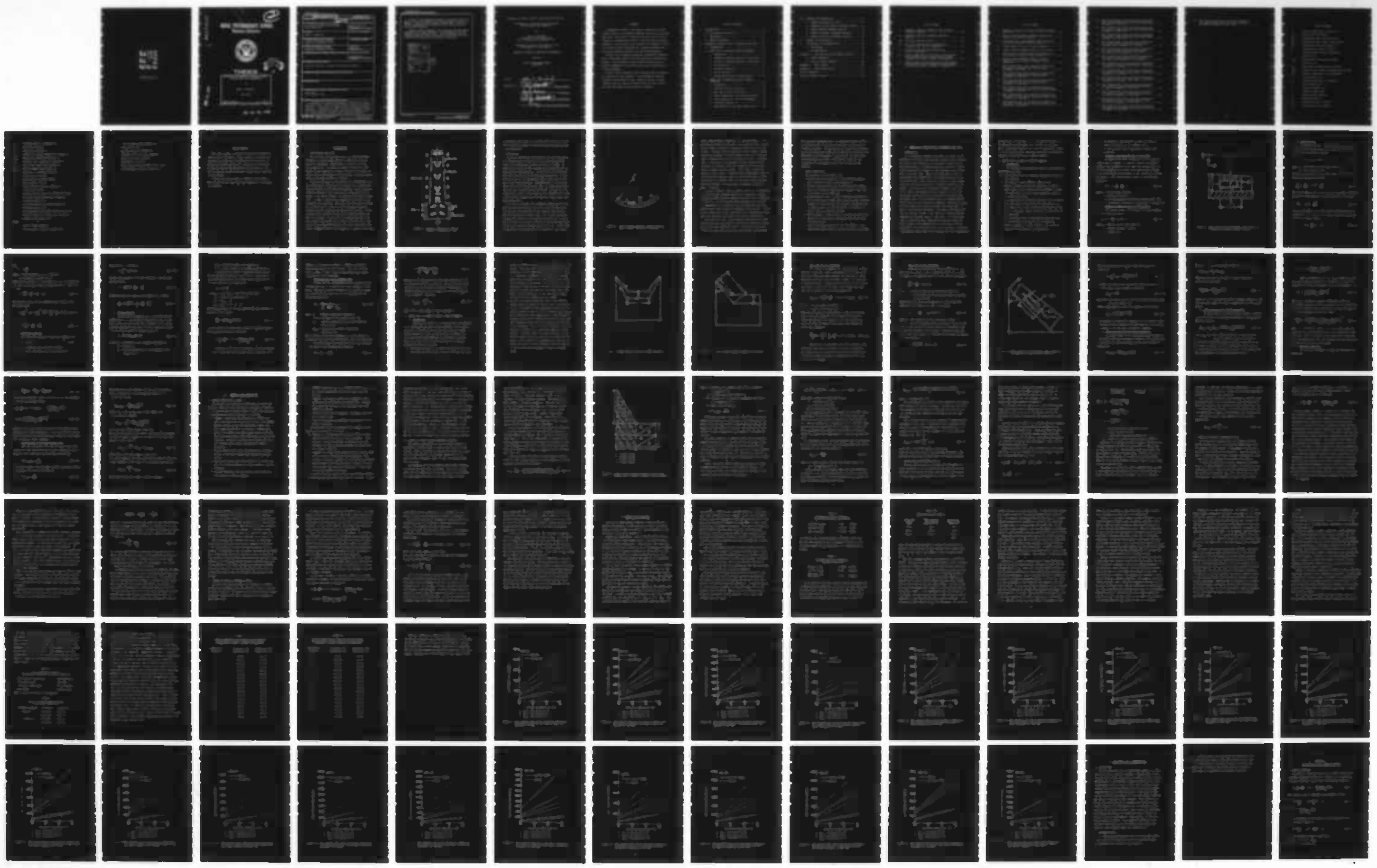
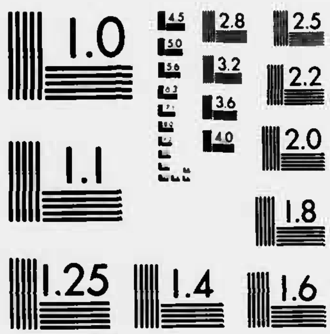
AD-A132 141 AN ANALYSIS OF SMOOTH AND AXIALLY FINNED ROTATING HEAT PIPE CONDENSERS(U) NAVAL POSTGRADUATE SCHOOL MONTEREY CA A F KLEINHOLZ JUN 83 1/2

UNCLASSIFIED F/G 12/1 NL





MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

AD A132141

NAVAL POSTGRADUATE SCHOOL

Monterey, California

DTIC
ELECTE
SEP 7 1983
S D
D

THESIS

AN ANALYSIS OF SMOOTH AND AXIALLY FINNED,
ROTATING HEAT PIPE CONDENSERS

by

Adam F. Kleinholz

June 1983

Thesis Advisor: P. J. Marto

Approved for public release, distribution unlimited.

DTIC FILE COPY

83 09 06 080

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER	2. GOVT ACCESSION NO. A132 141	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) An Analysis of Smooth and Axially Finned, Rotating Heat Pipe Condensers		5. TYPE OF REPORT & PERIOD COVERED Master's Thesis; June 1983
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Adam F. Kleinholz		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		12. REPORT DATE June 1983
		13. NUMBER OF PAGES 176
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release, distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Heat Pipe
Rotating Heat Pipe

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A mathematical model is developed to determine the heat transfer rate of a cylindrical condenser section of a rotating heat pipe. This model is coupled to an existing code and an analysis is accomplished on both a smooth and axially finned condenser. The results of this analysis are compared to those of a similar analysis performed on a tapered condenser heat pipe using identical geometric and operating parameters.

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE
S/N 0102-LF-014-6601

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

Results of the comparison indicate cylindrical condensers have a lower heat transfer rate than equivalent tapered condensers. This reduction in heat transfer rate is due to a greater condensate film thickness and is most significant in a smooth condenser.

Axially finned condensers with triangular and rectangular fin profiles are also compared. The rectangular fins are assumed to have adiabatic tips. Results indicate the heat transfer rates for these two profiles vary by only 0.40 per cent for both tapered and cylindrical condensers.

Accession For
NTIS GRA&I ☒
DTIC TAB ☐
Unannounced ☐
Justification

By
Distribution/
Availability Codes

Dist	Avail and/or Special
A	

DTIC COPY INSPECTED 2

Approved for public release, distribution unlimited.

An Analysis of Smooth and Axially Finned,
Rotating Heat Pipe Condensers

by

Adam F. Kleinholz
Lieutenant, United States Navy
B.S., University of Oklahoma, 1975

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

NAVAL POSTGRADUATE SCHOOL
June 1983

Author: Adam F Kleinholz

Approved by: P.J. Marto, Thesis Advisor

David Salinas, Co-Advisor

P. J. Marto, Chairman, Department of Mechanical Engineering

[illegible] Dyer, Dean of Science and Engineering

ABSTRACT

A mathematical model is developed to determine the heat transfer rate of a cylindrical condenser section of a rotating heat pipe. This model is coupled to an existing code and an analysis is accomplished on both a smooth and axially finned condenser. The results of this analysis are compared to those of a similar analysis performed on a tapered condenser heat pipe using identical geometric and operating parameters.

Results of the comparison indicate cylindrical condensers have a lower heat transfer rate than equivalent tapered condensers. This reduction in heat transfer rate is due to a greater condensate film thickness and is most significant in a smooth condenser.

Axially finned condensers with triangular and rectangular fin profiles are also compared. The rectangular fins are assumed to have adiabatic tips. Results indicate the heat transfer rates for these two profiles vary by only 0.40 per cent for both tapered and cylindrical condenser.

TABLE OF CONTENTS

I. INTRODUCTION-- 15

A. THE ROTATING HEAT PIPE---------------------------- 15

B. BACKGROUND-- 17

C. THESIS OBJECTIVES---------------------------------- 20

II. THEORETICAL ANALYSIS FOR A CYLINDRICAL HEAT PIPE---- 21

A. INTRODUCTION-- 21

B. THEORY FOR A CYLINDRICAL SMOOTH CONDENSER------- 22

1. Assumptions---------------------------------- 22

2. Condensate Momentum Equation (X-Direction)-- 23

3. Condensate Momentum Equation (Y-Direction)-- 23

4. Fluid Velocity------------------------------- 25

5. Continuity Equation-------------------------- 26

6. Energy Equation------------------------------ 27

7. Determination of Heat Transfer Rate--------- 29

C. THEORY FOR A CYLINDRICAL AXIALLY FINNED CONDENSER--- 30

1. Assumptions---------------------------------- 30

2. Mass Flow in the X-Direction---------------- 34

3. Mass Flow in the Z-Direction---------------- 35

4. Energy Equation for the Trough Condensate--- 37

5. Energy Equation for the Fin Condensate------ 38

6. Continuity Equation-------------------------- 39

7. Determination of the Heat Transfer Rate----- 40

III. COMPUTER CODE DESCRIPTION------------------------ 42

A. GENERAL DESCRIPTION OF CODE------------------- 42

B. INTERNALLY FINNED TAPERED CONDENSER SOLUTION--- 44

C. SMOOTH TAPERED CONDENSER SOLUTION------------- 49

D. SMOOTH CYLINDRICAL CONDENSER SOLUTION---------- 52

E. FINNED CYLINDRICAL CONDENSER SOLUTION---------- 56

IV. RESULTS AND DISCUSSIONS-------------------------- 60

V. CONCLUSIONS AND RECOMMENDATIONS------------------ 93

A. CONCLUSIONS---------------------------------- 93

B. RECOMMENDATIONS------------------------------ 93

APPENDIX A: FILM PROFILE FINITE ELEMENT SOLUTION-------- 95

A. SMOOTH CONDENSER----------------------------- 95

B. AXIALLY FINNED CONDENSER--------------------103

APPENDIX B: USER'S MANUAL----------------------------107

APPENDIX C: SOURCE CODE LISTING----------------------117

LIST OF REFERENCES-----------------------------------174

INITIAL DISTRIBUTION LIST----------------------------176

LIST OF TABLES

I. Condenser Geometric Parameters Held Constant During All Analyses ---------- 62

II. Condenser Geometric Parameters Applied as Required ---------- 62

III. Operating Parameter Matrix ---------- 63

IV. List of Parameters Used in Rectangular/Triangular Fin Profile Comparison ---------- 68

V. Results of Rectangular/Triangular Fin Profile Comparison ---------- 68

VI. Surface Temperature Distribution for Rectangular and Triangular Axially Finned Tapered Copper Condensers at Middle Increment of Condensers ------ 70

VII. Surface Temperature Distribution for Rectangular and Triangular Axially Finned Cylindrical Copper Condensers at Middle Increment of Condensers ------ 71

LIST OF FIGURES

1. Schematic Drawing of a Tapered Condenser Rotating Heat Pipe-- 16

2. Axially Finned Condenser Geometry Showing Fins, Troughs, and Lines of Symmetry---------------------- 18

3. Cross Section of Infinitesimal Fluid Element on Cylindrical Condenser Internal Surface-------------- 24

4. Trough Section of Axially Finned Condenser---------- 32

5. Axially Finned Condenser Symmetric Fin Section------ 33

6. Cross Section of Infinitesimal Fluid Element on Fin Surface of Axially Finned Condenser------------- 36

7. Axially Finned Condenser Symmetric Section Subdivided into 25 Linear Triangular Finite Elements-- 46

8. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 700 RPM------------------------------ 73

9. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 1400 RPM----------------------------- 74

10. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 2800 RPM----------------------------- 75

11. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Stainless Steel Condensers at 700 RPM------------------------------ 76

12. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Stainless Steel Condensers at 1400 RPM----------------------------- 77

13. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Stainless Steel Condensers at 2800 RPM----------------------------- 78

14. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Copper Condensers at 700 RPM---------------------------------- 79

15. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Copper Condensers at 1400 RPM--------------------------------- 80

16. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Taperd Copper Condensers at 2800 RPM--------------------------------- 81

17. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 700 RPM-------------- 82

18. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 1400 RPM------------- 83

19. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 2800 RPM------------- 84

20. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 700 RPM---------------------------------- 85

21. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 1400 RPM--------------------------------- 86

22. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 2800 RPM--------------------------------- 87

23. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stainless Steel Cylindrical Condensers at 700 RPM------------------ 88

24. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stailness Steel Cylindrical Condensers at 1400 RPM----------------- 89

25. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stainless Steel Cylindrical Condensers at 2800 RPM----------------- 90

26. Heat Transfer Rate versus Saturation Temperature for Copper and Stainless Steel Smooth Cylindrical Condensers at 1400 RPM--------------------------------- 91

27. Heat Transfer Rate versus Saturation Temperature for Copper and Stainless Steel Axially Finned Cylindrical Condensers at 1400 RPM------------------ 92

TABLE OF SYMBOLS

A	cross sectional area for flow (ft^2)
c_p	specific heat (Btu/lbm-F)
$d\dot{m}$	differential mass flow rate (lbm/hr)
dq	differential heat transfer rate (Btu/hr)
dq''	differential heat flux (Btu/hr-ft^2)
g	acceleration of gravity (ft/hr^2)
h	convective heat transfer coefficient (Btu/hr-ft^2-F)
h_{ext}	external heat transfer coefficient (Btu/hr-ft^2-F)
h_{fg}	latent heat of vaporization (Btu/lbm)
$\bar{h}_{fg}$	corrected latent heat of vaporization (Btu/lbm)
k_f	thermal conductivity of the condensate film (Btu/hr-ft-F)
k_w	thermal conductivity of condenser wall (Btu/hr-ft-F)
ℓ	length of element (ft)
L	condenser length (ft)
$\dot{m}$	mass flow rate (lbm/hr)
P	pressure (lbf/ft^2)
P_v	pressure of vapor (lbf/ft^2)
Q	heat transfer rate (Btu/hr)

r	internal radius of condenser (ft)
R	relaxation variable
T	temperature (degrees F)
T_{avg}	average fin surface temperature (degrees F)
T_{sat}	saturation temperature (degrees F)
T_w	condenser wall temperature (degrees F)
T_{fin}	fin surface temperature (degrees F)
T_∞	ambient temperature (degrees F)
thick	thickness of condenser wall (ft)
u	fluid velocity (ft/hr)
$\bar{u}$	average fluid velocity (ft/hr)
v	vapor velocity (ft/hr)
w	fluid velocity along fin surface in z-direction (ft/hr)
$\bar{w}$	average fluid velocity in z-direction (ft/hr)
x	coordinate measuring distance along the condenser length (ft)
y	coordinate measuring distance normal to condenser surface (ft)
z	coordinate measuring along surface of fin (ft)
z^*	distance along fin surface from fin tip to trough film thickness (ft)

GREEK

α	fin half angle (degrees)
δ	film thickness along fin surface (ft)

δ^*	film thickness along condenser wall (ft)
ε	trough width (ft)
ω	angular velocity (radians/hr)
ϕ	condenser cone half angle (degrees)
ρ_f	density of liquid (lbm/ft^3)
τ	shear stress (lbf/ft^2)
τ_v	vapor liquid interface shear stress (lbf/ft^2)
μ	liquid dynamic viscosity (lbm/ft-hr)

ACKNOWLEDGMENT

The author wishes to express his sincere gratitude and appreciation to Dr. Paul J. Marto, Professor of Mechanical Engineering and Dr. David Salinas, Associate Professor of Mechanical Engineering for their unselfish assistance and guidance throughout the development of this thesis. The author feels he was most fortunate in that he had "the best of both worlds."

Finally, the author wishes to thank his wife, Nanci, who pointed out the light when there appeared to be nothing but darkness.

I. INTRODUCTION

A. THE ROTATING HEAT PIPE

The rotating, wickless heat pipe is a closed container designed to transfer large amounts of heat from rotating machinery components. Essentially, it consists of three main components: a cylindrical evaporator section, a condenser section which may be either tapered or cylindrical in shape, and a fixed amount of working fluid. A typical tapered rotating heat pipe is shown in Figure 1.

When the heat pipe is rotated about its longitudinal axis at a speed above a certain critical value, the working fluid forms an annulus in the evaporator section. Note in Figure I that the diameter of the evaporator is larger than the condenser. This larger diameter provides a greater liquid reservoir. As heat is added to the evaporator, the fluid in the evaporator will vaporize. The vapor will flow axially towards the condenser as a result of a slight pressure difference, transporting the latent heat of vaporization with it. In the condenser end, external cooling of the condenser causes the vapor to condense. In the case of a tapered heat pipe, the centrifugal force due to the rotation of the pipe has a component acting along the condenser wall which accelerates the liquid condensate back to the evaporator to complete the cycle.

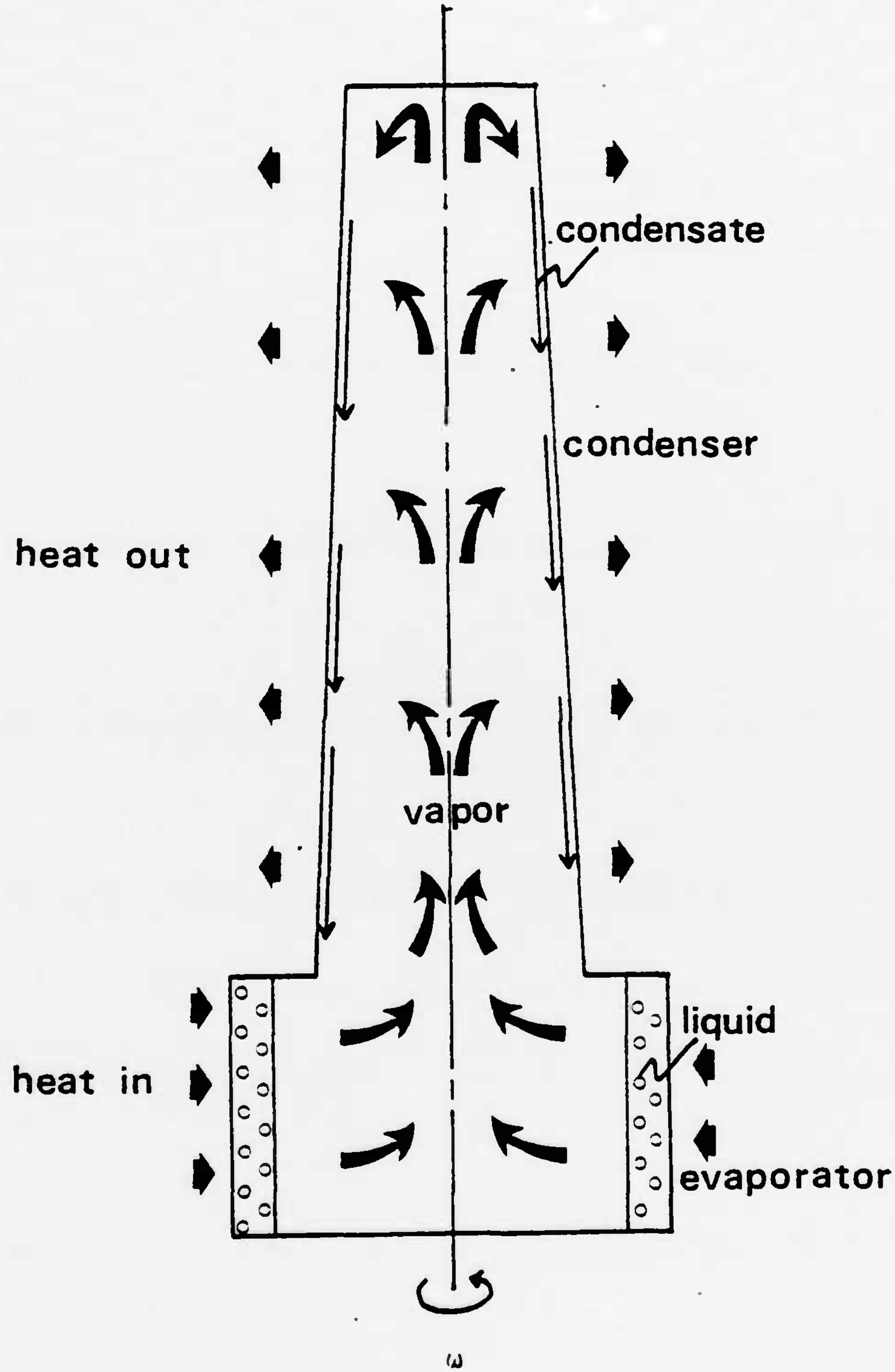


Figure 1. Schematic Drawing of a Tapered Condenser Rotating Heat Pipe

A cylindrical heat pipe, on the other hand relies on a hydrostatic pressure gradient to drive the liquid condensate back to the evaporator.

B. BACKGROUND

The first theoretical investigation into the performance of a tapered rotating heat pipe at the Naval Postgraduate School was accomplished by Ballback [Ref. 1] in 1969. He examined the limits in heat transfer controlled primarily by fluid dynamic considerations. In particular, he considered the following four limits on heat pipe performance: a) the boiling limit, b) the entrainment limit, c) the sonic limit and d) the condensing limit. Tantrakul [Ref. 2] calculated these limits for a specific heat pipe. He found the condensing limit was the controlling limitation. In fact, the calculated heat transfer rate, based on the condensing limit was 1/10th the heat transfer rate for the next lowest limit, the entrainment limit.

In order to overcome this condensing limitation and thus increase the heat transfer rate of the rotating heat pipe, the concept of an internally finned tapered rotating heat pipe was considered by Schafer [Ref. 3]. Schafer developed an analytical model for this tapered heat pipe with a triangular fin profile as shown in Figure 2. He assumed one dimensional heat conduction through the wall and fin. Corley [Ref. 4] developed a two-dimensional heat conduction model using a

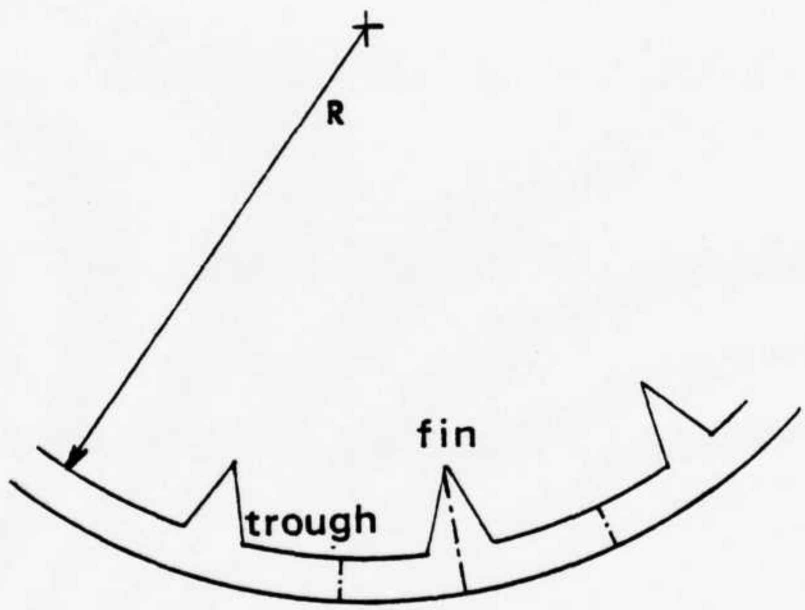


Figure 2. Axially Finned Condenser Geometry Showing Fins, Troughs, and Lines of Symmetry

Finite Element Method formulation for the same geometry. To overcome the problem of few nodal points along the fin surface, Corley assumed a parabolic temperature profile along the fin surface. Tantrakul [Ref. 2] modified Corley's computer code by increasing the number of finite elements from two to three in order to minimize the error at the apex of the fin. Purnomo [Ref. 5] developed a two-dimensional Finite Element Method solution to the steady state heat conduction problem using a linear triangular finite element. Davis [Ref. 6] modified Purnomo's code to make it compatible with COPES/CONMIN [Ref. 8], an optimization program. Davis, in his modification, found a coding error in Purnomo's [Ref. 5] code, that once corrected, permitted Purnomo's corrected code to converge to Schafer's [Ref. 3] results.

