     STO 02
271     2
272     ST+ 25
273     RCL 25
274     63
275     X=Y?
276     GTO 05
277     GTO 81
```

```
278    LBL 02
279    RCL 03
280    1/X
281    RCL 06
282    *
283    RCL 02
284    2
285    *
286    +
287    RCL 06
288    *
289    2
290    /
291    RCL 09
292    *
293    RCL 01
294    X<=0?
295    GTO 04
296    /
297    3
298    1/X
299    Y↑X
300    SCI 4
301    RCL 03
302    X<>Y
303    STO 03
304    -
305    ABS
306    1 E-7
307    X<=Y?
308    GTO 02
309    LBL 04
310    RCL 03
311    RTN
```

Subroutine to iterate for δ_n from equation (B.14b)

$$|\delta_i - \delta_{i+1}| \leq 10^{-7}$$

For n = 60, δ_{60} has no solution. Replace δ_{60} with δ_{59}.

```
312    LBL 03
313    RCL 08              Subroutine for equation
314    ST* 07                 (B.7)
315    RCL 07
216    SIN
317    RCL 23              r_n  from equation (B.7)
318    2
319    *
320    PI
321    *
322    RCL 24
323    /
324    STO 01
325    *
326    X↑2
327    1
328    +
329    1.5
330    Y↑X
331    RCL 07
332    COS
333    /
334    RCL 01
335    X↑2
336    /
337    RCL 23
338    *
339    RTN
340    LBL 05
341    BEEP
324    ADV
343    "************"
344    AVIEW
345    ADV
346    RTN
347    STOP
348    .END.
```

APPENDIX C

Fluted Condenser Sizing Procedure

Nomenclature

a — amplitude of the flute (ft)

A — area (ft^2)

B — dimensional group defined in Appendix B

D — diameter (ft)

F — number of flutes on the tube surface

g_c — gravitational constant (lbm ft/lbf hr^2)

h — heat transfer coefficient (BTU/hrft^2°F)

h_{fg} — latent heat (BTU/lbm)

k — thermal conductivity (BTU/hrft°F)

l — ligament (ft)

L — length (ft)

M — non-dimensional group defined in Chapter III

N — number of tubes

Nu — Nusselt number

p — pitch of the flute (ft)

P — pressure (in-Hg-abs)

q — heat flow (BTU/hr)

t — tube wall thickness (ft)

T — temperature (°T)

ΔT — $T_s - T_w$ (°F)

U — overall heat transfer coefficient (BTU/hrft^2°F)

V	velocity (ft/hr)
w	axial mass flow of liquid (lbm/hr)
Wt	weight (lbm or tons)

Greek letters

μ	dynamic viscosity (lbm/hrft)
ρ	density (lbm/ft^3)
σ	surface tension (lbf/ft)
Ω	non-dimensional group defined in Chapter III

Subscripts

B	tube bundle
c	coolant
D	tube bundle diameter
f	at flooding conditions
i	inside
n	nominal
o	outside
s	saturation
T	total
w	wall
sc	scale
x	cross section

Symbols

-	average

This procedure will consist of an example which sizes a submarine condenser. As required in reference [11], a submarine condenser has an even number of passes for the coolant flow. A maximum length will be specified.

Design Specificatons

Design depth		- 2000 ft
Material		- Cu-Ni 70-30
Tube:	D_o	= 5/8 in (.625 in)
	t	= .065 in (16 Gauge)
	a	= 18 mils (1.50×10^{-3} ft)
	a/p	= .31205
	F	= 36
	D_n	= .643 in
Coolant:	Sea water	
	Tc_1	= 65.0°F
	V_c	= 8.0 ft/sec
	W_c	= 7900 GPM (4.0559×10^6 lbm/hr)
No. of passes		= 2
No. of passes		= 3294
Condensate:		
	Pressure	= 5.0 in-Hg-abs
	T_{sat}	= 133.7°F
	ρ_ℓ	= 61.554 lbm/ft^3

$$k_\ell \qquad\qquad = .376 \text{ BTU/hrft°F}$$

$$\sigma_\ell \qquad\qquad = .00459 \text{ lbf/ft}$$

$$h_{fg} \qquad\qquad = 1017.4 \text{ BTU/lbm}$$

$$\mu_\ell \qquad\qquad = 1.20 \text{ lbm/hrft}$$

Thermal conductivity of tube wall = 17.0 BTU/hrft°F

$$h_{scale} = 2000 \text{ BTU/hrft}^2\text{°F}$$

Total tube length per pass = 12.0 ft

Preliminary calculations:

$$Nu_o \Omega^{1/4} = 4.612 \text{ from Figure III-4}$$

$$B \qquad = 3.7649 \times 10^{-12} \text{ ft/°F} \qquad \text{eqn. (III-7)}$$

$$W_f \qquad = 6.7157 \text{ lbm/hr} \qquad\qquad \text{eqn. (III-10)}$$

The condenser will be segmented into 6 sections as shown in Figure (IV-1). The length, L, of section 1 will be that length required for condenstate run-off of $w/w_f = 0.060$. This length will be used for sections 2, 5, and 6. The lengths for sections 3 and 4 will be the length necessary to make each pass total length equal to 12.0 ft.