Purnomo's [Ref. 5] code is limited in that it is restricted to one particular condenser geometric configuration, namely: an axially finned tapered condenser heat pipe with a triangular fin profile. In that tapered finned condenser are difficult to manufacture, it is doubtful that any widespread practical application of this geometric configuration will result. Cylindrical condensers, on the other hand, can be manufactured with much less difficulty and might find practical application.

This being the case, a more practical and beneficial code would be one that could analyze cylindrical condenser rotating heat pipes, both finned and smooth. In actuality, the most beneficial code would be one that could analyze the following four geometric configurations: 1) tapered-internally finned, 2) tapered-smooth, 3) cylindrical-internally finned, and 4) cylindrical-smooth. The theoretical heat transfer performance of the four geometries could then be compared to determine the advantages and disadvantages of each design. An additional advantage would be gained if different fin profiles, i.e., triangular vice rectangular, could also be analyzed and compared.

C. THESIS OBJECTIVES

The objectives of this thesis are:

1) Develop analytical models for both cylindrical-smooth and cylindrical-axially finned condensers.
2) Develop solution techniques to these analytical models that will account for temperature variations along the axial length of the condenser.
3) Modify Purnomo's [Ref. 5] code to provide a solution to the two-dimensional steady state conduction heat transfer problem for the following four geometric configurations: a) tapered-smooth, b) tapered-finned, c) cylindrical-smooth, and d) cylindrical-finned.
4) Modify Purnomo's [Ref. 5] code to provide the additional capability of analyzing a rectangular fin profile with an adiabatic tip.
5) Obtain and compare results of the four geometric configurations given above for various operating conditions.

II. THEORETICAL ANALYSIS FOR A CYLINDRICAL HEAT PIPE

A. INTRODUCTION

In a cylindrical condenser heat pipe, the radius of the condenser is constant along the axial length of the condenser. The flow of the condensate in the absence of vapor-liquid interfacial shear, is dependent upon the variation in hydrostatic pressure with changes in film thickness along the surface of the heat pipe. Leppert and Nimmo [Refs. 8 and 9] investigated the phenomenon of film condensation on a flat horizontal plate. This situation is similar to film condensation on the inside surface of a rotating cylindrical condenser. In the case of a cylindrical condenser, the body force, rather than being the force of gravity, is now the centrifugal force caused by the rotation of the heat pipe. Weigenseil [Ref. 10] and Tantrakul [Ref. 2] compared experimental results for a cylindrical condenser rotating heat pipe with the theoretical results of Leppert and Nimmo [Refs. 8 and 9] and found good agreement. The Leppert and Nimmo solution was limited in that it was based on a constant surface temperature along the length of the plate. A rotating heat pipe, in actuality, has a temperature variation along the axial length of the condenser which in some cases, may be significant. This being the case, it was necessary to develop a mathematical model which would consider the axial temperature variation in the solution of the heat transfer analysis. In the mathematical development that follows, a cylindrical smooth (unfinned) condenser will first be considered in that it is the simplest case. The model will then be extended to include a cylindrical axially finned condenser.

B. THEORY FOR A CYLINDRICAL SMOOTH CONDENSER

1. Assumptions

In developing the theoretical analysis, the following assumptions are made:

a) Film condensation, not dropwise condensation occurs in the condenser.

b) The condensate film undergoes laminar flow.

c) Momentum changes through the condensate are small. Thus, there is essentially a static balance of forces.

d) The vapor exerts no drag in the condensate; there is no interfacial shear.

e) The temperature distribution within the film is linear.

f) The vapor space is essentially at one pressure, P_v.

g) The density of the fluid is much greater than the density of the vapor. Thus, the density of the vapor can be neglected.

h) The centrifugal force is much greater than the force of gravity and, thus, gravity may be neglected.

i) Velocity gradients in the circumferential direction relative to the pipe wall are negligible.

j) The condensate film thickness is much less than the radius of curvature of the condenser wall.

k) The rotating heat pipe is operating at steady state conditions.

2. Condensate Momentum Equation (X-Direction)

By applying the above assumptions and the coordinate system shown in Figure 3, an analysis similar to Nusselt's original film condensation film theory may be used [Ref. 11]. Based on assumption c, a static force balance may be taken on an infinitesimal fluid element in the x-direction as shown in Figure 3. This force balance results in the following equation:

$$\Sigma F_x = 0 : \frac{\partial \tau}{\partial y} - \frac{\partial p}{\partial x} = 0 \qquad \text{(eqn 2.1)}$$

where τ = shear stress (lbf/ft^2)

p = pressure (lbf/ft^2)

x = co-ordinate measuring distance along surface (ft).

y = co-ordinate measuring distance normal to surface (ft).

3. Condensate Momentum Equation (Y-Direction)

In a similar manner, using Figure 3, a force balance in the y-direction yields:

$$\Sigma F_y = 0 : \frac{\partial p}{\partial y} + \rho_f \omega^2 r = 0 \qquad \text{(eqn 2.2)}$$

where ρ_f = density of the fluid (lbm/ft^3)

ω = angular velocity (rad/hr)

r = radius (ft)

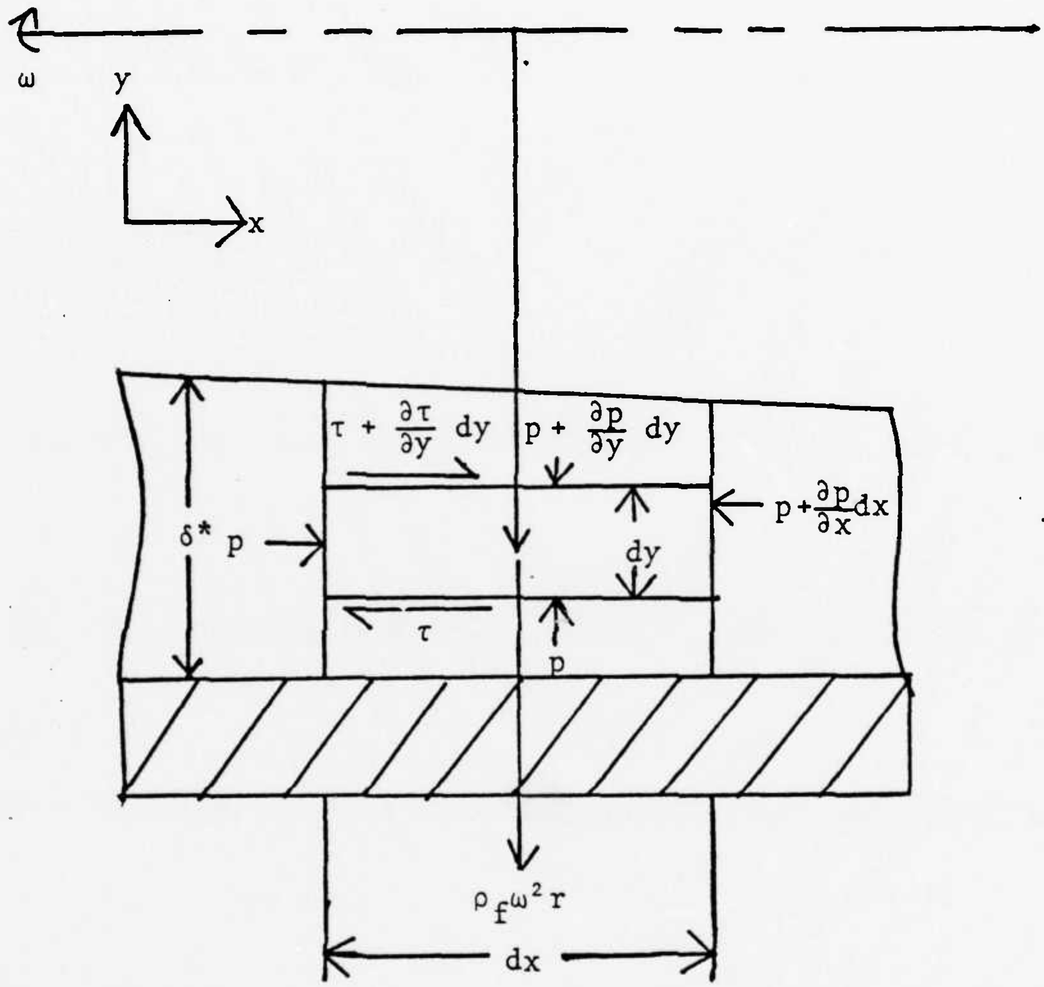


Figure 3. Cross Section of Infinitesimal Fluid Element on Cylindrical Condenser Internal Surface

4. Fluid Velocity

Integrating equation (2.2) between the limits y and δ^* for y and corresponding limits of P and P_v for pressure results in the following equation:

$$P = P_v + \rho_f \omega^2 r\ (\delta^* - y) \qquad \text{(eqn 2.3)}$$

where P_v = the pressure of the vapor (lbf/ft^2), and

δ^* = film thickness (ft)

Differentiating equation (2.3) with respect to x yields the following expression for dP/dx:

$$\frac{dP}{dx} = \frac{dP_v}{dx} + \rho_f \omega^2 r \frac{d\delta^*}{dx} \qquad \text{(eqn 2.4)}$$

Applying assumption (f), (P_v is constant, therefore, $dP_v/dx=0$) and substituting equation (2.4) into equation (2.1) yields:

$$\frac{\partial \tau}{\partial y} = \rho_f \omega^2 r \frac{d\delta^*}{dx} \qquad \text{(eqn 2.5)}$$

Integrating equation (2.5) with the corresponding limits of integration y to δ^* and τ to 0 results in the following expression for shear stress:

$$\tau = \rho_f \omega^2 r \frac{d\delta^*}{dx} [y - \delta^*] \qquad \text{(eqn 2.6)}$$

But,

$$\tau = \mu \frac{\partial u}{\partial y} \qquad \text{(eqn 2.7)}$$

where μ = fluid dynamic viscosity (lbm/ft-hr)

u = condensate velocity (ft/hr)

Substituting equation (2.7) into equation (2.6) and integrating with the corresponding limits of integration 0 to y and 0 to u yields:

$$u = \frac{\rho_f \omega^2 r}{\mu} \frac{d\delta^*}{dx} \left[\frac{y^2}{2} - y\delta^* \right] \qquad \text{(eqn 2.8)}$$

The average velocity of the condensate may be found in the following manner:

$$\bar{u} = \frac{1}{\delta^*}\int_o^{\delta^*} u dy = \frac{1}{\delta^*}\int_o^{\delta^*} \frac{\rho_f \omega^2 r}{\mu} \frac{d\delta^*}{dx} \left[\frac{y^2}{2} - y\delta^* \right] dy \qquad \text{(eqn 2.9)}$$

or

$$\bar{u} = -\frac{\rho_f \omega^2 r}{\mu} \frac{d\delta^*}{dx} \left[\frac{\delta^{*2}}{2}\right] \qquad \text{(eqn 2.10)}$$

5. Continuity Equation

The continuity equation for mass flow requires that:

$$\dot{m} = \rho_f \bar{u} A \qquad \text{(eqn 2.11)}$$

where $\dot{m}$ = condensate mass flow rate (lbm/hr)

A = cross sectional area of the fluid (ft^2)

This can also be written as

$$\dot{m} = \int_0^{\delta^*} \rho_f \bar{u}\ 2\pi r dy \qquad \text{(eqn 2.12)}$$

Substituting equation (2.10) into equation (2.12) and integrating yields:

$$\dot{m} = -\frac{2\pi\rho_f^2 \omega^2 r^2}{\mu} \frac{d\delta^*}{dx} \frac{\delta^{*3}}{3} \qquad \text{(eqn 2.13)}$$

Differentiating this equation with respect to x yields:

$$\frac{d\dot{m}}{dx} = -\frac{2\pi\rho_f^2 \omega^2 r^2}{\mu} \frac{d}{dx} \left[\frac{d\delta^*}{dx} \frac{\delta^{*3}}{3}\right] \qquad \text{(eqn 2.14)}$$

6. Energy Equation

Having applied assumption (e), if the film surface temperature is at the saturation temperature (T_{sat}) of the vapor and if the wall of the axial increment is at a given constant temperature (T_w), then the heat transfer by conduction of a fluid element of surface area dA is:

$$dq = \frac{k_f (T_{sat} - T_w)\ dA}{\delta^*} \qquad \text{(eqn 2.15)}$$

where dq = differential heat transfer rate (Btu/hr)

dA = $2\pi r dx (ft^2)$

k_f = thermal conductivity of the condensate film (Btu/hr-ft-F)

T_{sat} = saturation temperature (degrees F)

T_w = inside condenser wall temperature (degrees F)

Considering the change of phase and defining $\bar{h}_{fg}$ as the average enthalpy change of the vapor in condensing to a liquid and subcooling to the average liquid temperature of the film, then dq is also defined by:

$$dq = \bar{h}_{fg}\, d\dot{m} \qquad \text{(eqn 2.16)}$$

where h_{fg} = laten heat of vaporization (Btu/lbm)

c_p = specific heat (Btu/lbm R)

ΔT = $(T_{sat} - T_w)$

$\bar{h}_{fg} = h_{fg} + 0.35 \cdot c_p \cdot \Delta T$

Rearranging equation (2.16) and substituting this equation into equation (2.15) yields:

$$\frac{d\dot{m}}{dx} = \frac{k_f(T_{sat} - T_W)\ 2\pi r}{\delta^*} \qquad \text{(eqn 2.17)}$$

Finally coupling the energy and continuity equations result in the following differential equation:

$$\delta^* \frac{d}{dx}\left[\frac{d\delta^*}{dx}\ \delta^{*3}\right] = -\frac{3k_f(T_{sat} - T_W)\mu}{\rho_f{}^2\omega^2 r\ \bar{h}_{fg}} \qquad \text{(eqn 2.18)}$$

Equation (2.18) can be solved using the Finite Element Method to provide the film thickness profile along the axial length of a cylindrical condenser. Appendix A provides a detailed description of this solution. Once the film profile is known, a steady state two-dimensional heat conduction analysis can be performed.

7. Determination of Heat Transfer Rate

Assume that the cylindrical condenser section of the rotating heat pipe is divided axially into a number of increments. Then for any axial increment of a cylindrical condenser, the differential heat flux can be determined by the following expression.

$$dq'' = \frac{(T_{sat}^{*} - T_\infty)}{\frac{\delta}{k_f} + \frac{thick}{k_w} + \frac{1}{h_{ext}}} \qquad \text{(eqn 2.19)}$$

where T_∞ = ambient temperature (degrees F)

thick = thickness of the condenser wall (ft)

k_w = thermal conductivity of the wall material (Btu/hr-ft-F)

h_{ext} = external heat transfer coefficient (Btu/hr-ft^2-F)

Note the three terms in the denominator are the thermal resistances of the film, wall and external convection respectively.

The differential heat transfer rate for any increment can be found by the following relationships:

$$dq = dq'' \cdot 2\pi r dx \qquad \text{(eqn 2.20)}$$

or

$$dq = \frac{2\pi r(T_{sat} - T_W)\,dx}{\frac{\delta^*}{k_f} + \frac{thick}{k_w} + \frac{1}{h_{ext}}} \qquad \text{(eqn 2.21)}$$

Equation (2.21) represents the total heat transfer rate for an incremental section of width dx. To find the total heat transfer rate for the entire cylindrical condenser, the incremental heat rates must be summed over the entire length of the condenser. Therefore:

$$Q_{total} = \sum_{i=1}^{NDIV} dq \qquad \text{(eqn 2.22)}$$

where NDIV = total number of axial increments.

C. THEORY FOR A CYLINDRICAL AXIALLY FINNED CONDENSER

1. Assumptions

Referring to Figure 4, it is obvious that the analysis of a cylindrical internally finned condenser is more complicated due to the mass flow from the fins into the trough region between the fins. For this reason, in addition to the simplifying assumptions made for the smooth condenser which are listed in the previous section, the following assumptions must also be made:

a) Referring to Figure 5, the mass flow along the fin surface does not flow axially in the x-direction, but only along the surface of the fin in the z-direction into the trough. Thus, mass flow in the axial direction is only permitted in the trough region between the fins. This is a reasonable assumption in that the film thickness along the fin surface is very small in relation to the film thickness in the trough. This being the case, the hydrostatic force in the x-direction on a fin fluid element will be much less than the centrifugal force component in the z-direction on that same fluid element forcing that fluid element into the trough.

b) Just as in the axial direction, there is no pressure change along the surface of the fin in the z-direction.

c) It will be assumed that the temperature along the convective surface of the fin is at a constant value (T_{avg}). This average fin surface temperature is the arithmetic average of the fin tip temperature and the fin base surface temperature where the fin intersects with the wall of the condenser. This is a valid assumption if the fin section is divided into a sufficient number of finite elements. Purnomo's [Ref. 5] results indicate a less than one degree variation, even for very large fin half angles. This variation in temperature will have an insignificant effect on film thickness along the surface of the fin and can be neglected by using an average value.

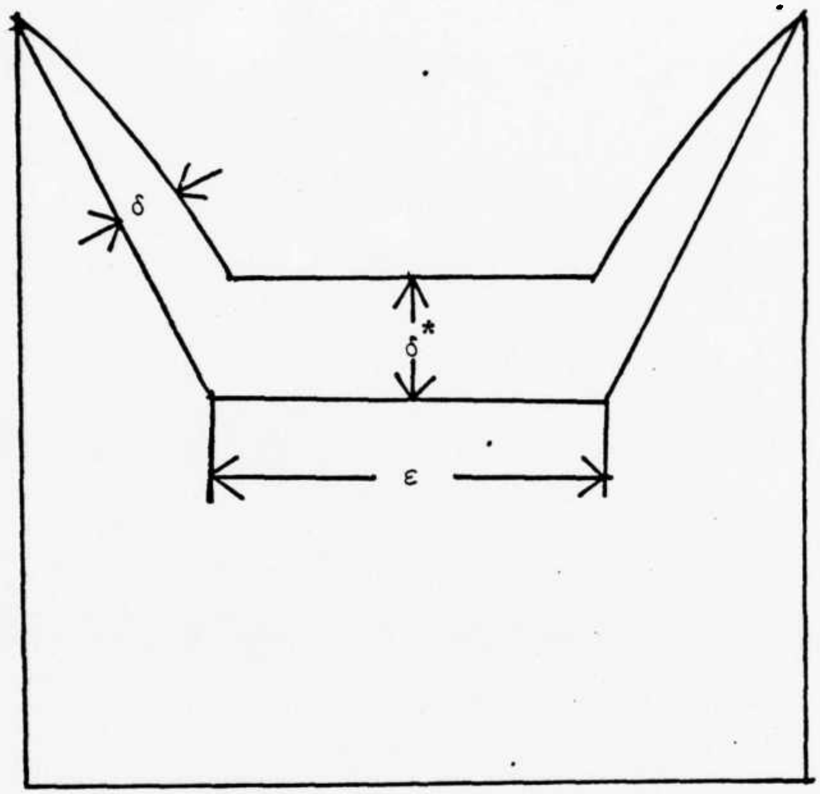


Figure 4. Trough Section of Axially Finned Condenser

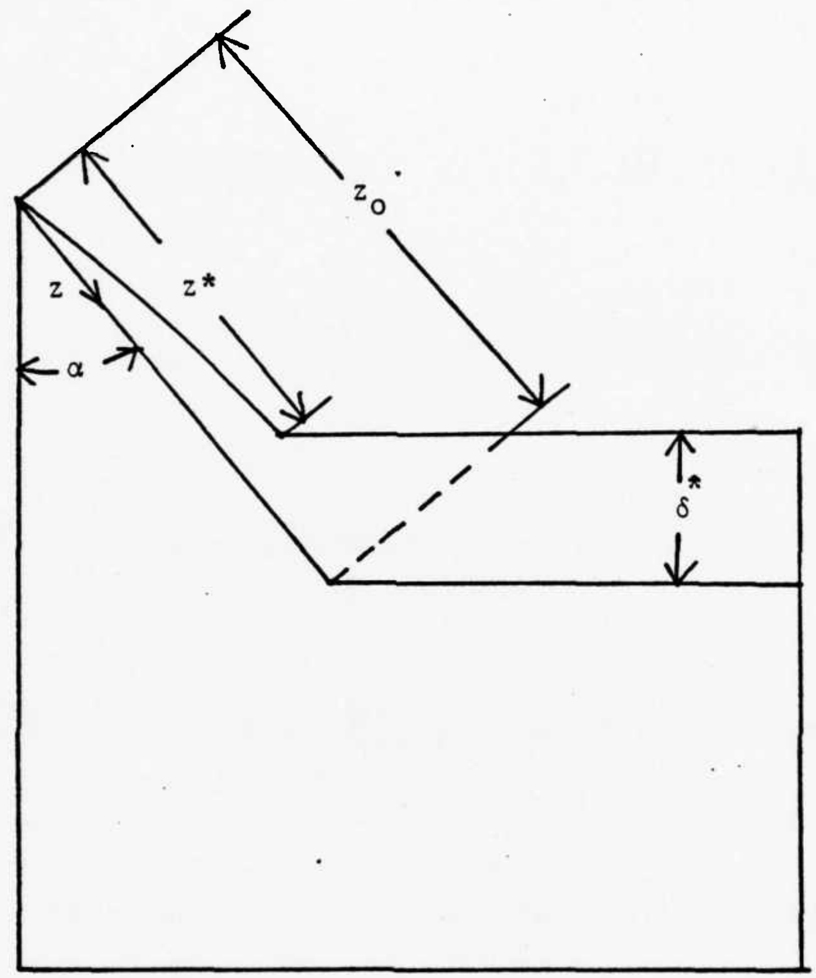


Figure 5. Axially Finned Condenser Symmetric Fin Section

2. Mass Flow in the X-Direction

As a result of assumption (a), the resulting momentum equation in both the x and y directions as well as the equations for velocity and mean velocity are identical to those developed in section B for the smooth condenser and will not be redeveloped here. Looking now at mass flow in the x-direction which is limited to flow only in the trough, the mass flow rate is given by the following expression:

$$\dot{m}_{total} = -\frac{\rho_f{}^2\omega^2 r}{3\mu}\frac{d\delta^*}{dx}\,\delta^{*2}(\varepsilon\delta^* + \delta^{*2}\tan\alpha) \qquad \text{(eqn 2.23)}$$

where α = fin half angle (radians)

ε = width of the trough (ft)

Note that the quantity in parentheses is the cross sectional area of the film condensate in the trough (See Figure 4). Taking the derivative of equation (2.23) with respect to x yields the rate of change of mass flow in the trough for a given axial increment.

$$\frac{d\dot{m}_{total}}{dx} = \frac{\rho_f\omega^2 r}{3\mu}\frac{d}{dx}\left[\frac{d\delta^*}{dx}(\varepsilon\delta^{*3} + \delta^{*4}\tan\alpha)\right] \qquad \text{(eqn 2.24)}$$

Equation (2.24) represents the rate of change of the total mass flow rate with respect to x in the x-direction. This equation must be coupled with the energy equations for the fin and trough to develop a representation of the film profile in the trough.

3. Mass Flow in the Z-Direction

Examining an infinitesimal fluid element on the surface of a fin for any axial increment of width Δx, as shown in Figure 6, the momentum equation in the z-direction becomes:

$$\frac{\partial \tau_a}{\partial y} = \frac{\partial P}{\partial z} - \rho_f \omega^2 r\cos\alpha \qquad \text{(eqn 2.25)}$$

where τ_z = shear stress in the z-direction (lbf/ft²)

z = co-ordinate measuring distance along the surface of the fin (ft)

Neglecting dP/dz based on assumption (b), and integrating equation (2.25) from τ_z to 0 and y to δ yields:

$$\tau_z = \mu \frac{\partial w}{\partial y} = \rho_f \omega^2 r\cos\alpha(\delta - y) \qquad \text{(eqn 2.26)}$$

where w = fluid velocity in the z-direction (ft/hr)

δ = fin film thickness along the surface of the fin (ft)

Note, δ , the fin film thickness should not be confused with δ^*, the film thickness in the trough. Integrating equation (2.26) from 0 to w and 0 to y provides the following expression for fluid velocity:

$$w = \frac{\rho_f \delta^2 r\cos\alpha}{\mu} \left(\delta(z)y - \frac{y^2}{2}\right) \qquad \text{(eqn 2.27)}$$

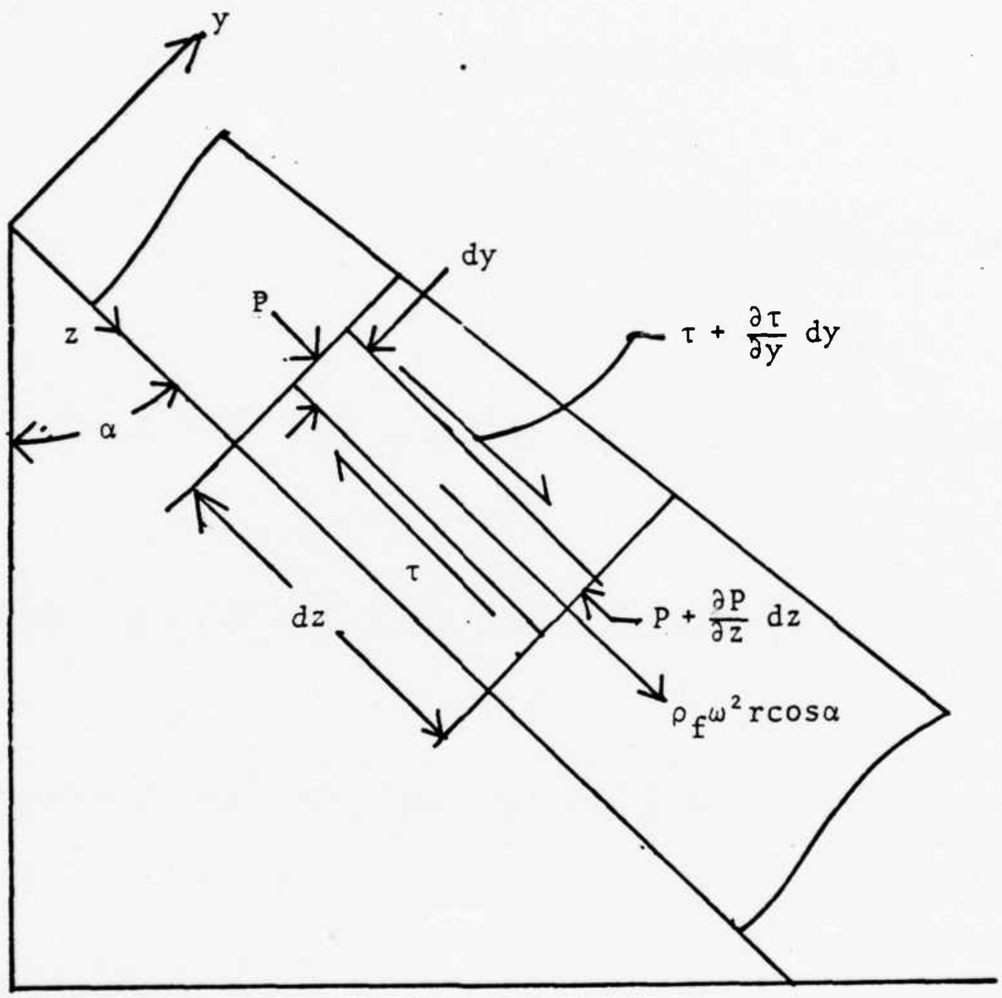


Figure 6. Cross Section of Infinitesimal Fluid Element on Fin Surface of Axially Finned Condenser

This relationship may be used to find the average fin fluid velocity $\overline{w}$:

$$\overline{w} = \frac{1}{\delta}\int_{o}^{\delta} w dy = \frac{\rho_f \omega^2 r\cos\alpha \ \delta(z)^2}{3\mu} \qquad \text{(eqn 2.28)}$$

The mass flow rate in the z-direction along the surface of the fin for a given axial increment is given by:

$$\dot{m}_{fin} = \rho_f \overline{w} dA \qquad \text{(eqn (2.29)}$$

where dA = the cross sectional area of the fluid flowing along the fin surface (ft^2)

Substituting equation (2.28) into equation (2.29) yields:

$$\dot{m}_{fin} = \frac{\rho_f^{\,2} \omega^2 r\cos\alpha \ \delta^3(z) dx}{3\mu} \qquad \text{(eqn 2.30)}$$

This equation is identical to equation (15) of Schafer's [Ref. 3] analysis if the condenser cone half angle (Ø) is set equal to 0 which is the case for a cylindrical condenser.

4. Energy Equation for the Trough Condensate

An energy balance on an infinitesimal fluid element in the trough of an axial increment of width dx with surface area $\varepsilon \cdot dx$ yields the following expression for heat transfer by conduction:

$$dq_{trough} = \frac{k_f(T_{sat} - T_w)\varepsilon dx}{\delta^*} \qquad \text{(eqn 2.31)}$$

Note also, that the trough heat transfer rate is given by:

$$dq_{trough} = \bar{h}_{fg}\, d\dot{m}_{trough} \qquad \text{(eqn 2.32)}$$

Combining equations (2.31) and (2.32) and dividing by dx results in the following:

$$\frac{d\dot{m}_{trough}}{dx} = \frac{k_f(T_{sat} - T_w)\varepsilon}{\bar{h}_{fg}\,\delta^*} \qquad \text{(eqn 2.33)}$$

Equation (2.33) is an expression for incremental change in mass flow rate with respect to x due to condensation in the trough region.

5. Energy Equation for the Fin Condensate

An energy balance on a differential element of surface area dx·dz yields the following relationship for differential heat into the fin:

$$dq_{fin} = h_{fg}\, d\dot{m}_{fin} = \frac{k_f(T_{sat} - T_{fin}(z))dxdz}{\delta(z)} \qquad \text{(eqn 2.34)}$$

where $T_{fin}(z)$ = fin surface temperature at some position z along the surface of the fin (degrees F)

Since the fin condensate mass is assumed to flow only in the z-direction, equation (2.30) is differentiated with respect to z and substituted into equation (2.34). After substitution and rearrangement, the following equation results:

$$\delta(z)^3 d\delta(z) = \frac{k_f(T_{sat} - T_{fin}(z))dz}{\rho_f{}^2\omega^2 r\ h_{fg}\ \cos\alpha} \qquad \text{(eqn 2.35)}$$

Applying assumption (c) i.e., $T_{fin}(z)$ equals T_{avg} for all z and integrating equation (2.35) from 0 to δ and 0 to z yields the following relationship for fin film thickness δ (z):

$$\delta(z) = \left[\frac{4\ k_f(T_{sat} - T_{avg})\mu z}{\rho_f{}^2\omega^2 r\ h_{fg}\ \cos\alpha}\right]^{1/4} \qquad \text{(eqn 2.36)}$$

where T_{avg} = average fin surface temperature (degrees F). Substituting equation (2.36) into equation (2.30) and solving for rate of change of mass flow rate of the fin with respect to x for an increment of width dx yields:

$$\frac{d\dot{m}_{fin}}{dx} = \frac{2\rho_f{}^2\omega^2 r\cos\alpha}{3\mu}\left[\frac{4\ k_f(T_{sat} - T_{avg})\mu z^*}{\rho_f{}^2\omega^2 r\ h_{fg}\ \cos\alpha}\right]^{1/4} \qquad \text{(eqn 2.37)}$$

where $z^* = z - \delta^*/\cos(\alpha)$. Note, z^* is the distance along the surface of the fin from the fin tip to the trough film thickness (δ^*). Note also, that the right hand side of equation (2.37) is multiplied by two; this accounts for mass flow from the fins on both sides of the trough.

6. Continuity Equation

For any axial increment of length dx, continuity dictates that:

$$\frac{d\dot{m}_{total}}{dx} = \frac{d\dot{m}_{fin}}{dx} + \frac{d\dot{m}_{trough}}{dx} \qquad \text{(eqn 2.38)}$$

Substituting equations (2.24), (2.33) and (2.37) into equation (2.38) and rearranging yields:

$$\delta^* \frac{d}{dx}\left[\frac{d\delta^*}{dx}\left(\varepsilon\delta^{*3} + \delta^{*4}\tan\alpha\right)\right] = -\frac{3\,k_f(T_{sat} - T_w)\mu\varepsilon}{\rho_f^{\,2}\omega^2 r\,\bar{h}_{fg}}$$

$$- 2\delta^*\cos\alpha\left[\frac{4\,k_f(T_{sat} - T_{avg})\mu z^*}{\rho_f^{\,2}\omega^2 r\,h_{fg}\cos\alpha}\right]^{3/4} \qquad \text{(eqn 2.39)}$$

Equation (2.39) can be solved using the Finite Element Method formulation provided in Appendix A. The solution of this equation provides the film thickness profile along the axial length of a cylindrical finned condenser.

7. <u>Determination of the Heat Transfer Rate</u>

Once the film profile has been determined within the trough, the local convective heat transfer coefficient can be found for the trough using the following relationship:

$$h(x)_{trough} = \frac{k_f}{\delta^*(x)} \qquad \text{(eqn 2.40)}$$

In a similar manner, the local heat transfer coefficient along the surface of the fin can be found by:

$$h(z)_{fin} = \frac{k_f}{\delta(z)} \qquad \text{(eqn 2.41)}$$

The differential heat transfer rate for any fin section, as shown in Figure 4, of axial incremental length dx is:

a) for the trough:

$$dq_{trough} = \frac{(T_{sat} - T_w)\varepsilon dx}{h(x)_{trough}} \qquad \text{(eqn 2.42)}$$

where $\varepsilon \cdot dx$ is the surface area of the trough, and

b) for the fin surface:

$$dq_{fin} = 2\int_0^{z_o} \frac{(T_{sat} - T_{avg})dxdz}{h(z)_{fin}} \qquad \text{(eqn 2.43)}$$

where z_o is the surface length of the fin.

The total differential heat transfer rate per axial increment is found by summing equation (2.42) and (2.43) for the total number of fins. That is:

$$dq_{total} = \sum_{1}^{NFIN} (dq_{fin} + dq_{trough}) \qquad \text{(eqn 2.44)}$$

where NFIN is the total number of axial fins.

In a similar manner, the total heat transfer rate for the entire finned condenser can be found by the following relationship:

$$Q_{total} = \sum_{1}^{NDIV} dq_{total} \qquad \text{(eqn 2.45)}$$

where NDIV is the total number of axial increments.

III. COMPUTER CODE DESCRIPTION

A. GENERAL DESCRIPTION OF CODE

The computer code consists of a main body and eight subroutines. Basically, the code which is provided in Appendix C is a modification of Purnomo's [Ref. 5] code. The function of each subroutine used in the code is as follows:

a) "CORRES" established the correspondence between the local and global nodal points used in the Finite Element Method solution for the two-dimensional steady state heat conduction problem. In so doing, "CORRES" also numbers all elements and nodal points in the finite element model and assigns local nodal points to each of the elements. In addition "CORRES" also defines major element numbers used in other subroutines as control parameters.

b) "COORD" defines the x and y coordinates for all nodal points in the finite element heat conduction problem model.

c) "DLSTAR" determines the film thickness (δ^*) on the surface of a smooth condenser or in the trough in a finned condenser.

d) "HTCOEF" determines the heat transfer coefficient for all convective surface elements.

e) "FORMAF" formulates the Finite Element Method equations for the two-dimensional steady state heat conduction problem.

f) "BANDEC" is an equation solver for a symmetric matrix which has been transformed into banded form. "BANDEC" will return the solution to the two-dimensional heat conduction problem.

g) "HTCALC" determines the elemental, incremental and total heat transfer rates.

h) "DELCRV" determines the condensate film profile in a cylindrical condenser.

Two additional Naval Postgraduate School computer library routines are also used in the code:

a) "DPOLRT" is a nonIMSL double precision library routine that determines the roots of a real polynomial. This routine is called by "DLSTAR" to determine the film thickness for the succeeding increment in the analysis of a tapered condenser.

b) "LEQT2F" is an IMSL double precision libary routine that solves a set of simultaneous linear equations. This routine is called by "DELCRV" to solve the Finite Element Method equations for the cylindrical condenser film profile problem. The resulting film profile is then used in the heat conduction analysis.

In order to use the computer code to analyze heat transfer in a rotating heat pipe, nine data cards are required. A user's guide describing these data cards and required input is provided in Appendix B. The input data, describing the geometric configuration of the rotating heat pipe as well as the operating parameters determines which solution technique is utilized in the analysis. The solution technique for each of the four condenser geometries, i.e., tapered-smooth, tapered-axially finned, cylindrical-smooth and cylindrical-axially finned is different. In all cases however, the Finite Element Method is used to solve the two-dimensional steady state heat conduction problem. This solution is the one developed by Purnomo [Ref. 5] and has not been modified. Details of the development of this solution are described in detail in Purnomo's thesis [Ref. 5] and will not be repeated here. This being the case, each of the four solution techniques will now be discussed in detail.

B. INTERNALLY FINNED TAPERED CONDENSER SOLUTION

The complete development of this solution technique is provided in Purnomo's [Ref. 5] thesis and will not be redeveloped. When an equation is required for clarity, the equation in final form will be provided. Where there is a modification to Purnomo's [Ref. 5] code, this modification will be noted.

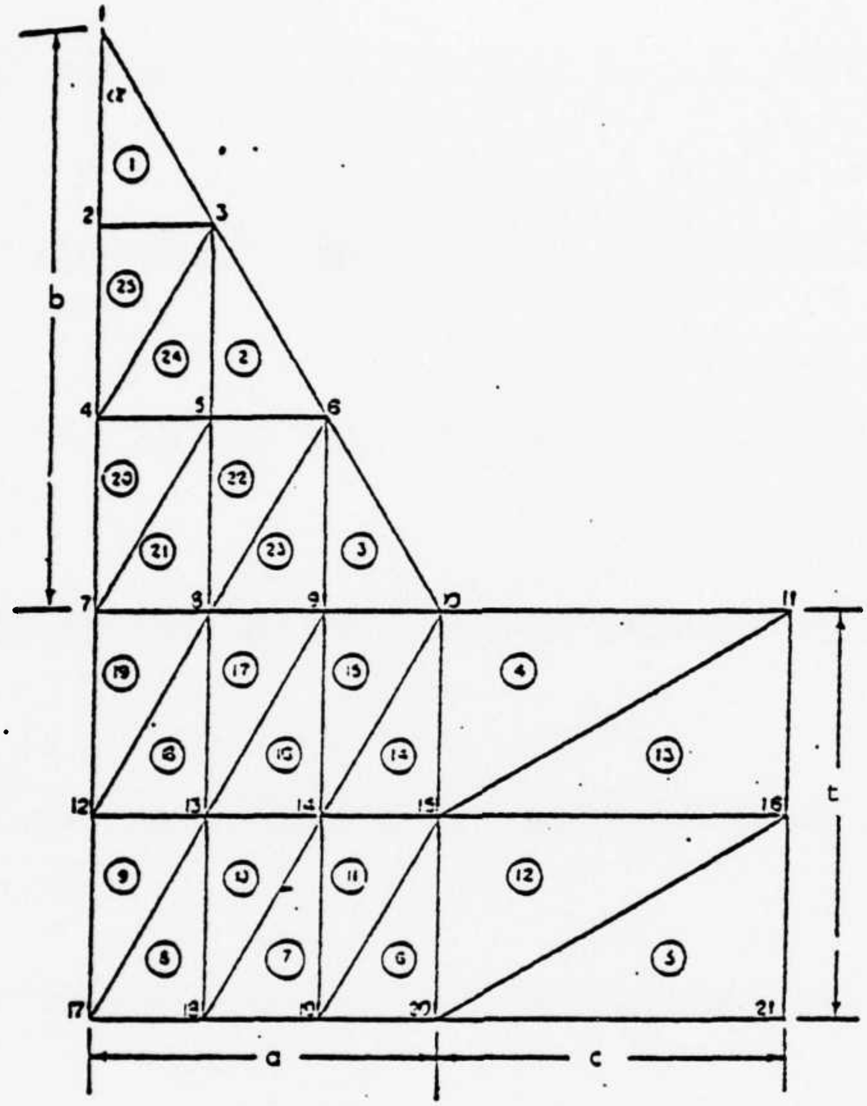
The condenser of the internally finned-tapered condenser is divided into NDIV axial increments. These axial increments are then subdivided circumferentially into ZFIN number of subincrements where ZFIN is the total number of fins.

These subincrements are then divided in half to form the basic symmetric unit, one-half of a fin-trough section as shown in Figure 1. This unit section is then divided into a number of linear triangular elements. The number of elements depend on the input parameters. The only limitation is that the number of system nodal points must not exceed 100; otherwise, certain variables, e.g., x and y, would exceed allotted storage values. Figure 7 shows a unit section subdivided into 25 elements. After this unit is subdivided, each nodal point is assigned an x and y coordinated based on the geometric input parameters.

To start the iteration, two initial values are required: 1) an initial temperature for the nodal points along the internal convective boundary, and 2) an initial trough film thickness (δ^*). The initial temperature is provided as an input parameter and the initial trough film thickness at the first increment is provided by a relationship taken from an analysis by Sparrow and Gregg for condensation on a rotating disk [Ref. 12].

Once these values are known, the heat transfer coefficient for the internal convective surface elements are found using the following relationships:

$$h(z)_{fin} = \frac{k_f}{\delta(z)} = \left[\frac{k_f \rho_f{}^2 \omega^2 (r + x\sin\emptyset) h_{fg} \cos\emptyset \cos\alpha}{4\mu(-AA \cdot z^3/3 - BB \cdot z^2/2 + (T_{sat} - T_1) z}\right]^{1/4} \quad \text{(eqn 3.1)}$$



$a = 0.02953$ inches

$b = 0.05906$ inches

$c = 0.04412$ inches

$t = 0.05118$ inches

Figure 7. Axially Finned Condenser Symmetric Section Subdivided into 25 Linear Triangular Finite Elements

where x = distance from condenser end (x=0.0) to midpoint of increment (ft)

T_1 = fin apex temperature (degrees F)

AA, BB = constants in the parabolic temperature determined by a Langrangian fit.

For the trough surface elements:

$$h(x)_{trough} = \frac{k_f}{\delta^*(x)} \qquad \text{(eqn 3.2)}$$

These heat transfer coefficients, along with the thermal conductivity and x and y coordinates of the nodal points are used to form the Finite Element Method equations for the two-dimensional heat conduction problem. The equations are then solved to yield a temperature distribution in the symmetric section.

The above iteration is repeated, where, now the solution temperature distribution from the previous iteration is used to calculate the heat transfer coefficients along the convective surface fin elements as well as a new δ^* using Sparrow's and Gregg's relationship [Ref. 12]. This new δ^*, in turn, is used to determine the heat transfer coefficients of the trough elements.

Again, the Finite Element Method subroutines will yield a temperature distribution for the symmetric section. At this point, the nodal point temperatures are checked for convergence using the following relationship:

$$\text{Max} \left| \frac{T_{i,j} - T_{i,j-1}}{T_{i,j}} \right| \leq \text{CRIT} \quad i = 1,2,\ldots \text{NSNP} \qquad \text{(eqn 3.3)}$$

where NSNP = number of system nodal points

j = present iteration

j-1 = previous iteration

CRIT = convergence criterion

If this convergence test is successful for all nodal points, the increment is considered solved. If any one nodal point fails, the iterative process is repeated until convergence is met. The convergence test in equation (3.3) is different than the one used by Purnomo [Ref. 5] in his thesis. Purnomo compared incremental heat transfer rates per unit of condenser length, Q_i, rather than temperature as is done in the modified code.

If convergence is met, the heat transfer rate is determined. From this heat transfer rate, the incremental mass flow rate is determined by the following equation:

$$\dot{m}_{total} = \frac{2Q_i \Delta x}{h_{fg}} \qquad \text{(eqn 3.4)}$$

where Q_i = heat transfer rate per unit length (Btu/hr-ft)

Δx = incremental width (ft)

Using this value of incremental mass flow rate determined by equation (3.4), the following equation is used to calculate the subsequent interval's trough condensate film thickness (δ^*) with a polynomial rootfinder subroutine:

$$\dot{m}_{total} = \frac{\rho_f^{\,2}\omega^2(r+x\sin\emptyset)\,\delta^{*2}(x)\sin\emptyset(\delta^*(x)\varepsilon+\delta^{*2}(x)\tan\alpha)}{3\mu} \qquad \text{(eqn 3.5)}$$

where ε = trough width (ft)

This resulting value of $\delta^*(x)$ is then defined as the trough film thickness for the next increment. In addition, the solution temperature distribution from the previous iteration is used as the starting temperature distribution for the next increment.

This iterative process at each increment is repeated until convergence is met, and, is continued at each increment until the entire length of the condenser has been transversed. Incremental heat rates are then summed to yield the total heat transfer rate. That is:

$$Q_{total} = 2*ZFIN*\sum_{i=1}^{NDIV} Q_i \cdot \Delta x \qquad \text{(eqn 3.6)}$$

where ZFIN is the total number of axial fins.
Once the total heat transfer rate has been determined, the problem is solved and pertinent data is provided as output.

C. SMOOTH TAPERED CONDENSER SOLUTION

The heat pipe condenser is divided into NDIV number of axial increments as in the finned-tapered condenser solution. These axial increments are then subdivided into 360 segments of equal length; these segments are the basic symmetric unit.

This unit section, is divided into a number of linear triangular elements with the same limitation as before; the number of system nodal points must not exceed 100. The system nodal points are then assigned x and y coordinates based on the input geometric parameters.

To start the iterative process, as in the finned-tapered case, the initial value of temperature which is an input parameter is used to solve for the initial value of fin thickness (δ*) based on the Sparrow and Gregg analysis [Ref. 12].