A. Calculation of L_1

1. Specify w/w_f

   then $M = .0820$ from equation (III-19)

2. Assume $Tc_2 = 66°F$; $i = 1$

3. Assume $L = 10.0$ ft; $i \quad 1$

   a. Solve for T_i from equation (III-29)

   b. Solve for $(q/A_n)_i$ from equation (III-25)

   c. Solve for L_{i+1} from equation (III-23)

   d. Repeat a, b, and c until $L_i = L_{i+1}$

4. Solve for Tc_{2j+1} from equation (III-24)

5. Repeat 3 and 4 until $Tc_{2j} = Tc_{2j+1}$

Results are shown in Table C-1.

B. Calculation of $(w/w_f)_2$

1. Specify $L_2 = L_1 = 4.957$ ft.

2. Assume $Tc_2 = 77.36°F$; $j = 1$

3. Assume $\Delta T = 0.50°F$; $i = 1$

   a. Solve for $(q/A_n)_i$ from equation (III-25)

   b. Solve for $(w/w_f)_i$ from equation (III-23)

   c. Solve for M from equation (III-19)

   d. Solve for ΔT_{i+1} from equation (III-29)

   e. Repeat a-d until $\Delta T_i = \Delta T_{i+1}$

TABLE C-1

Solution for L_1 with $(w/w_f)_1 = 0.060$

j	Tc_{2j} (°F)	i	L_i (ft)	ΔT_i (°F)	$(q/A_n)_i$ (BTU/hr)	L_{i+1} (ft)	Tc_{2j+1} (°F)
1	66.0	1	10.0	.976	33681	4.510	
		2	4.510	2.823	32756	4.638	
		3	4.638	2.720	32807	4.631	
		4	4.631	2.725	32804	4.631	76.36
2	76.36	1	10.0	.976	31419	4.835	
		2	4.835	2.573	30606	4.963	
		3	4.963	2.484	30651	4.956	
		4	4.956	2.489	30649	4.957	76.36

Results:

$(w/w_f)_1 = 0.060$

$L_1 = 4.957$ ft

$\Delta T_1 = 2.489$ °F

$(q/A_n)_1 = 30649$ BTU/hrft^2

$Tc_2 = 76.36$ °F

4. Solve for Tc_{2j+1} from equation (III-22)

5. Repeat 3 and 4 until $Tc_{2j} = Tc_{2j+1}$

Results are shown in Table C-2.

Calculation of all remaining length sections pro-
ceeds as for section 2, with the desired length speci-
fied. The coolant exit temperature for each section is
the inlet temperature for the succeeding section.
Results for the entire condenser are shown in Table C-3.

A check on the solution to equation (III-22)
verify the results in Table C-3.

$$\bar{h} = .6027 \left[\frac{h_{fg} \, w_f}{L \, \Delta T} \right]^{.0774} \frac{a^{.2307} \, [f(a/p)]}{p}^{.9226} \left[\frac{K^3 \rho c h_{fg} g_c}{\mu \Delta T} \right]^{.2307}$$

$$(III-22)$$

where $f[a/p] = Nu_o \Omega^{1/4}$

This should be compared to

$$\frac{(q/A_n)}{\Delta T} = \bar{h} \quad \text{shown in Table C-3.}$$

Table C-4 shows this comparison.