Once this initial value of δ* is known, the heat transfer coefficients for the internal convective elements can be determined using equation (3.2). These heat transfer coefficients are used in the Finite Element Method equations. The equations are solved yielding a temperature distribution. The iteration is repeated until convergence is met, just as in the finned-tapered case.

When convergence is met, that is equation (3.3) has been satisfied, a new film thickness $\delta^*(x)$ can be found by one of the following equations for $\delta^*(x)$:

$$Sh_x\left[\frac{\delta^*}{x}\right]^4 - \frac{1}{3}\, Dr_x \left[\frac{\delta^*}{x}\right]^3 - \frac{1}{4}\, Rev_x \left[\frac{\delta^*}{x}\right]^2 - 1 = 0 \qquad \text{(eqn 3.7)}$$

or

$$Sh_x \left[\frac{\delta^*}{x}\right]^4 - 1 = 0 \qquad \text{(eqn 3.8)}$$

or

$$\delta^*(x) = \frac{3k_f \mu (T_{sat} - T_w)}{2\rho_f^2 \omega^2 r \sin^2 \emptyset h_{fg}} \left[1 - \left\{\frac{r}{(r + x\sin\emptyset)}\right\}^{8/3}\right]^{1/4} \qquad \text{(eqn 3.9)}$$

where
$$Sh = \frac{\rho_f^2 (\omega^2 r - g)\sin\emptyset \bar{h}_{fg} x^3}{4 \mu k_f (T_{sat} - T_w)}$$

$$Dr = \frac{\rho_f \tau_v h_{fg} x^2 \cos\emptyset}{\mu k_f (T_{sat} - T_w)}$$

$$Rev = \frac{\rho_f v \cos\emptyset}{\mu}$$

g = acceleration due to gravity (ft/hr^2)

v = vapor velocity (ft/hr)

T_w = local wall temperature (degrees F0

τ_v = shear stress vapor-liquid interface (lbf/ft^2)

Equation (3.7) defines the film thickness distribution for a smooth tapered rotating heat pipe derived by Daniels and Al-Jumaily [Ref. 13]. This equation takes into account the drag effects of counter-flowing vapor. Equation (3.8) is a modification of Equation (3.7) neglecting the drag losses. Equation (3.9) was developed by Ballback [Ref. 1]. This equation also neglects drag.

Depending on a particular control parameter which is part of the input data (See Appendix B) one of these equations is used to solve for the film thickness ($\delta^*(x)$) for the next increment. In addition, the solution temperature distribution from the previous iteration is used as the starting temperature distribution for the next increment.

This iterative process at each increment is repeated until convergence is met, and is continued at each increment until the entire length of the condenser has been transversed, just as in the finned-tapered case. Total heat transfer rates are then determined by summing the incremental heat transfer rates for the entire length of the condenser by the following relationship:

$$Q_{total} = 360 * \sum_{i=1}^{NDIV} Q_i \Delta x \qquad \text{(eqn 3.10)}$$

D. SMOOTH CYLINDRICAL CONDENSER SOLUTION

As in the smooth-tapered case, the condenser is first divided axially, then it is divided circumferentially into 360 segments of equal length. These segments are the basic symmetric unit section to be considered. Again the symmetric unit is subdivided into linear triangular elements and x and y coordinates are assigned to the system nodal points.

To begin the iteration, an initial temperature estimate which is an input parameter is assigned to the convective surface nodal points. Using this initial temperature estimate, the maximum film thickness, δ^*_{max} which is located at $x = 0$, is determined. This maximum film thickness value is then used as one of the boundary conditions in the solution of equation (2.18) using the Finite Element Method. Equation (2.18) is repeated here for reference.

$$\delta^* \frac{d}{dx} [\frac{d}{dx}{}^* \quad \delta^{*3}] = -\frac{3k_f(T_{sat} - T_w)\mu}{\rho_f{}^2\omega^2 r\ \bar{h}_{fg}} \qquad \text{(eqn 3.11)}$$

The finite element solution of Equation (3.11) provides the film thickness profile ($\delta^*(x)$) at the midpoint of each increment along the length of the condenser, but only the first increment value is applied at this point of the analysis. Once $\delta^*(x)$ is known at the first increment the heat transfer coefficients for the internal convective surface elements can be determined using equation (3.2). The steady state heat conduction problem is then solved and a temperature distribution for the unit section results. This process is now repeated. A new maximum film thickness, film profile and thus $\delta^*(x)$ at the first increment is found based on the solution temperature distribution from the first iteration. With this new value for $\delta^*(x)$, the heat conduction problem is again solved for a new temperature distribution. This iterative process, involving solution of the film profile with each iteration is continued until temperature convergence is met at the first increment. When convergence is met, the film profile determined on the iteration in which convergence was met is used to provide the values of $\delta^*(x)$ for the remaining increments along the length of the condenser.

Using this predetermined value of $\delta^*(x)$, the iterative process is continued at each increment until temperature convergence is reached. As convergence is reached at each increment, the internal wall temperature, which is the same for all internal wall nodal points of the section, is stored for future application.

When convergence is reached at the final increment, equation (3.11) is once again solved for the film profile. It should be noted, however, that the right hand side of equation (3.11) is temperature dependent. It should also be noted that the temperature varies axially along the length of the condenser. In order to account for this temperature dependence and temperature variation, the right side of equation (3.11) is now determined for each increment using the wall temperatures that were stored at each increment. The finite element solution of the film profile will now account for the temperature variation along the length of the condenser.

This final film profile provides the value of $\delta^*(x)$ for each increment. The iterative process of solving for heat transfer coefficient, temperature distribution, and temperature convergence is continued for each increment until the final increment is reached.

Equation (2.13) provides a relationship for total mass flow rate as a function of position. The total mass flow rate at the overfall into the evaporator is given by:

$$\dot{m} = -\frac{2\pi\rho_f^2\omega^2 r}{\mu} \cdot \frac{d\delta^*(L)}{dx} \cdot \frac{\delta^{*3}(L)}{3} \qquad \text{(eqn 3.12)}$$

where $\delta^*(L)$ and $d\delta^*(L)/dx$ are the values for the film thickness and rate of change of film thickness with respect to x at the evaporator end of the condenser. Another relationship for mass flow rate based on the steady state heat conduction solution is given by:

$$\dot{m} = 360^* \sum_{i=1}^{NDIV} \frac{Q_i \Delta x}{\bar{h}_{fg}} \qquad \text{(eqn 3.13)}$$

If, in fact, the solution of equations (3.12) and (3.13) are equal, then the mass flow rate of the condensate returning to the evaporator is equal to the mass flow rate of the vapor being condensed on the surface of the condenser. Or, to put it another way, continuity is satisfied.

It should be noted that the film profile maintains the same basic shape, that is, $\delta^*(x)$ at x=0 is always equal to δ^*_{max} and decreases to a specific minimum value at x=L. This being the case, if the maximum film thickness is varied, the film thicknexx profile will vary in the same manner. For example, if the maximum film thickness is increased, the entire profile will also increase. This will result in an increased internal thermal resistance and thus lower heat transfer rate. As a result of the lower heat transfer rate, the mass flow rate as determined by equation (3.13) will be less. At the same time, however, the greater film profile will result in a greater value of $\delta^*(x)$ at the overfall, $x = L$. Yet, since the profile maintained the same basic shape, the derivative at the overfall remains relatively constant. Thus the mass flow rate as determined by equation (3.12) will increase. A decrease in the maximum film thickness will result in an opposite effect to the mass flow rates determined by equations (3.12) and (3.13).

This being the case, the mass flow rates, as determined by equations (3.12) and (3.13) are now compared to determine if the film profile is in fact the solution profile to the problem. If continuity is not satisfied, the maximum film thickness is varied and the entire iterative process is restarted. This process is continued until the film profile mass flow rate, equation (3.12) converges towards the heat transfer mass flow rate, equation (3.13). When the absolute difference between these mass flow rates is less than a specific value, the resulting heat transfer rate is considered the solution to the problem.

E. FINNED CYLINDRICAL CONDENSED SOLUTION

As in the finned-tapered case, the condenser is divided axially and circumferentially into the basic symmetric section as shown in Figure 1. This unit is then subdivided into linear triangular elements and x and y coordinates are assigned to each nodal point.

To begin the iterative process, as before, an initial temperature estimate is assigned to each nodal point along the internal convective surface. An initial trough film profile is determined, using equation (3.11) and this initial temperature estimate. In this case, however, the maximum film thickness (δ^*_{max}) is not calculated but is an input parameter.

Once $\delta^*(x)$ is known at the first increment, the internal heat transfer coefficients are determined, using equations (3.1) and (3.2). These values are then used in the Finite Element Method solution of the steady state heat conduction problem. A temperature distribution is determined and the iteration is repeated until temperature convergence is met. Note that a new film profile is determined for each iteration.

As in the cylindrical-smooth condenser case, once convergence is met at the first increment, the film profile that was determined for the iteration prior to convergence at the first increment is then used to provide the film thickness $\delta^*(x)$ for the remaining increments.

At the final increment, a new film profile for a finned cylindrical condenser is then determined by solving the following equation developed in Chapter II:

$$\delta^* \frac{d}{dx}\left[\frac{d\delta^*}{dx}\left(\delta^{*3}\varepsilon + \delta^{*4}\tan\alpha\right)\right] = -\frac{3k_f(T_{sat} - T_W)\mu\varepsilon}{\rho_f^2\omega^2 r\,\bar{h}_{fg}}$$

$$-\,2\delta^*\cos\alpha\left[\frac{4k_f(T_{sat} - T_{avg})\mu z^*}{\rho_f^2\omega^2 r\, h_{fg}\cos\alpha}\right]^{3/4} \qquad \text{(eqn 3.14)}$$

A description of the solution of this differential equation using the Finite Element Method is provided in Appenxix A.

The iterative process of finding the solution temperature distribution for all increments is then repeated until the length of the condenser has been transversed.

At this point, the total mass flow rate of the condenser returning to the evaporator is determined by the following relationship:

$$\dot{m} = -\frac{\rho_f^{2}\omega^{2} r}{3\mu} \quad \frac{d\delta^*}{dx} (\varepsilon\delta^{*3}+\delta^{*4}\tan\alpha) * ZFIN \qquad \text{(eqn 3.15)}$$

where ZFIN is the number of axial fins.
This mass flow rate is compared to the mass flow rate given by the following equation:

$$\dot{m} = ZFIN*2*\sum_{i=1}^{NDIV} \frac{Q_i \Delta x}{h_{fg}} \qquad \text{(eqn 3.16)}$$

Just as in the smooth-cylindrical condenser case, if the absolute difference between the two mass flow rates is less than a mass flow convergence criterion, the problem is considered solved. If not, δ^*_{max} is varied and the entire iterative process is started again. As in the smooth-cylindrical condenser, varying δ^*_{max} will have the same effect on the film profile and the heat transfer rate. Since the temperature distribution along the condenser has been solved once and closely approximates the final solution to the problem, on successive iterations, equation (3.14) may be used to solve for the film profile rather than equation (3.11). Equation (3.11) was used on the first iteration because the finite element solution converges more quickly than equation (3.14) when an estimated temperature is used.

A word of caution is required. The solution of equation (3.14) is highly sensitive to the value of δ^*_{max}. If the initial value of δ^*_{max} is inconsistent with the actual solution, e.g. too small, the Finite Element Method solution of the film profile will not converge. If this is in fact the case, the problem will be automatically terminated. A new value of δ^*_{max} should then be chosen and the problem restarted.

One additional topic which should be addressed is the rectangular fin solution process. The rectangular fin profile is a slight modification of the finned-tapered or finned-cylindrical condenser solution. The only variation is that the top elements of the rectangular fin are assigned heat transfer coefficients of 0.0. Thus, the tip of the rectangular fin is considered adiabatic and no heat is transferred through this top face. Other than this modification, the solution techniques are the same as for the finned cases addressed above.

IV. RESULTS AND DISCUSSION

Prior to the heat transfer analysis of the various condenser configurations, it was necessary to verify the finite element solution of the film profile. The development of this solution is discussed in Appendix A.

Leppert and Nimmo [Refs. 8 and 9] had developed an analytical solution for film condensation on a horizontal plate at a constant surface temperature. Their analysis and resulting differential equation is identical to the development in Chapter II for a smooth cylindrical condenser if the acceleration due to gravity is replaced by a radial acceleration term. This being the case, this modification was made and the analytical solution was used as a reference for comparison.

For the test runs of the finite element solution, a constant surface temperature was assumed. In addition, a value for the maximum film thickness (δ^*_{max}) and minimum film thickness (δ^*_{min}) at the overfall were required. The value for δ^*_{max} was determined based on a relationship developed by Leppert and Nimmo [Ref. 8]. The value for δ^*_{min} was arbitrarily chosen.

For identical geometry, surface temperature and maximum and minimum film thicknesses, the results of both analyses were identical. In order to develop confidence in the finite element solution, δ^*_{min} was varied from $0.10 \cdot \delta^*_{max}$ to $0.97 \cdot \delta^*_{max}$. The resulting profiles agreed at all locations along the length of the condenser. However, when the temperature was permitted to vary along the length of the condenser in the finite element solution and a corresponding average temperature was used in the analytical solution of Leppert and Nimmo [Refs. 8 and 9], the profiles were no longer in agreement. This was to be expected, particularly in the case where there was a sixteen degree Fahrenheit variation along the length of the condenser. This substantial temperature variation resulted in a significant variation of fluid properties which would account for a difference in film profile.

Due to the agreement between the finite element solution and the analytical solution of Leppert and Nimmo [Refs. 8 and 9] for a constant surface temperature, it was decided that the finite element solution does provide a satisfactory representation of the film profile. This being the case, the finite element solution was then incorporated into the code to provide the film profile for the cylindrical condenser.

Once the finite element solution of the film profile was verified, the heat transfer analysis could be accomplished. The analysis considered both copper and stainless steel condensers with the following four geometries: a) tapered-smooth, b) tapered-axially finned, c) cylindrical-smooth and d) cylindrical-axially finned. Table I lists the geometric parameters held constant for all analyses. In all cases, the working fluid was water.

TABLE I

Condenser Geometric Parameters
Held Constant During All Analyses

condenser length	=	8.500	inches
minimum radius	=	0.51575	inches
wall thickness	=	0.05118	inches

In addition, the following geometric parameters were also utilized when required. This requirement was based on the condenser geometry being considered, i.e., tapered-axially finned.

TABLE II

Condenser Geometric Parameters
Applied as Required

height of fin	=	0.05906	inches
fin half angle	=	26.565	degrees
condenser cone half angle	=	1.00	degrees

In the analysis, the heat transfer rate was determined for the four different geometries listed above for both copper and stainless steel. The ambient temperature was set at $60.0^{O}F$ and the heat transfer rate was determined for each possible combination of the operating parameters given in Table III.

TABLE III

Operating Parameter Matrix

Rotational Speed (RPM)	Heat Transfer Coefficient (btu/hr-ft^2-F)	Saturation Temperature (degree F)
700.0	100.0	90.0
1400.0	500.0	120.0
2800.0	1000.0	150.0
		180.0

Thus, for each condenser geometry, there was a total of 72 analyses, 36 for the case of the condenser with a copper wall and 36 for the case of the condenser with a stainless steel wall.

The first condenser geometry considered was a smooth condenser. Figures 8-13 compare the heat transfer rates of smooth cylindrical condensers with those of smooth tapered condenser. In particular, Figures 8,9, and 10 indicate the results of the analyses of smooth copper condensers at rotational speeds of 700, 1400 and 2800 revolutions per minute(RPM) respectively. Figures 11, 12, and 13 are for smooth stainless steel condensers at 700, 1400, and 2800 RPM respectively. For both stainless steel and copper smooth condensers, the following general observations apply: a) For the same external heat transfer coefficient, the heat transfer rate for the cylindrical smooth condenser is less than the equivalent tapered condenser. b) As the external heat transfer coefficient increases, this difference in heat transfer rate also increases. c) The rotational speed has a greater effect on the tapered condenser heat transfer rate. For example, the maximum heat transfer rate of a copper tapered condenser will increase by a factor of 1.67 when the rotational speed is increased from 700 to 2800 RPM. In the cylindrical copper condenser, for the same change in rotational speed, the heat transfer rate only increases by a factor of 1.51.

These same observations hold true for the smooth stainless steel condensers. But, due to a greater thermal resistance in the wall of the stainless steel condenser, the heat transfer rates are less for all cases considered. It should be noted that the thermal conductivity of stainless steel is only 4% the thermal conductivity of copper. This accounts for the increased thermal resistance.

Figures 14, 15, and 16 compare axially finned cylindrical with axially finned tapered copper condensers at 700, 1400 and 2800 RPM respectively. Note that the heat transfer rates of the cylindrical condensers are only slightly less than those of the tapered condensers. This is because the heat is primarily transferred through the extended surface, i.e., the fin. Thus the film condensate in the trough has less effect on the heat transfer rate. In fact, the average difference in heat transfer rate for 700 RPM and an external heat transfer coefficient of 100 Btu/hr-ft^2-F is 12.65%. This difference increases to 15.6% as the heat transfer coefficient is increased to 1000 Btu/hr-ft^2-F. However, as the rotational speed is increased to 2800

RPM, the corresponding average differences in heat transfer rates decrease to 12.58% and 13.33% respectively.

Figures 17, 18, and 19 compare the heat transfer rates of axially finned stailness steel cylindrical condensers with those of axially finned stainless steel tapered condensers at 700, 1400 and 2800 RPM respectively. Note the difference in heat transfer rate increases as the external heat transfer coefficient increases more than in the case of the copper condensers. At the low heat transfer coefficient, the limiting thermal resistance is that of the external surface ($1/h_{ext}$). However, as the external heat transfer coefficient is increased, the limiting thermal resistance becomes that of the wall due to the low thermal conductivity of stainless steel.

Another observation to be noted is the fact that the rotational speed has a greater effect on the heat transfer rate of the cylindrical condensers than on the tapered condensers. As the rotational speed increases, the film thickness decreases due to the greater centrifugal force exerted on the film. For the cylindrical condenser, the maximum film thickness is much greater than for a tapered condenser. For example, an axially finned stainless steel cylindrical condenser rotating at 1400 RPM with an external heat transfer coefficient of 1000 Btu/hr-ft^2-F has a maximum film thickness twice that of a corresponding tapered condenser. This being the case, higher rotational speeds will have a greater effect on the greater film thickness and the difference in heat transfer rates will decrease.

Figures 20, 21, and 22 compare the heat transfer rates of copper smooth cylindrical condensers with copper axially finned cylindrical condensers at 700, 1400 and 2800 RPM respectively. The figures indicate that for low external heat transfer coefficients, little is to be gained by the addition of axial fins. But, as the external heat transfer coefficient increases, the advantage becomes significant. As an example, consider Figure 21. For h=100 Btu/hr-ft^2-F, axial finning increase the heat transfer rate by 16%. But, for an external heat transfer coefficient of 1000 Btu/hr-ft^2-F, the heat transfer rate increases by 194%.

Note also, that as rotational speed increases the advantage to be gained by axial finning decreases. This can be explained by the fact that for an axially finned condenser, the majority of the heat is transferred by the fin surface. As the rotational speed increases, the film thickness in the trough decreases. This in turn will expose slightly more fin surface area. On the other hand, for the smooth condenser, the decrease in the film thickness will have a greater effect on heat transfer rate in that the thermal resistance of the film has decreased. Comparing the two geometries, the change in overall thermal resistances for the smooth condenser will be greater than that for the axially finned condenser accounting for the slight decrease in advantage with increasing rotational speed.

Figures 23, 24, and 25 correspond to Figures 20, 21, and 22 but for stainless steel condensers. The results are similar to the copper situation discussed above, by the effect caused by the increasing external heat transfer coefficient is not as dominant due to the high thermal resistance of the stainless steel wall material.

Figure 26 compares a smooth cylindrical copper condenser with a smooth cylindrical stainless steel condenser at 1400 RPM. As to be expected, the heat transfer rate of the copper condenser is greater than that of the stainless steel condenser due to the difference in the thermal conductivity of the two materials. The difference in the heat transfer rate is least for a low external heat transfer coefficient where the external thermal resistance is dominant. As the heat transfer coefficient increases, the thermal resistance of the wall of the condenser becomes more important resulting in an increasing difference between the two condensers.

Figure 27 provides a comparison of axially finned cylindrical copper and stainless steel condensers at 1400 RPM. The advantage of copper over stainless steel is obvious. Note that the heat transfer rate for the copper condenser at 500 Btu/hr-ft^2-F is nearly identical to the stainless steel condenser heat transfer rate at 1000 Btu/hr/-ft^2-F indicating the advantage of copper over stainless steel.

The final analysis compared the heat transfer rate of an axially finned copper condenser with a triangular fin profile to the heat transfer rate of an axially finned condenser with a rectangular fin profile. The rectangular fin was assumed to have an adiabatic tip. This comparison was accomplished for both cylindrical and tapered condensers. Table IV lists the parameters used in the analysis. These parameters are in addition to those listed in Table I. This comparison was only accomplished for the one set of operating parameters listed in Table IV. Note that the operating parameters chosen were the median values.

TABLE IV

List of Parameters Used in Rectangular/Triangular Fin Profile Comparison

Heat Transfer Coefficient	=	500 Btu/hr-ft^2-F
Rotational Speed	=	1400 RPM
Saturation Temperature	=	120.0 degrees F
Fin Height	=	0.05906 inches

Table V lists the results of the analyses.

TABLE V

Results of Rectangular/Triangular Fin Profile Comparison

CONDENSER GEOMETRY	FIN PROFILE	Q (Btu/hr)
tapered	triangle	6168.93
	rectangle	6143.03
cylindrical	triangle	5358.4
	rectangle	5338.00

Note that the heat transfer rate varied by only 0.4% for both the tapered and cylindrical condensers. The reason for this insignificant variation is provided in Table VI and VII. Table VI lists the convective surface temperature distribution of a symmetric unit section of a tapered condenser at the middle increment of the condenser. Temperature location 1 is located at the tip of the fin. For the rectangular profile, temperature location 1 is located at the intersection of the adiabatic surface (the tip of the fin), and the vertical surface of the fin. Temperature location 5 is located at the base of the fin. Locations 6-9 are located in the trough region and the remaining temperature locations are along the external surface of the section. Thus temperatures 1-5 provide the temperature distribution along the convective surface of the fin. Note that the temperature distribution is lower for the rectangular fin. Thus, the driving force for heat transfer, the temperature difference between the saturation temperature and the surface temperature is greater for the rectangular fin. Thus, in spite of the fact that the fin surface area for heat transfer has decreased by 11%, the average surface temperature difference has increased by 25%. It should also be noted that the average heat transfer coefficient for the fin surface for the rectangular fin also decreases by 12%. However, the increase in temperature difference is the dominant change that only allows a 0.4% decrease in heat transfer rate.

TABLE VI

Surface Temperature Distribution for Rectangular and Triangular Axially Finned Tapered Copper Condensers at Middle Increment of Condensers

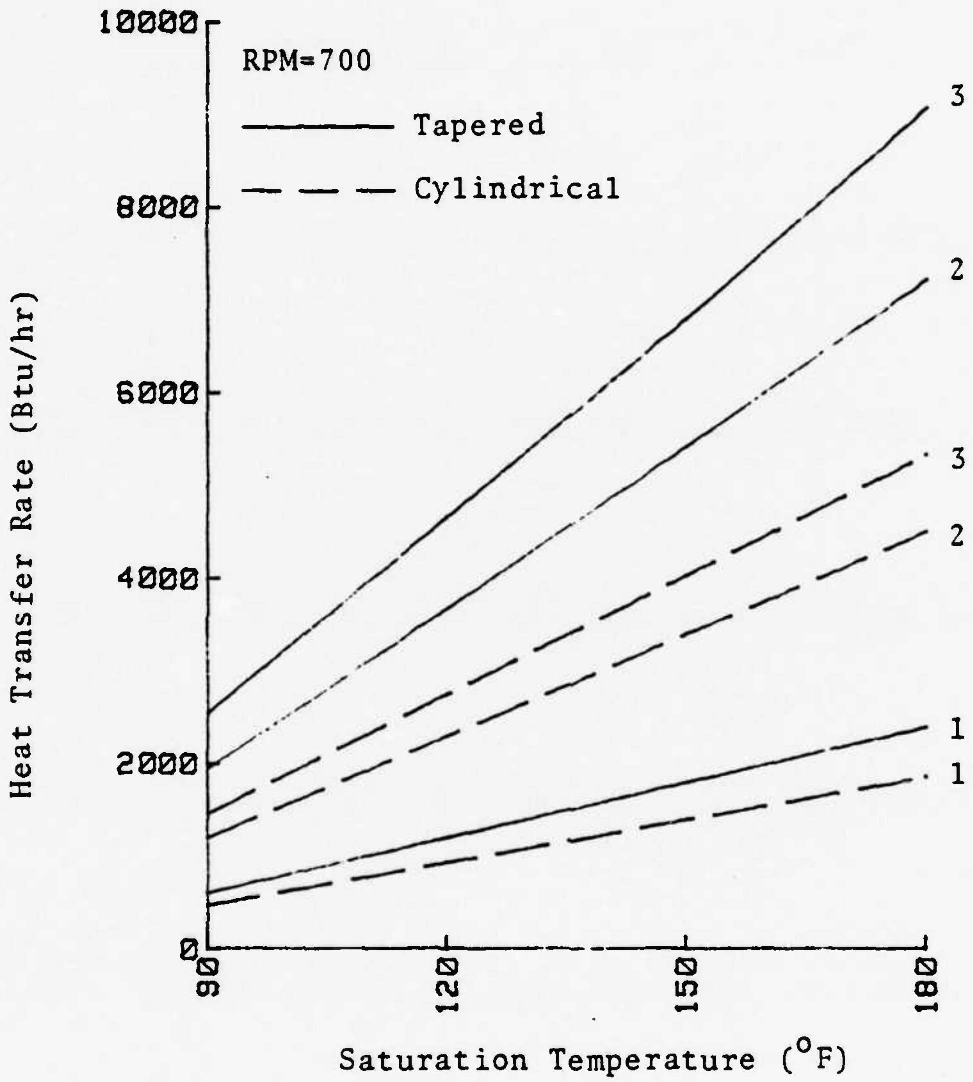
Temperature Location	Triangular Fin Temperature (F)	Rectangular Fin Temperature (F)
1	118.85	117.95
2	118.45	117.88
3	118.04	117.68
4	117.60	117.36
5	117.05	116.59
6	116.82	116.45
7	116.61	116.35
8	116.59	116.35
9	116.48	116.54
10	116.47	116.53
11	116.46	116.23
12	116.46	116.21
13	116.44	116.20
14	116.42	116.17
15	116.31	116.12
16	116.31	116.07
17	116.27	116.04
18	116.26	116.03

TABLE VII

Surface Temperature Distribution for Rectangular and Triangular Axially Finned Cylindrical Copper Condensers at Middle Increment of Condensers

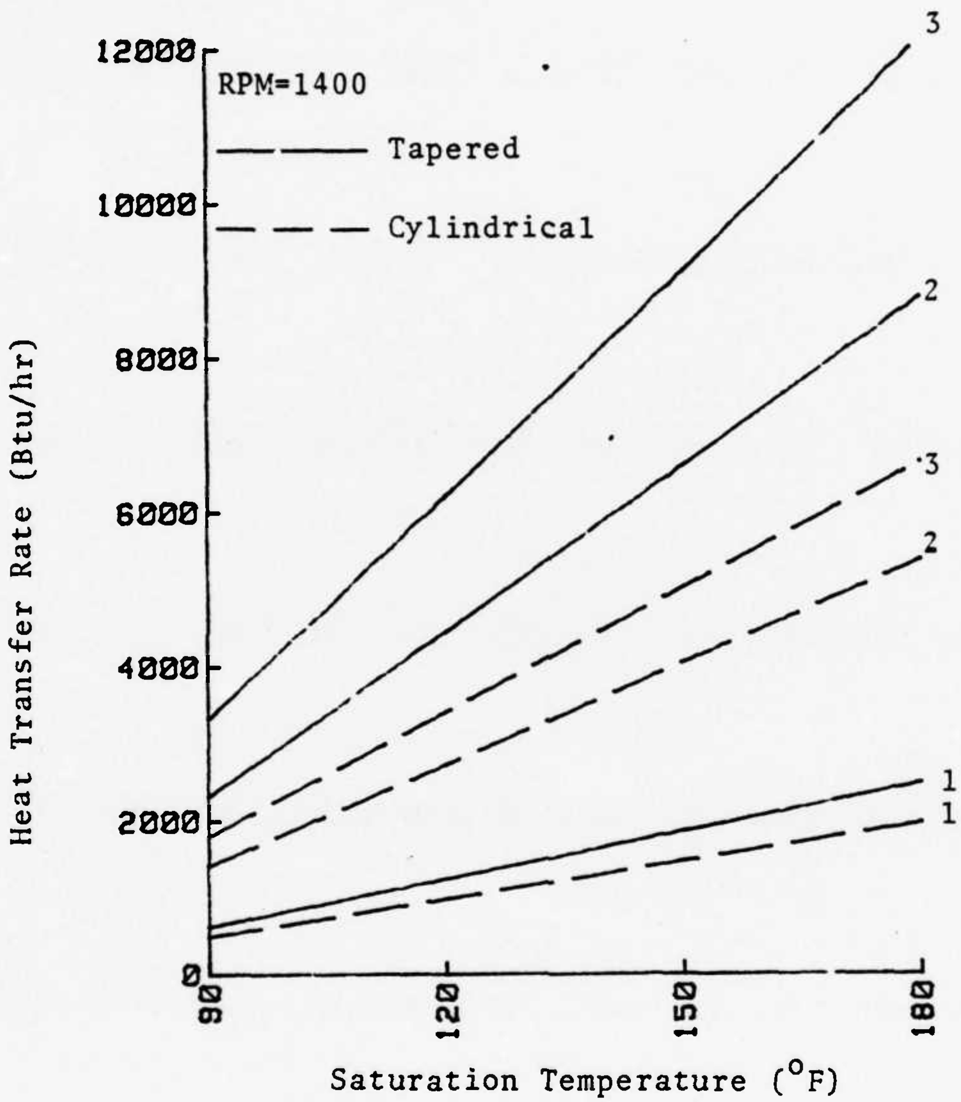
Temperature Location	Triangular Fin Temperature (F)	Rectangular Fin Temperature (F)
1	118.55	117.55
2	118.09	117.44
3	117.62	117.25
4	117.10	116.87
5	116.55	116.36
6	116.38	116.18
7	116.28	116.07
8	116.22	116.01
9	116.20	115.99
10	116.03	115.80
11	116.02	115.79
12	116.01	115.79
13	115.99	115.77
14	115.97	115.75
15	115.94	115.72
16	115.91	115.67
17	115.88	115.67
18	115.88	115.66

Table VII indicates a similar situation exists in the cylindrical condenser. Again note the decrease in the average surface temperature of the fin for the rectangular profile. The surface area and heat transfer coefficient has decreased, but the increased temperature difference compensates for these changes, limiting the overall decrease in heat transfer rate.



1. h_{ext} = 100 Btu/hr-ft^2-F
2. h_{ext} = 500 Btu/hr-ft^2-F
3. h_{ext} = 1000 Btu/hr-ft^2-F

Figure 8. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 700 RPM



1. h_{ext} = 100 Btu/hr-ft^2-^{o}F
2. h_{ext} = 500 Btu/hr-ft^2-^{o}F
3. h_{ext} = 1000 Btu/hr-ft^2-^{o}F

Figure 9. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 1400 RPM

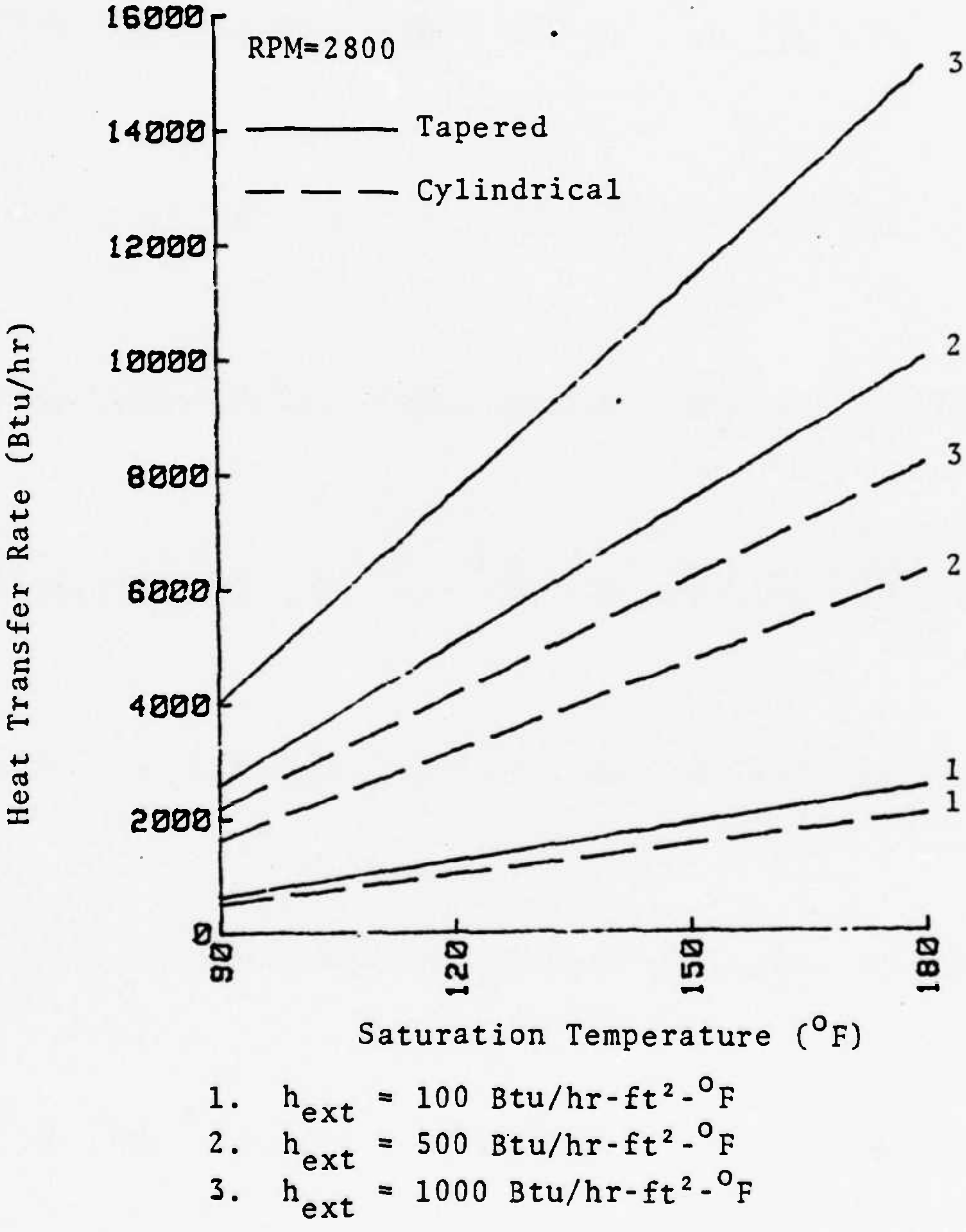
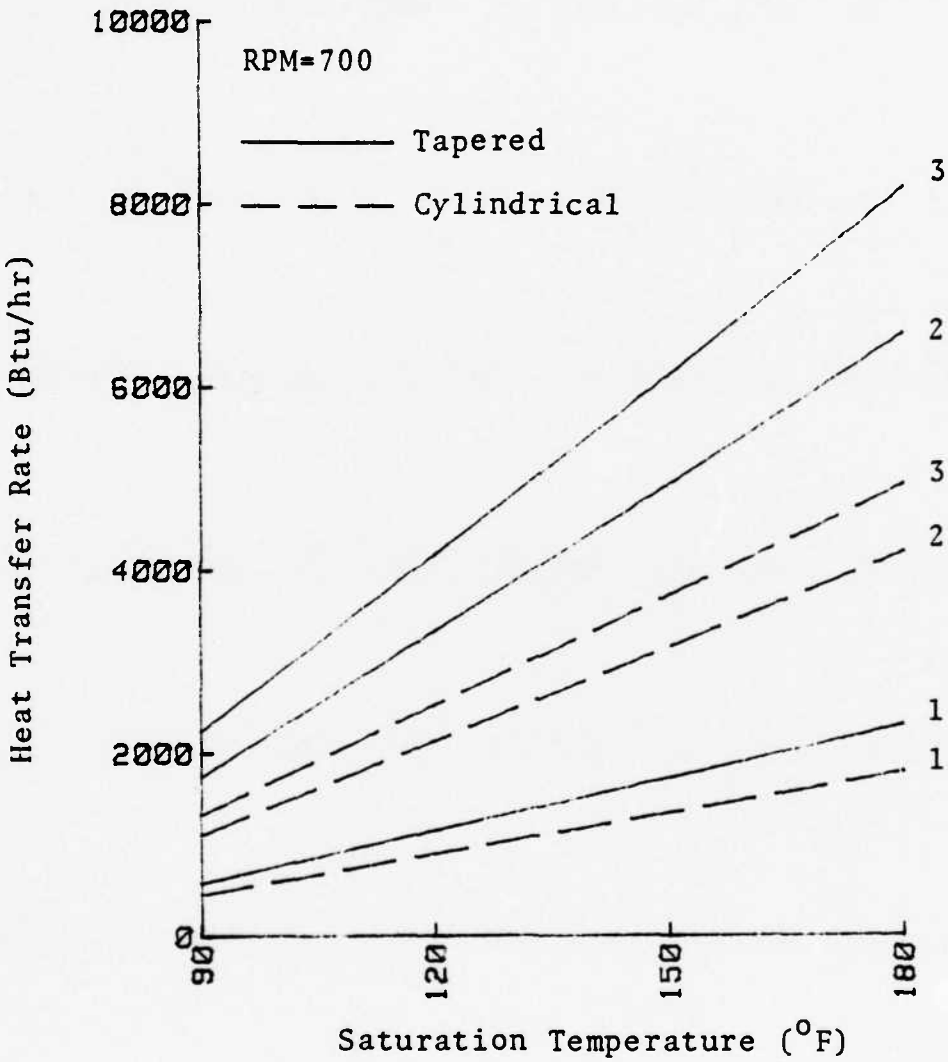
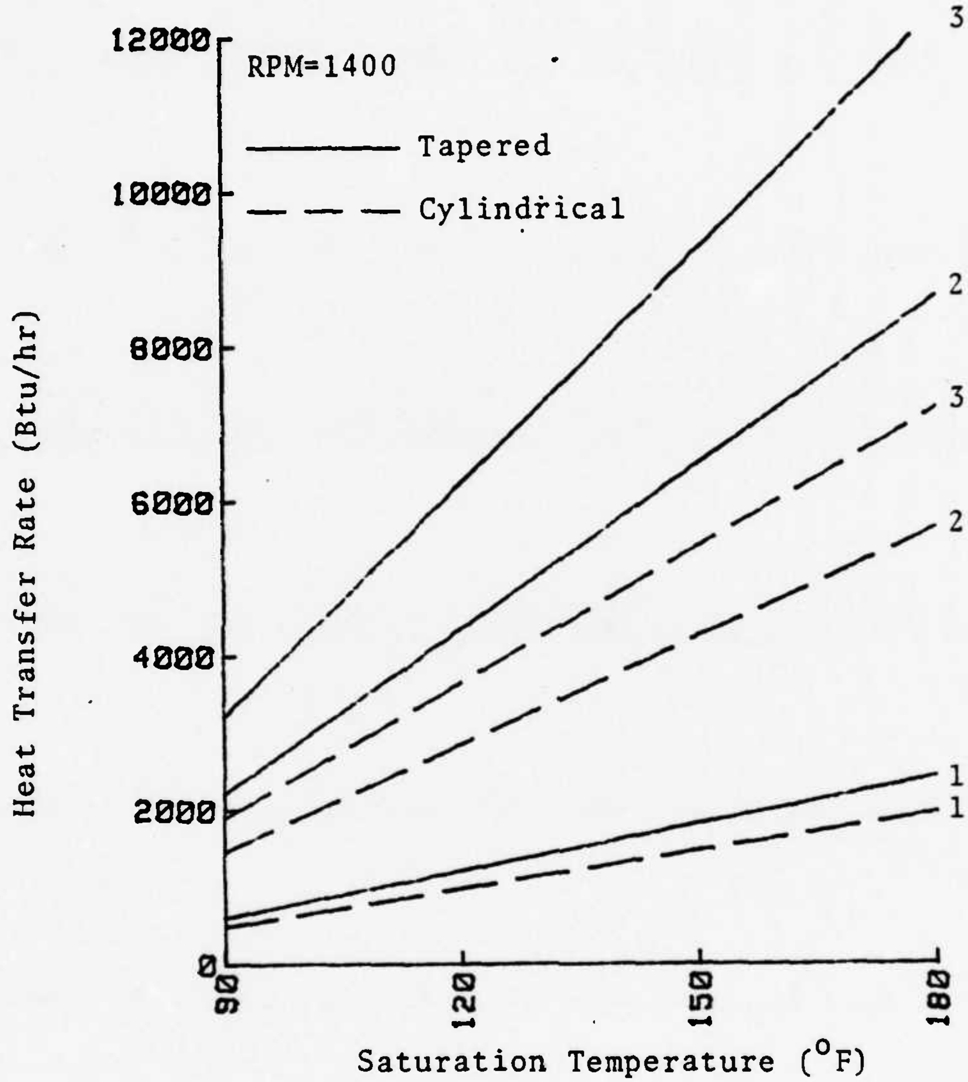


Figure 10. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 2800 RPM



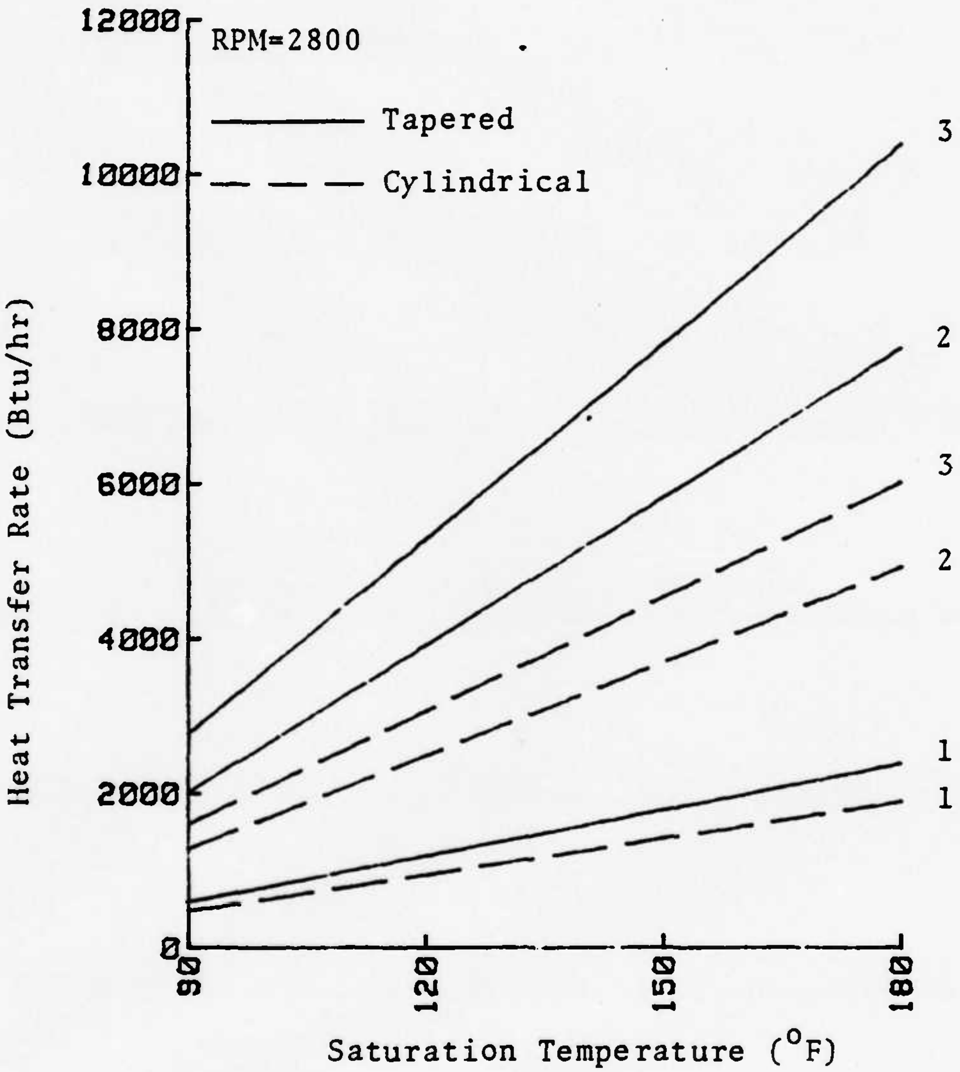
1. h_{ext} = 100 Btu/hr-ft^2-^{o}F
2. h_{ext} = 500 Btu/hr-ft^2-^{o}F
3. h_{ext} = 1000 Btu/hr-ft^2-^{o}F

Figure 11. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Stainless Steel Condensers at 700 RPM