TABLE C-2

Solution for $(w/w_f)_2$ with $L_2 = L_1 = 4.957$ ft

j	Tc_{2j} (°F)	i	ΔT_i (°F)	$(q/A_n)_i$ (BTU/hr)	$(w/w_f)_i$	T_{i+1} (°F)	Tc_{2j+1} (°F)
1	77.36	1	0.50	29056	.057	2.304	
		2	2.304	28126	.055	2.198	
		3	2.198	28181	.055	2.205	
		4	2.205	28177	.055	2.204	86.80
2	86.80	1	0.50	26828	.053	2.053	
		2	2.053	26015	.051	1.964	
		3	1.964	26062	.051	1.969	
		4	1.969	26059	.051	1.969	86.01
3	86.01	1	0.50	27026	.053	2.075	
		2	2.075	26203	.051	1.985	
		3	1.985	26250	.051	1.990	
		4	1.990	26247	.051	1.989	86.08
4	86.08	1	0.50	27008	.053	2.073	
		2	2.073	26186	.051	1.983	
		3	1.983	26233	.051	1.988	
		4	1.988	26230	.051	1.988	86.08

Results: $(w/w_f)_2 = .051$ $\Delta T_2 = 1.988$ °F $Tc_2 = 86.08$°F

$L_2 = 4.957$ ft $(q/A_n)_2 = 26230$ BTU/hrft^2

TABLE C-3

Results for Sizing the Condenser Specified

Section	1	2	3	4	5	6
h_c (BTU/hrft^2°F)	1603	1706	1766	1798	1843	1896
U_w (BTU/hrft^2°F)	507.8	521.1	528.4	532.3	537.4	543.3
$\bar{h}_{condensing}$ (BTU/hrft^2°F)	12312	13197	15302	15800	15178	16441
U_n (BTU/hrft^2°F)	487.7	501.3	510.8	514.9	519.0	525.9
T(°F)	2.489	1.988	1.528	1.377	1.262	0.974
w/w_f	0.060	0.051	0.019	0.018	0.038	0.301
L (ft)	4.957	4.957	2.086	2.086	4.957	4.957
q/A_n (BTU/hrft^2)	30649	26230	23383	21759	19160	16011
Tc_1 (°F)	65.0	76.36	86.08	89.72	93.12	100.22
Tc_2 (°F)	76.36	86.08	89.72	93.12	100.22	106.15

$q_{total} = 1.57 \times 10^8$ BTU/hr

TABLE C-4

Comparison Between Equation (III-22)

and Calculated Results

Section	$\bar{h}_n$ by eqn (22)	$\bar{h}_n$ by iterative solution
1	12322	12312
2	13206	13197
3	15313	15302
4	15812	15800
5	15190	15178
6	16452	16441

C.   Sizing the Tube Bundle Diameter

The number of tubes across the diameter, N_D, of a circular bundle containing N_T total number of tubes is

$$N_D = \sqrt{\frac{4N_T}{\pi}} \qquad (C.1)$$

The geometric diameter for the individual tube used in sizing the tube bundle is the extreme diameter, D_x

$$D_x = D_o + 4a \qquad (C.2)$$

The physical separation in the tube sheet between outer edges of adjacent tubes is termed the ligament, l. Minimum ligament distance is 0.212 in. [42]. This limit is set by manufacturing constraints. Thus the tube bundle diameter, D_B, is

$$D_B = N_D \cdot D_x + (N_D - 1) \cdot l \qquad (C.3)$$

For the condenser considered,

$$N_D = 65$$

$$D_x = .6970 \text{ in.}$$

$$D_B = 58.87 \text{ in.}$$

Allowing 5 percent for steam flow passages, structural members, air ejectors, and downcomers gives an external diameter for the condenser bundle

$$D_B = 61.82 \text{ in.}$$

D.   Weight Estimation of the Tube Bundle

The weight of an individual tube can be determined by the product of the tube length, L, tube cross section, A_x, and material density. Thus the weight of the tube bundle is

$$Wt_B = A_x \, L \, \rho \, N_T \qquad (C.4)$$

The cross sectional area for a sine geometry flute is approximated by

$$A_x = \frac{(D_n^2 - D_i^2)}{4} \qquad (C.5)$$

For the condenser considered,

$$A_x = .1507 \text{ in}^2$$
$$Wt_B = 23091 = 10.31 \text{ tons}$$

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