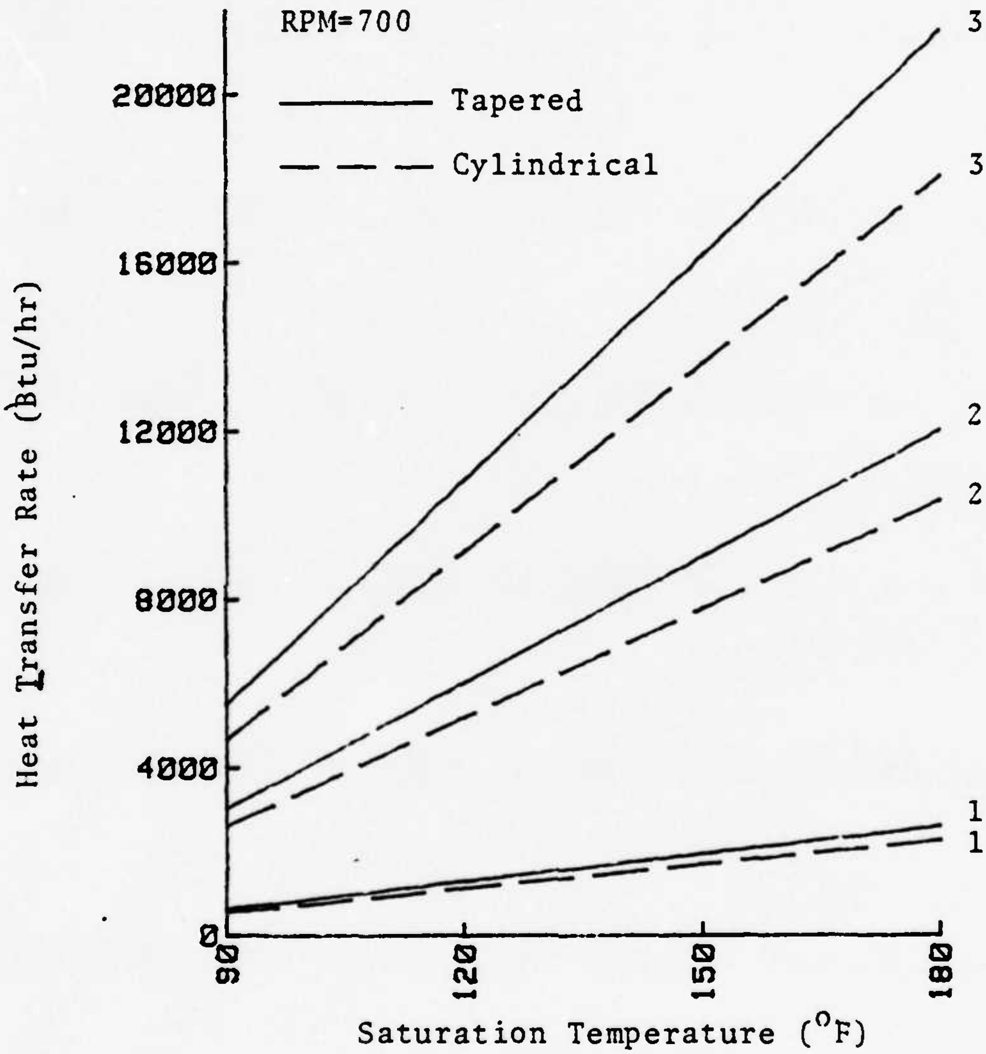
1. h_{ext} = 100 Btu/hr-ft^2-F
2. h_{ext} = 500 Btu/hr-ft^2-F
3. h_{ext} = 1000 Btu/hr-ft^2-F

Figure 12. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Stainless Steel Condensers at 1400 RPM



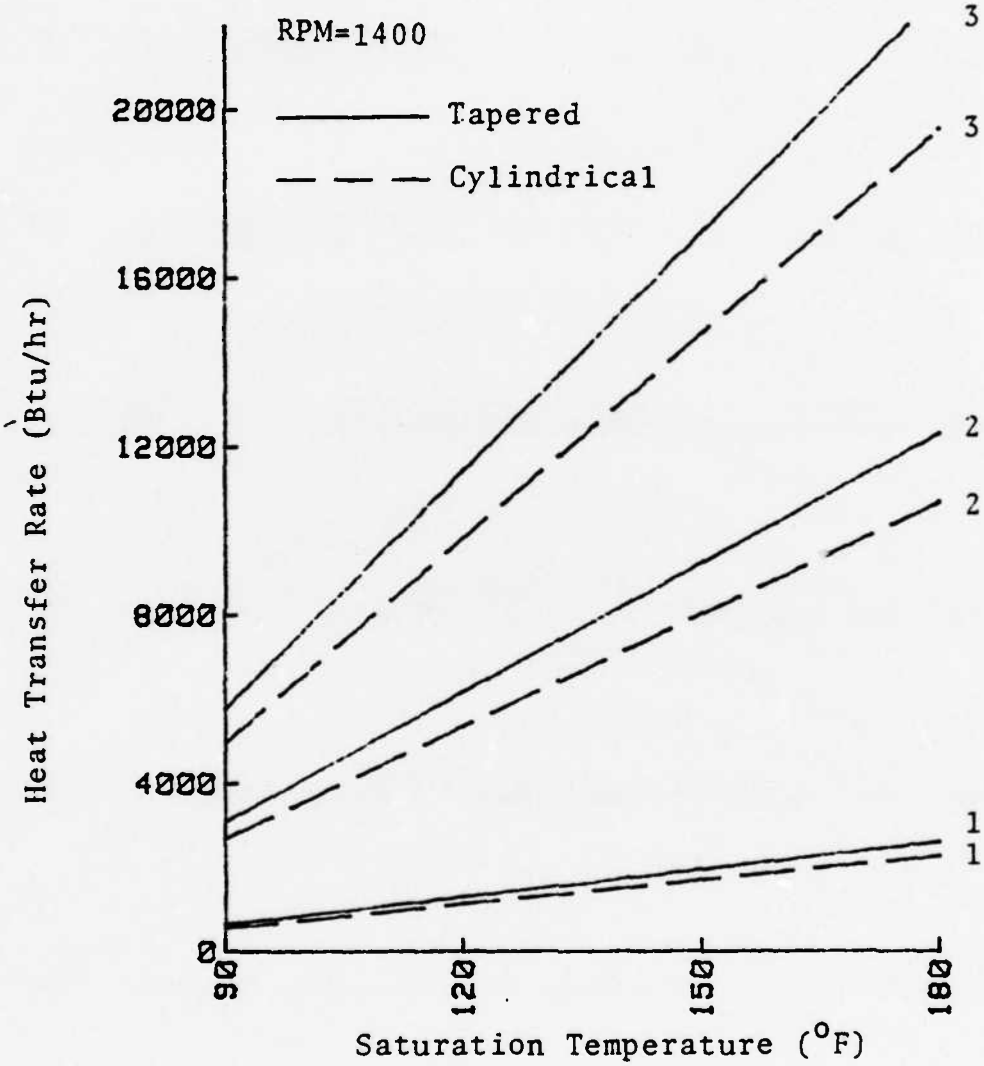
1. h_{ext} = 100 Btu/hr-ft^2-F
2. h_{ext} = 500 Btu/hr-ft^2-F
3. h_{ext} = 1000 Btu/hr-ft^2-F

Figure 13. Heat Transfer Rate versus Saturation Temperature for Smooth Tapered and Cylindrical Stainless Steel Condensers at 2800 RPM



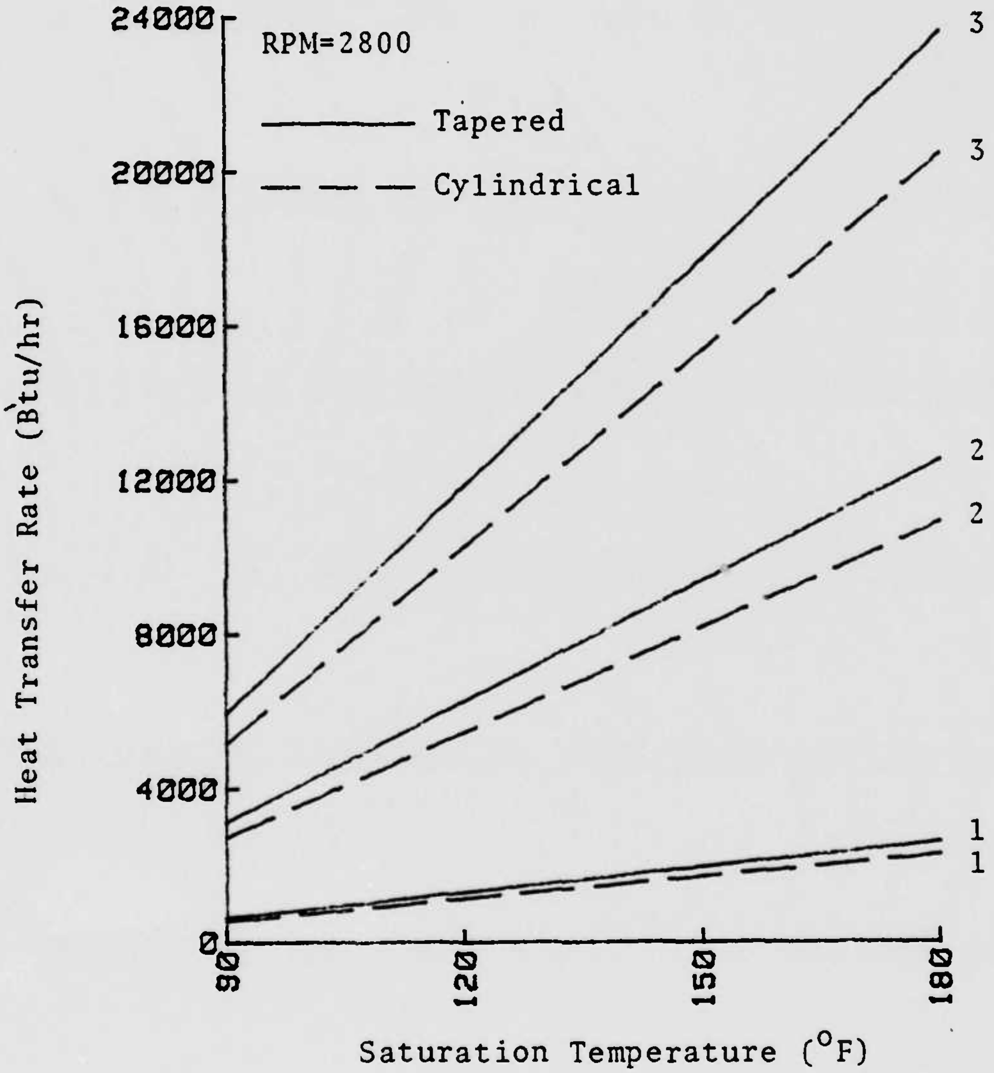
1. h_{ext} = 100 Btu/hr-ft^2-^oF
2. h_{ext} = 500 Btu/hr-ft^2-^oF
3. h_{ext} = 1000 Btu/hr-ft^2-^oF

Figure 14. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Copper Condensers at 700 RPM



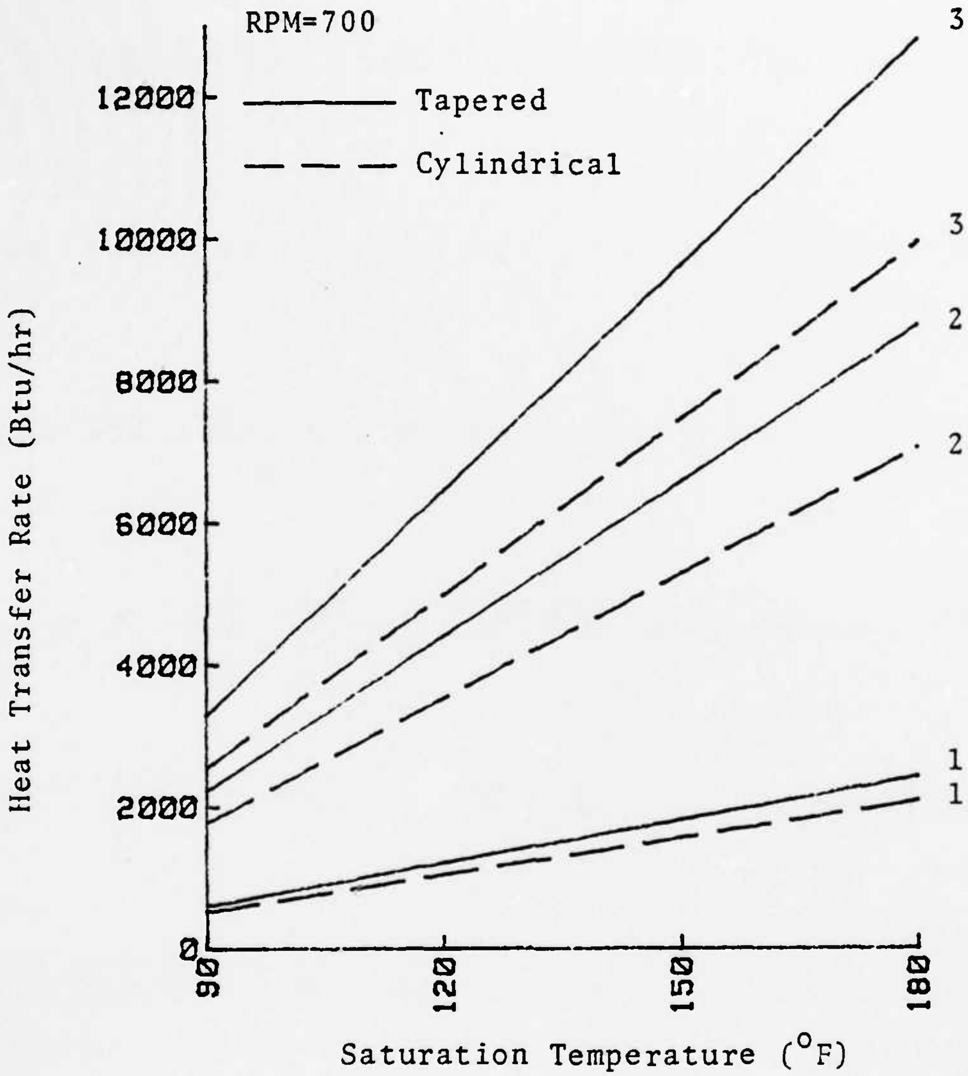
1. h_{ext} = 100 Btu/hr-ft^2-F
2. h_{ext} = 500 Btu/hr-ft^2-F
3. h_{ext} = 1000 Btu/hr-ft^2-F

Figure 15. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Copper Condensers at 1400 RPM



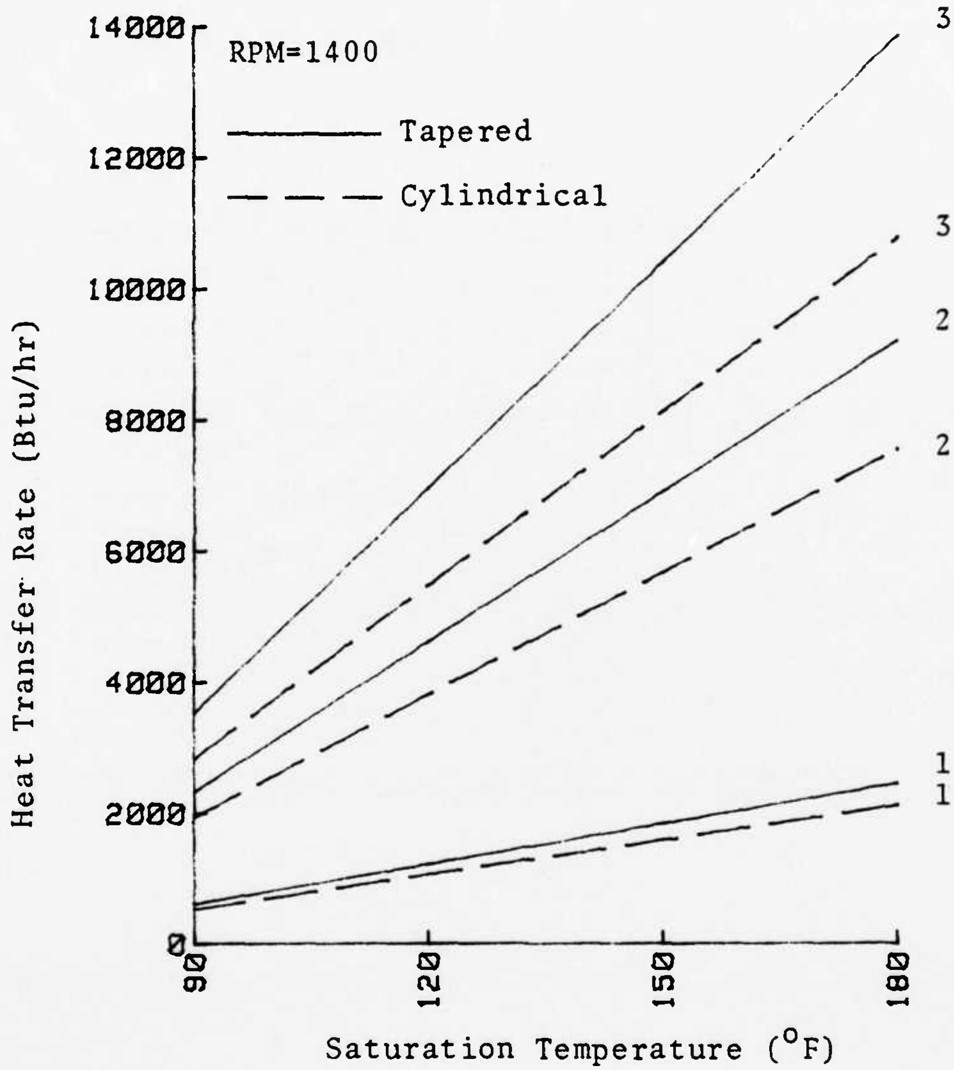
1. h_{ext} = 100 Btu/hr-ft²-°F
2. h_{ext} = 500 Btu/hr-ft²-°F
3. h_{ext} = 1000 Btu/hr-ft²-°F

Figure 16. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Copper Condensers at 2800 RPM



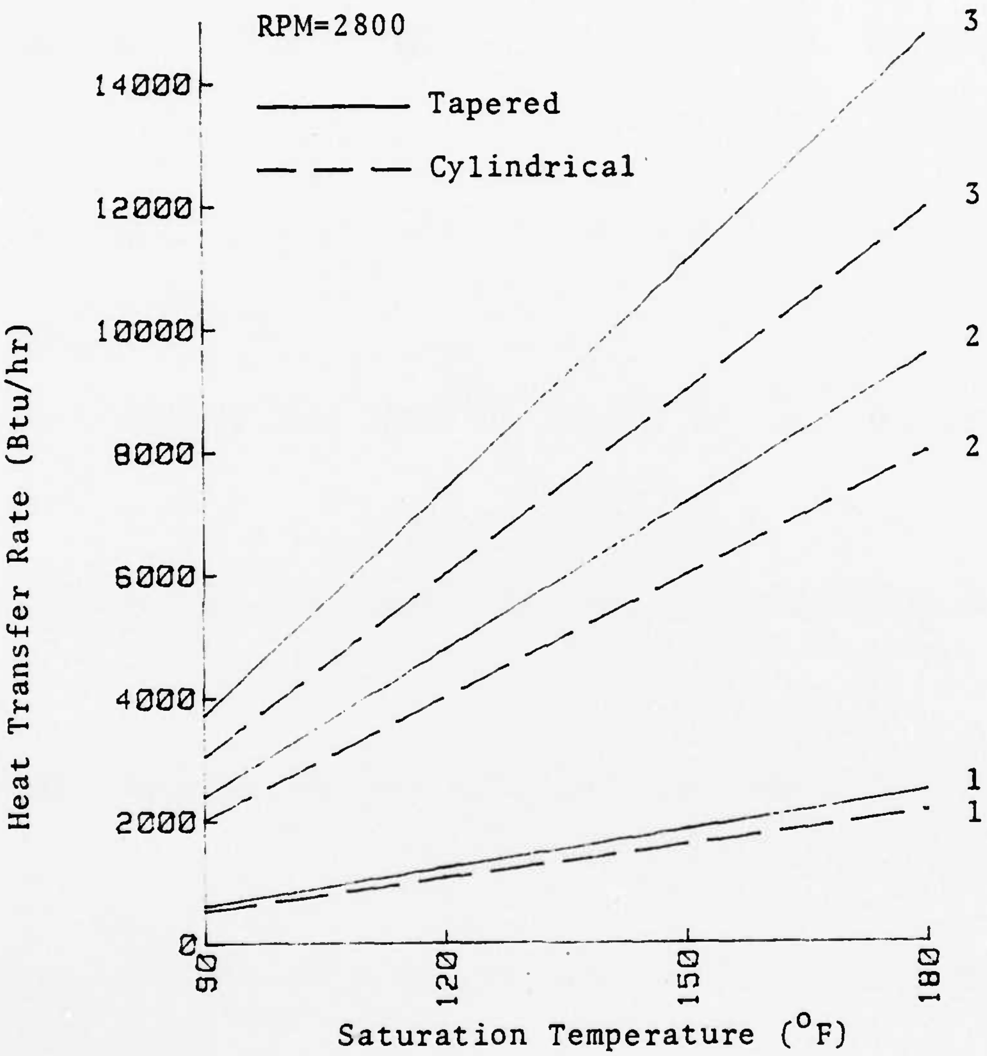
1. h_{ext} = 100 Btu/hr-ft²-°F
2. h_{ext} = 500 Btu/hr-ft²-°F
3. h_{ext} = 1000 Btu/hr-ft²-°F

Figure 17. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 700 RPM



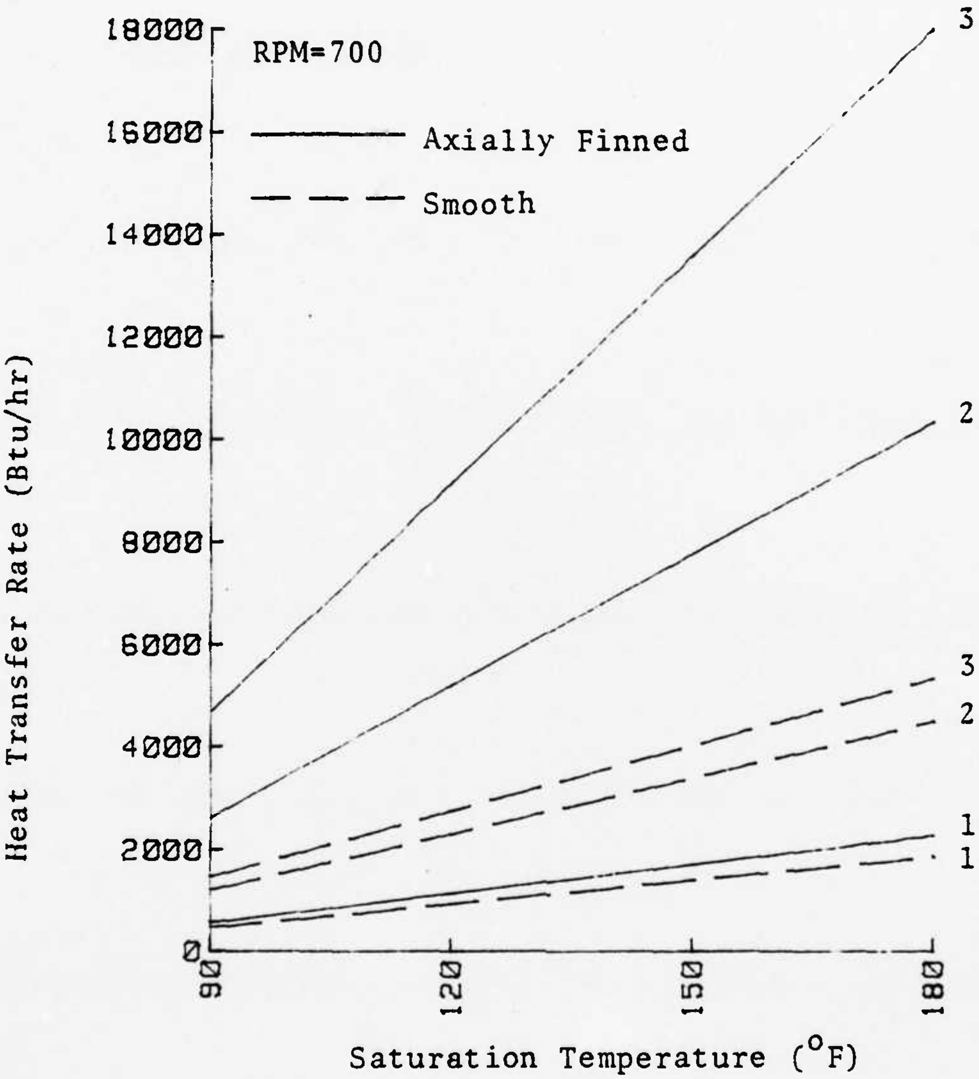
1. h_{ext} = 100 Btu/hr-ft^2-^{o}F
2. h_{ext} = 500 Btu/hr-ft^2-^{o}F
3. h_{ext} = 1000 Btu/hr-ft^2-^{o}F

Figure 18. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 1400 RPM



1. h_{ext} = 100 Btu/hr-ft²-°F
2. h_{ext} = 500 Btu/hr-ft²-°F
3. h_{ext} = 1000 Btu/hr-ft²-°F

Figure 19. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 2800 RPM



1. h_{ext} = 100 Btu/hr-ft²-°F
2. h_{ext} = 500 Btu/hr-ft²-°F
3. h_{ext} = 1000 Btu/hr-ft²-°F

Figure 20. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 700 RPM

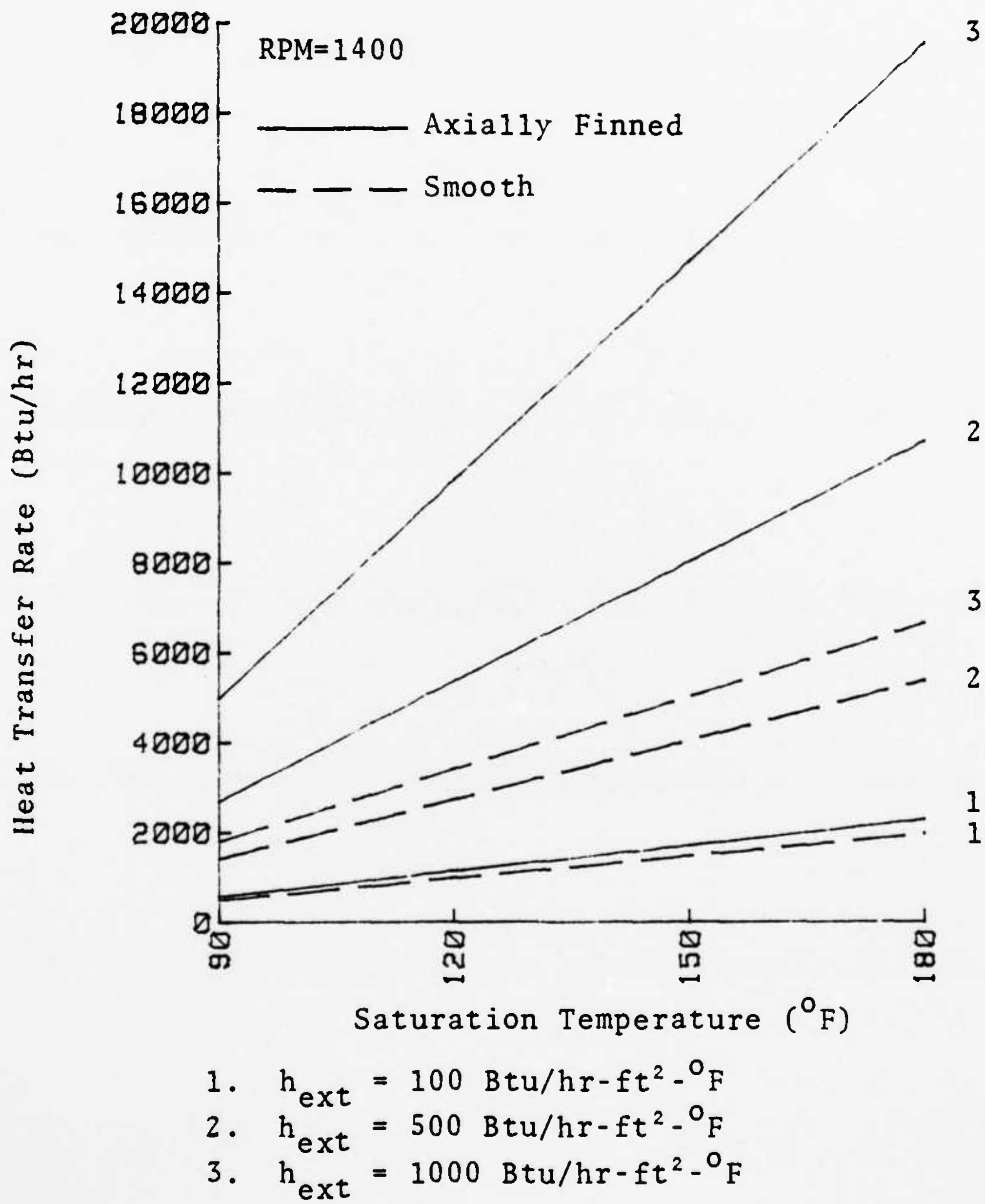
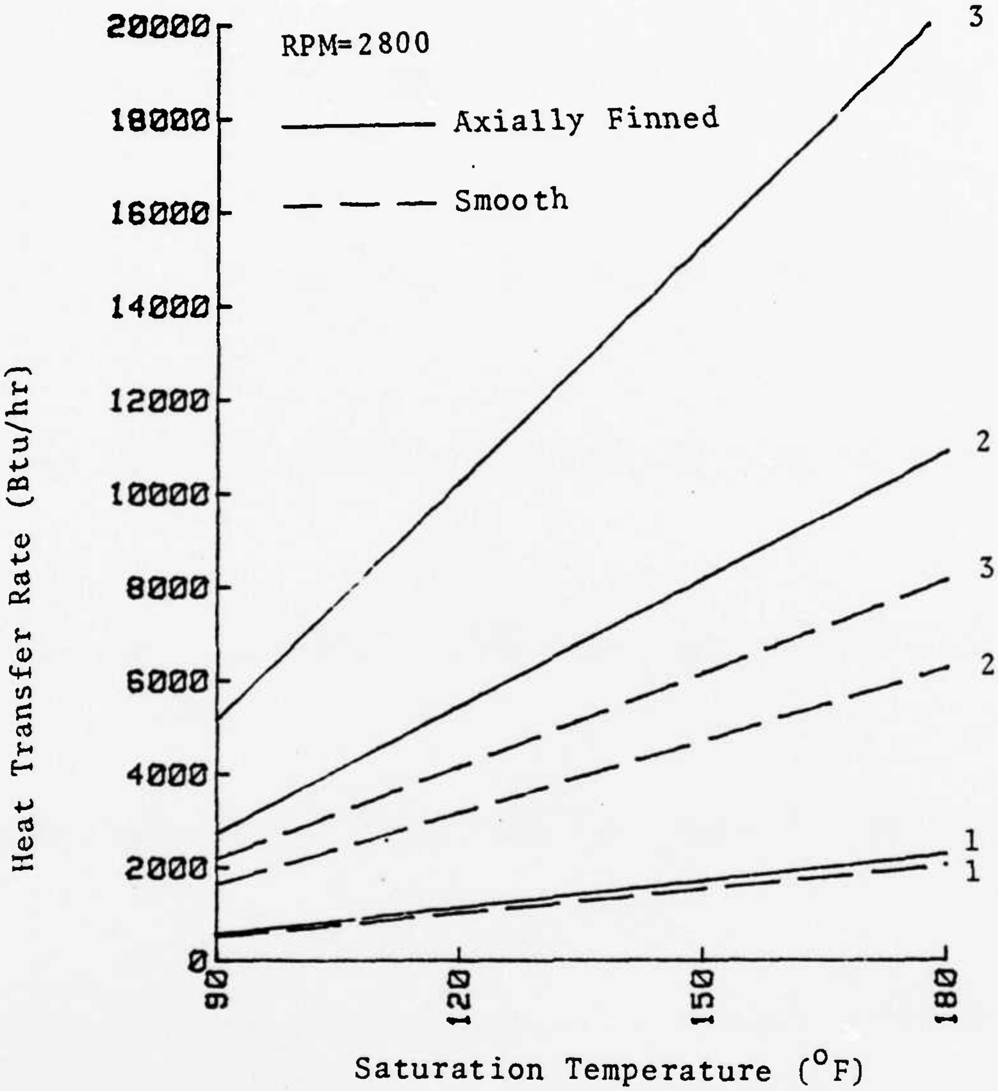
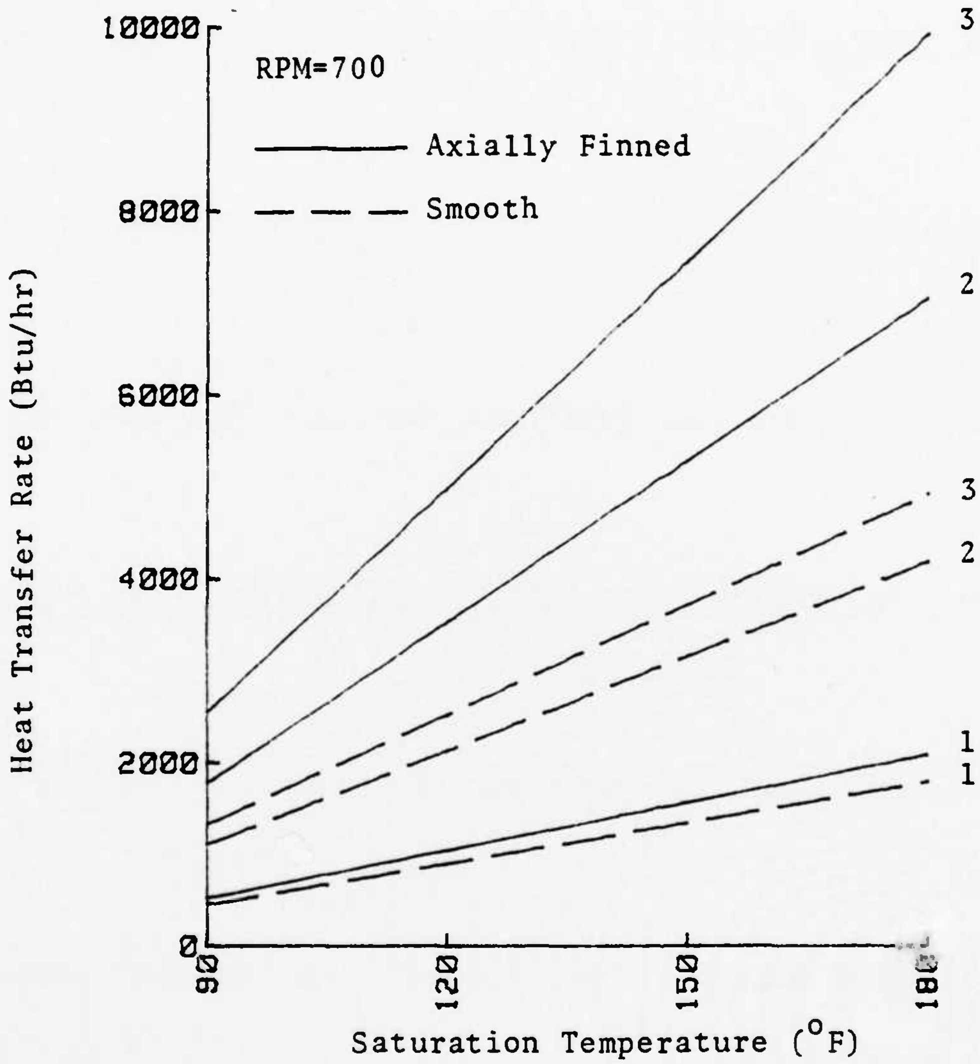


Figure 21. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 1400 RPM



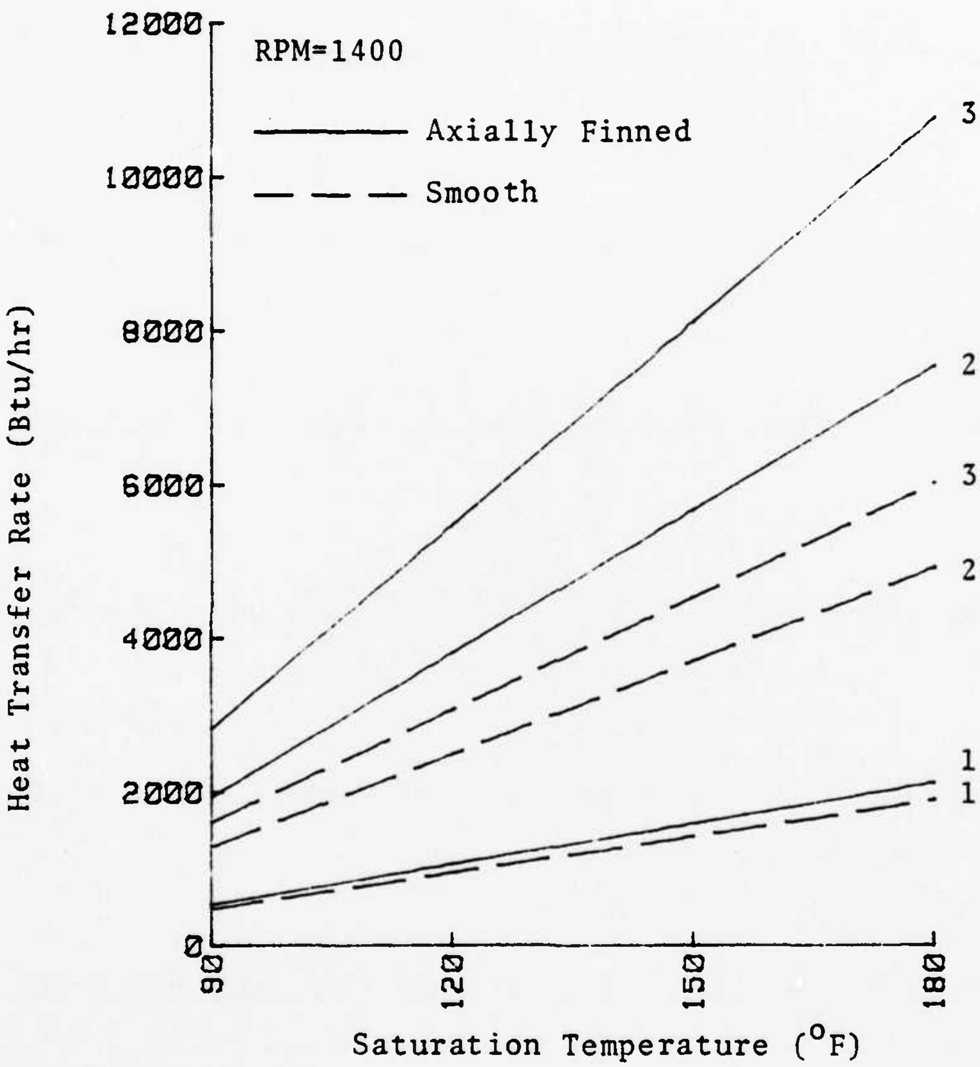
1. h_{ext} = 100 Btu/hr-ft²-^{o}F
2. h_{ext} = 500 Btu/hr-ft²-^{o}F
3. h_{ext} = 1000 Btu/hr-ft²-^{o}F

Figure 22. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 2800 RPM



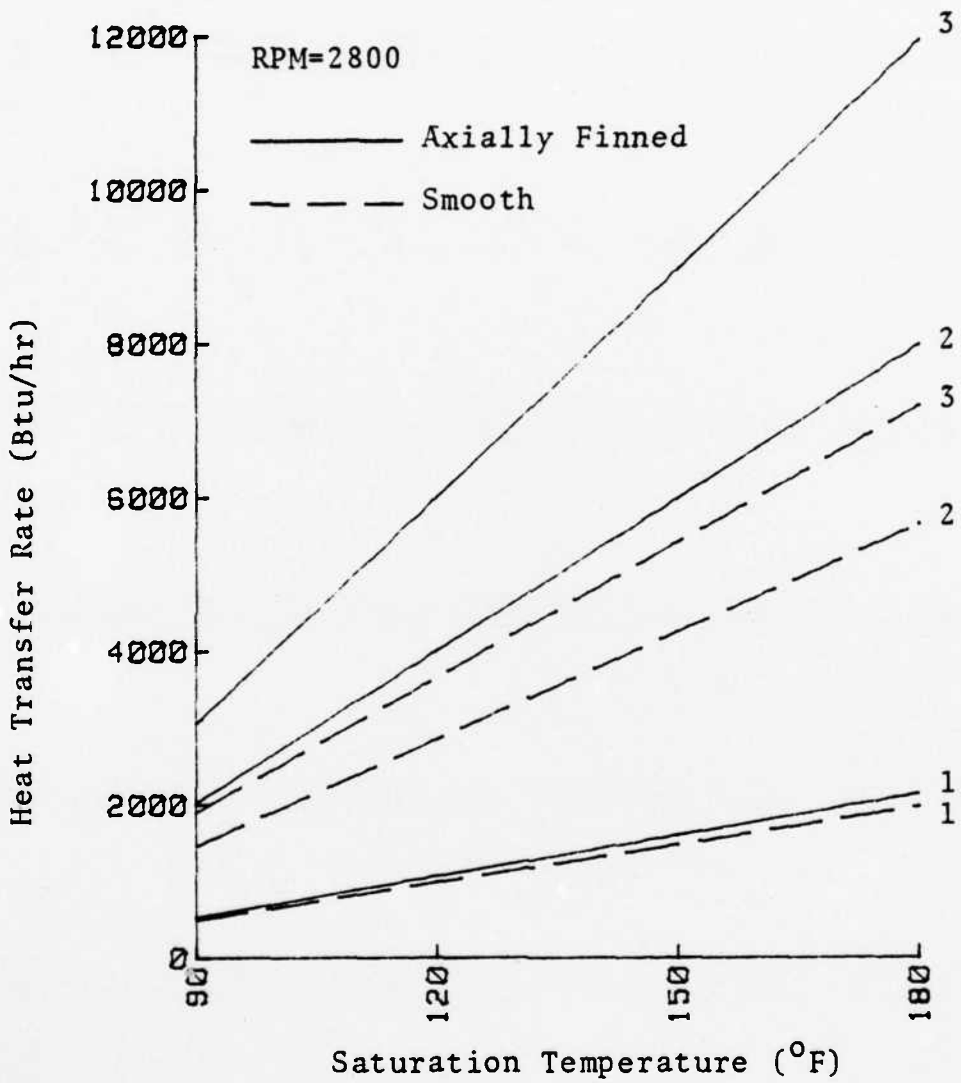
1. h_{ext} = 100 Btu/hr-ft²-°F
2. h_{ext} = 500 Btu/hr-ft²-°F
3. h_{ext} = 1000 Btu/hr-ft²-°F

Figure 23. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stainless Steel Cylindrical Condensers at 700 RPM



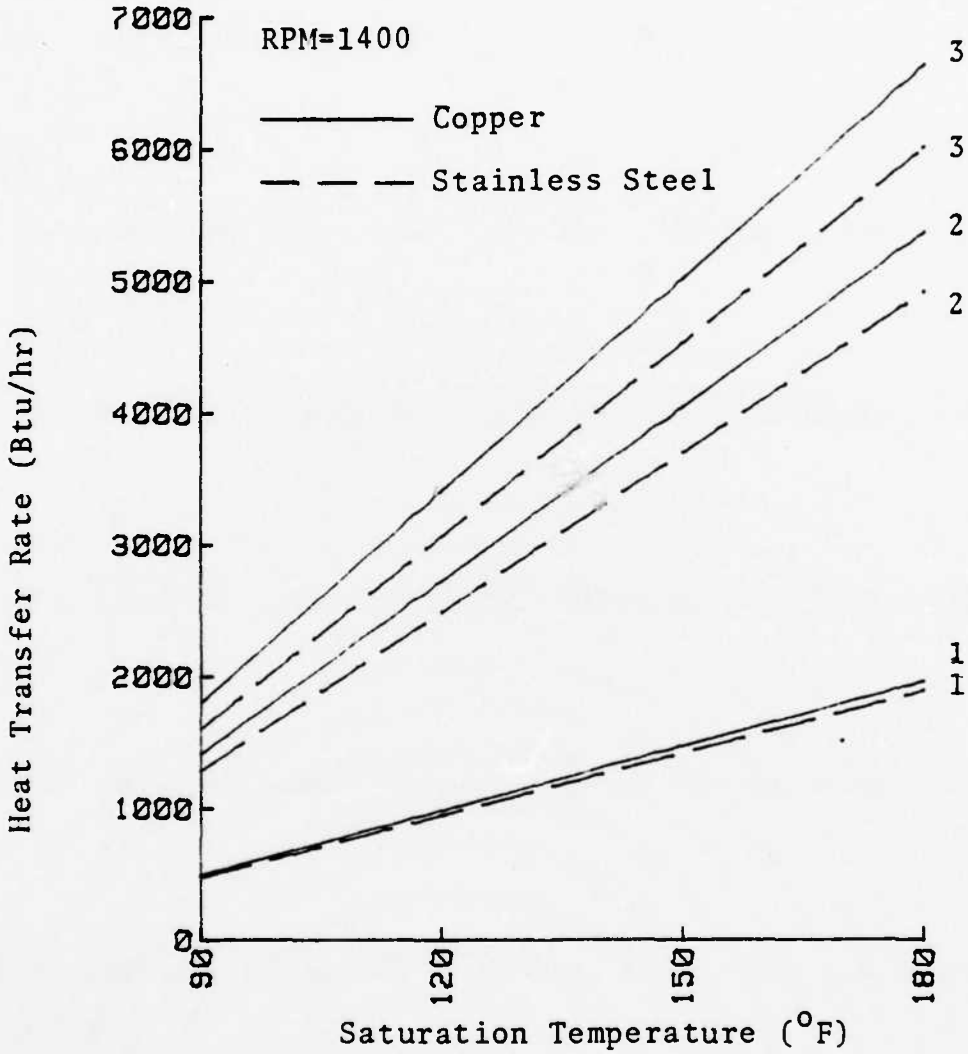
1. h_{ext} = 100 Btu/hr-ft²-°F
2. h_{ext} = 500 Btu/hr-ft²-°F
3. h_{ext} = 1000 Btu/hr-ft²-°F

Figure 24. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stainless Steel Cylindrical Condensers at 1400 RPM



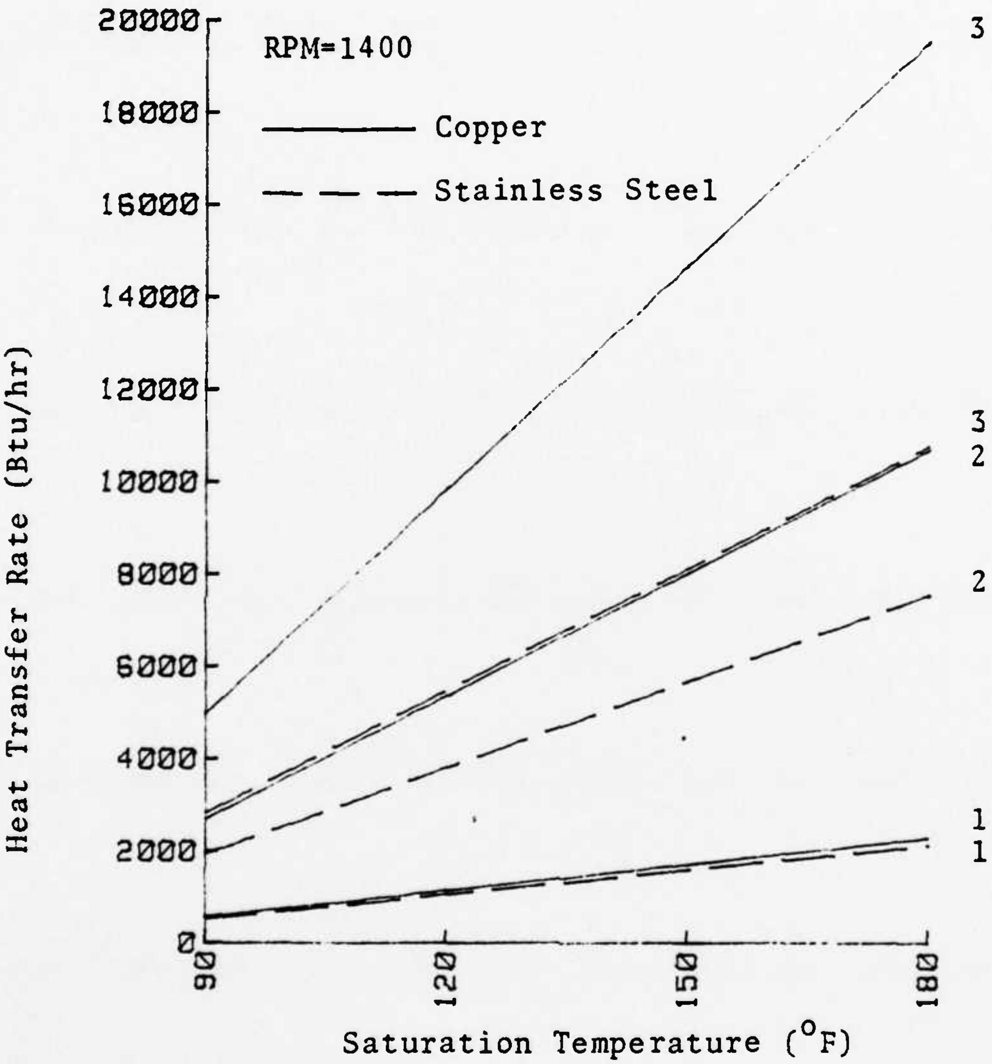
1. h_{ext} = 100 Btu/hr-ft^2-^{o}F
2. h_{ext} = 500 Btu/hr-ft^2-^{o}F
3. h_{ext} = 1000 Btu/hr-ft^2-^{o}F

Figure 25. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stainless Steel Cylindrical Condensers at 2800 RPM



1. h_{ext} = 100 Btu/hr-ft²-°F
2. h_{ext} = 500 Btu/hr-ft²-°F
3. h_{ext} = 1000 Btu/hr-ft²-°F

Figure 26. Heat Transfer Rate versus Saturation Temperature for Copper and Stainless Steel Smooth Cylindrical Condensers at 1400 RPM



1. h_{ext} = 100 Btu/hr-ft^2-$^{\circ}$F
2. h_{ext} = 500 Btu/hr-ft^2-$^{\circ}$F
3. h_{ext} = 1000 Btu/hr-ft^2-$^{\circ}$F

Figure 27. Heat Transfer Rate versus Saturation Temperature for Copper and Stainless Steel Axially Finned Cylindrical Condensers at 1400 RPM

V. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

The heat transfer rate of a cylindrical condenser is less than an equivalent tapered condenser. This decrease in heat transfer rate is most significant in a smooth condenser, where, depending on the external heat transfer coefficient and rotational speed, can be as great as 45%. This decrease in heat transfer rate becomes less significant for an axially finned condenser where the average decrease, for the range of heat transfer coefficients examined, was 13% for a copper condenser and 18% for a stainless steel condenser. When such factors as the cost and difficulty in manufacturing tapered axially finned condensers are considered, this 13% decrease in heat transfer rate becomes tolerable. From a practical standpoint, development and analysis of cylindrical axially finned condensers should be encouraged by these results. If environmental conditions permit, copper should be preferred over stainless steel due to its exceptionally high thermal conductivity and resulting higher heat transfer rate.

B. RECOMMENDATIONS

1) Build and experimentally test both smooth and axially finned cylindrical condensers to obtain experimental data for comparison with results of this analysis.

2) Develop models for rectangular and trapezoidal fin profiles with non-adiabatic tips and incorporate into code.

3) Experimentally test axially finned cylindrical condensers with rectangular and trapezoidal fin profiles to obtain data for comparison with theoretical results.

APPENDIX A

FILM PROFILE FINITE ELEMENT SOLUTION

A. SMOOTH CONDENSER

The analysis of the film profile in a smooth cylindrical condenser developed in Chapter II resulted in the following ordinary, nonlinear, second order, differential equation:

$$\delta^* \frac{d}{dx} \left[\frac{d\delta^*}{dx} \delta^{*3}\right] = - \frac{3k_f(T_{sat} - T_w)\mu}{\rho_f{}^2\omega^2 r\, \bar{h}_{fg}} \qquad \text{(eqn A.1)}$$

This equation can be rearranged and expanded to yield:

$$\delta^{*4} \frac{d^2\delta^*}{dx^2} + 3\delta^{*3} \left(\frac{d\delta^*}{dx}\right)^2 = -K \qquad \text{(eqn A.2)}$$

where $K = \dfrac{3k_f(T_{sat} - T_u)\mu}{\rho_f{}^2\omega^2 r\, \bar{h}_{fg}}$

The statement of the problem for the formulation of the Finite Element Method is:

$$\delta^{*4} \frac{d^2\delta^*}{dx^2} + 3\delta^{*3} \left(\frac{d\delta^*}{dx}\right)^2 = -K \qquad \text{(eqn A.3)}$$

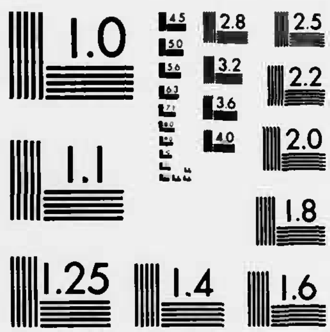
with the following boundary conditions:

a) at $x = 0$, $\delta^* = \delta^*_{max}$

b) at $x = 0$, $d\delta^*/dx = 0$

AD-A132 141 AN ANALYSIS OF SMOOTH AND AXIALLY FINNED ROTATING HEAT PIPE CONDENSERS(U) NAVAL POSTGRADUATE SCHOOL MONTEREY CA A F KLEINHOLZ JUN 83 2/2

UNCLASSIFIED F/G 12/1 NL



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

The domain (length of the condenser) is divided into elements of length Δx with the exception of the first and final elements which have length $\Delta x/2$, where Δx is the length of the condenser divided by the number of axial increments (NDIV). A system nodal point is located at each end of an element. Thus, system nodal point 1 is located at x=0 and the last system nodal point, which is equal to the number of elements plus 1 is located at the overfall into the evaporator, x=L. All internal nodal points are located at a position corresponding to the midpoint of the axial increment.

Define the approximate value of the film thickness in the following manner:

$$\bar{\delta} \simeq \delta_n^* (x) = \sum_{1}^{h} G_i d_i = G^T \underset{\sim}{d} \qquad \text{(eqn A.4)}$$

where $\bar{\delta}$ = the approximate value of the film thickness (δ*).

G_i = the global basis functions.

n = the number of system nodal points.

d = the solution vector.

On an element level, equation (A.4) becomes:

$$\bar{\delta}_e \simeq \bar{\delta}_e(x) = \sum_{i=1}^{4} g_i d_{i_e} \qquad \text{(eqn A.5)}$$

where g_i = the local basis functions.

Define two degrees of freedom at each nodal point, i.e., both $\bar{\delta}$ and $d\bar{\delta}/dx$ are continuous. Thus, the local basis functions used in the finite element solution are:

$$g_1 = 1 - \frac{3\zeta^2}{\ell^2} + \frac{2\zeta^3}{\ell^3} \qquad \text{(eqn A.6)}$$

$$g_2 = \zeta - \frac{2\zeta^2}{\ell} + \frac{\zeta^3}{\ell^2} \qquad \text{(eqn A.7)}$$

$$g_3 = \frac{3\zeta^2}{\ell^2} - \frac{2\zeta^3}{\ell^3} \qquad \text{(eqn A.8)}$$

$$g_4 = -\frac{\zeta^2}{\ell} + \frac{\zeta^3}{\ell^2} \qquad \text{(eqn A.9)}$$

where ℓ = the length of an element.

g_1 and g_3 = magnitude basis functions.

g_2 and g_4 = slope basis functions.

Substituting the approximate film thickness $\bar{\delta}_e$ into the differential equation (A.3) results in:

$$\bar{\delta}_e^4 \frac{d^2\bar{\delta}_e}{dx} + 3\bar{\delta}_e^3 \left(\frac{d\delta_e}{dx}\right)^2 = -K_e \qquad \text{(eqn A.10)}$$

To remove the "nonlinearity" from the problem, equation (A.10) is modified in the following manner:

$$\eta^4 \frac{d^2\bar{\delta}_e}{dx^2} + 3\eta^3\eta' \frac{d\bar{\delta}_e}{dx} = -K_e \qquad \text{(eqn A.11)}$$

η is defined as the approximate value of the film thickness from the previous iteration. In like manner, η' is defined as the approximate value of the rate of change of film thickness with respect to x from the previous iteration.

Forming the residual of equation (A.11) yields:

$$R_n = \eta^4(G^{T''}\underline{d}) + 3\eta^3\eta'(G^{T'}\underline{d}) + K \qquad \text{(eqn A.12)}$$

Invoking the Galerkin criterion for the determination of the solution vector d, i.e.

$$\int \underset{\sim}{G}_i R_n dx = \underset{\sim}{0} \quad i = 1,2,3,\ldots n \qquad \text{(eqn A.13)}$$

yields:

$$\eta^4 \int \underset{\sim}{G} G^{T''} \underset{\sim}{d} dx + 3\eta^3\eta' \int \underset{\sim}{G} G^{T'} \underset{\sim}{d} dx + K \int \underset{\sim}{G} dx = \underset{\sim}{0} \qquad \text{(eqn A.14)}$$

Each of the integrals in equation (A.14) are defined in the following manner:

$$\int \underset{\sim}{G} G^{T''} \underset{\sim}{d} dx = \sum_1^n g g^{T''} d_e dx \qquad \text{(eqn A.15)}$$

Or on an elemental level:

$$\int g g^{T''} d_e dx = \int_\ell \begin{Bmatrix} g_1 \\ g_2 \\ g_3 \\ g_4 \end{Bmatrix} \langle g_1'' \; g_2'' \; g_3'' \; g_3'' \rangle \, d_e dx \qquad \text{(eqn A.16)}$$

This integration results in the following 4x4 local elemental A matrix for any element:

$$[A]_e = \begin{bmatrix} \frac{-6}{\ell} & \frac{-11}{10} & \frac{6}{5\ell} & \frac{-1}{10} \\ \frac{-1}{10} & \frac{-2\ell}{15} & \frac{1}{10} & \frac{\ell}{30} \\ \frac{6}{5\ell} & \frac{1}{10} & \frac{-6}{5\ell} & \frac{11}{10} \\ \frac{-1}{10} & \frac{\ell}{10} & \frac{1}{10} & \frac{-2\ell}{15} \end{bmatrix}$$

In a similar manner, let

$$\int \underset{\sim}{G}G^{T'}\underset{\sim}{d}dx = \sum_{1}^{n} \int_{\ell} g\underset{\sim}{g}^{T'} d_e dx \qquad \text{(eqn A.17)}$$

and on an elemental level:

$$\int_{\ell} \underset{\sim}{g}g^{T} d_e dx = \int_{\ell} \begin{Bmatrix} g_1 \\ g_2 \\ g_3 \\ g_4 \end{Bmatrix} \langle g_1{}' \; g_2{}' \; g_3{}' \; g_4{}' \rangle \; d_e dx \qquad \text{(eqn A.18)}$$

This integration results in the following 4x4 local elemental B matrix for any element:

$$[B]_e = \begin{bmatrix} \frac{-1}{2} & \frac{\ell}{10} & \frac{1}{2} & \frac{-\ell}{10} \\ \frac{-\ell}{10} & 0 & \frac{\ell}{10} & \frac{-\ell^2}{60} \\ \frac{-1}{2} & \frac{-\ell}{10} & \frac{1}{2} & \frac{\ell}{10} \\ \frac{\ell}{10} & \frac{\ell^2}{60} & \frac{-\ell}{10} & 0 \end{bmatrix}$$

Lastly, let

$$\int \underset{\sim}{G} dx = \sum_{1}^{n} \int_{\ell} \underset{\sim}{g} dx \qquad \text{(eqn A.19)}$$

or, on an elemental level this becomes:

$$\int_{\ell} \underset{\sim}{g} dx = \int_{\ell} \begin{Bmatrix} g_1 \\ g_2 \\ g_3 \\ g_4 \end{Bmatrix} dx \qquad \text{(eqn A.20)}$$

This integration results in the following local elemental column F vector:

$$[F]_e = \begin{bmatrix} \frac{\ell}{2} \\ \frac{\ell^2}{12} \\ \frac{\ell}{2} \\ \frac{-\ell^2}{12} \end{bmatrix}$$

Thus, for a given element, equation (A.12) becomes:

$$[\eta_e^{4} [A]_e + 3\, \eta_e^{3}\, \eta_e' [B]_e]\, \underline{d}_e = -K_e \underline{F}_e \qquad \text{(eqn A.21)}$$

As mentioned above, η and η' are the approximate values of δ^* and $d\delta^*/dx$ respectively from the previous iteration. For the initial iteration, η is set equal to δ^*_{max} and η' is set equal to 0.

Each elemental matrix is placed in a global system [A] matrix with the location in the matrix based upon the local/global nodal point correspondence. For example, for element Nr. 2 the nodal points corresponding to local nodal points 1,2,3, and 4 are global nodal points 3,4,5, and 6 respectively. Therefore, the sum of the two 4x4 elemental matrices ([A] and [B]) and multiplying constants will form a single 4x4 local matrix. Element (1,1) of this local matrix will be placed in the global A matrix location (3,3) and element (4,4) of the local elemental matrix will be placed in the global A matrix location (6,6). Ultimately, the following global system would be assembled:

$$[A]_{mxm} \cdot \underset{\sim}{d} = \underset{\sim}{F} \qquad \text{(eqn A.22)}$$

Note that the global A matrix would be an mxm size matrix where m is equal to twice the number of system nodal points to account for the two degrees of freedom at each nodal point. In a similar fashion, $\underline{d}$ and $\underline{F}$ would be column vectors of size m.

Once the system is assembled the boundary conditions are applied. This is done in the following manner: For boundary condition (a), A(1,1) is set equal to 1.0 and the remaining elements in the first row, i.e. A (1,i) i=2,3,4,...m are set equal to 0.0. Then F(1) is set equal to δ^*_{max}. δ^*_{max} is initially determined by a relationship developed by Leppert and Nimmo [Refs. 8 and 9]. This establishes the value of d(1) as δ^*_{max} . In a similar manner, boundary condition (b) is applied by setting A(2,2) equal to 1.0 and all other second row elements of the global A matrix are set equal to 0.0. Then, F(2) is set equal to 0.0. This establishes the solution vector d(2) as 0.0. Note that d(2) corresponds to the slope at the first nodal point.

One additional boundary condition is required; this is the value of the film thickness at x=L. This boundary condition is necessary to completely specify the problem. The value of the film thickness at the overfall may take on any value depending on the geometry at the overfall. In the case of this analysis, this value was taken as $0.25 \cdot \delta^*_{max}$. This value was chosen for the following reason: Leppert and Nimmo [Refs. 8 and 9], in a similar analysis for laminar film condensation on a horizontal surface derived an analytical solution to equation (A.1), assuming a constant surface temperature. They found the film profiles for δ^* with the overfall value less than $0.40 \cdot \delta^*_{max}$ were essentially constant and thus any value of δ^* at the overfall less than $0.40 \cdot \delta^*_{max}$ would result in the same profile. In the verification of the finite element solution, not only was this found to be the case, but it was also found that the heat transfer rate was relatively insensitive to the shape of the film profile. In fact, it was found that the film thickness at the overfall could be increased to a value as great a $0.90 \cdot \delta^*_{max}$ and the resulting variation in heat transfer rate was only 10%. This being the case, the value of $0.25 \cdot \delta^*_{max}$ was arbitrarily chosen for δ_{min}.

The third boundary condition was applied by setting A(m-1,i)= 0.0 where i=1,2,3...m. Then, the global matrix element A(m-1,m-1) was set equal to 1.0. Finally, F(m-1) was set equal to $0.25 \cdot \delta^*_{max}$.

Once all three boundary conditions were applied, the system given by equation (A.22) was solved for $\underline{d}$. The values of the approximate film thickness, i.e., d(i), i=1,3,5,...m-1 are then compared to the values of film thickness from the previous iteration. If the relative difference is less than or equal to a specified convergence criterion (i.e., 0.0001) at all nodal points, convergence is met and the latest values of d are the solution values of δ^*.

If convergence is not met, the values of d are saved for the next iteration where they are used to determine η and η' as discussed above. This iterative process is continued until convergence is met or until a maximum number of iterations have occurred.

B. AXIALLY FINNED CONDENSER

The finite element solution for the film profile in an axially finned cylindrical condenser is very similar to that of a smooth cylindrical condenser. For this reason, only the variations in the development will be addressed. From Chapter II, the differential equation for mass flow rate in an axially finned cylindrical condenser is given by:

$$\delta^* \frac{d}{dx}\left[\frac{d\delta^*}{dx}\left(\varepsilon\delta^{*3} + \delta^{*4}\tan\alpha\right)\right] = -\frac{3k_f(T_{sat} - T_w)\mu\varepsilon}{\rho_f^{\,2}\omega^2 r\,\bar{h}_{fg}}$$

$$-2\delta^*\cos\alpha\left[\frac{4k_f(T_{sat} - T_{avg})\mu z^*}{\rho_f^{\,2}\omega^2 r\,h_{fg}\cos\alpha}\right]^{3/4} \qquad \text{(eqn A.23)}$$

This equation can be rearranged and expanded to yield:

$$\frac{d^2\delta^*}{dx}(\varepsilon\delta^{*4} + \delta^{*5}\tan\alpha) + \frac{d\delta^*}{dx}\left(3\varepsilon\delta^{*3}\frac{d\delta^*}{dx} + 4\delta^{*4}\tan\alpha\frac{d\delta^*}{dx}\right)$$

$$= -K_1 - K_2\delta^* \qquad \text{(eqn A.24)}$$

where $K_1 = \dfrac{3k_f(T_{sat} - T_w)\mu\varepsilon}{\rho_f{}^2\omega^2 r\,\bar{h}_{fg}}$

$$K_2 = 2\cos\alpha\left[\frac{4k_f(T_{sat} - T_{avg})\mu z^*}{\rho_f{}^2\omega^2 r\, h_{fg}\cos\alpha}\right]^{3/4}$$

Substituting equation (A.5) into equation (A.24) results in:

$$\frac{d^2\bar{\delta}_e}{dx^2}(\varepsilon\bar{\delta}_e^4 + \bar{\delta}_e^5\tan\alpha) + \frac{d\bar{\delta}_e}{dx}\left(3\varepsilon\bar{\delta}_e^3\frac{d\bar{\delta}_e}{dx} + 4\bar{\delta}_e^4\tan\alpha\frac{d\bar{\delta}_e}{dx}\right)$$

$$= -K_1 - K_2\bar{\delta}_e \qquad \text{(eqn A.25)}$$

To remove the "nonlinearity" from the problem, equation (A.25) is modified in the following manner"

$$\frac{d^2\bar{\delta}_e}{dx^2}(\varepsilon\gamma^4 + \gamma^5\tan\alpha) + \frac{d\bar{\delta}_e}{dx}(3\varepsilon\gamma^3\gamma' + 4\gamma^4\gamma'\tan\alpha)$$

$$= -K_1 - K_2\gamma \qquad \text{(eqn A.26)}$$

Here, γ and γ' are defined by the following relationships:

$$\gamma = \bar{\delta}_i + R^*(\bar{\delta}_i - \bar{\delta}_{i-1}) \qquad \text{(eqn A.27)}$$

$$\gamma' = \bar{\delta}'_i + R^*(\bar{\delta}'_i - \bar{\delta}'_{i-1}) \qquad \text{(eqn A.28)}$$

where $\bar{\delta}_i$ = approximate value of the film thickness for the present iteration.

$\bar{\delta}_{i-1}$ = approximate value of film thickness from the previous iteration.

$\bar{\delta}'_i$ = approximate value of the derivative of the film thickness for the present iteration

$\bar{\delta}'_{i-1}$ = approximate value of the derivative of the film thickness from the previous iteration.

R = relaxation factor.

These two variables, γ and γ', are in actuality, adjusted approximation of film thickness and derivative of film thickness respectively. This adjustment is required in order to converge to a solution.

The finite element solution is now identical to that of the smooth condenser, that is, the residual is formed, the Galerkin criterion is invoked, and identical 4x4 local matrices are derived. Finally, the equivalent of equation (A.21) is formed.

$$[(\varepsilon\gamma_e{}^4 + \gamma_e{}^3\tan\alpha)[A]_e + (3\varepsilon\gamma_e{}^3\gamma'_e + \gamma_e{}^4\gamma'_e \tan\alpha)[B]_e] \cdot \underline{d}_e$$

$$= (-K_{1e} - K_{2e}\,\gamma_e)\,\underline{F}_e \qquad \text{(eqn A.29)}$$

Notice that this equation has two forcing terms. The additional term ($K_2\gamma$) resulted from the nonlinearization of the problem.

Just an in the smooth condenser film profile solution, the global system given by equation (A.22) is formed, the boundary conditions applied and the system solved for a solution vector $\underline{d}$. The iterative process is continued until convergence is met. With each iteration, γ and γ' are updated and used for the next iteration. When convergence is met, the latest values of d(i), i=1,3,5,...m-1 are the solution values of δ^*.

APPENDIX B

USER'S MANUAL

This appendix describes the data cards required to use the computer code.

The data is divided into "blocks" for convenience. Each page of this user's guide is a separate block. For each block, a general description, the required format, and appropriate comments are provided.

It is imperative that input data be consistent or errors will result. For example, if a smooth geometry is being analyzed, the finite element parameters must also result in a smooth model. In addition, all data fields must be filled with an input value, even if that value is not needed for the analysis. For example, in a smooth analysis, no fin half angle is required for the calculations; however, a value of the correct format must be provided in the fin half angle field or an INPUT/OUTPUT error will result.

DATA BLOCK A

DESCRIPTION: FINITE ELEMENT PARAMETERS

FORMAT: 8I5

1	2	3	4	5	6	7	8
NDIV	NCMREC	NCMTRI	NRWFIN	NRWTRF	NCMTRF	NCOL	NPRNT

FIELD	CONTENTS
1	NDIV---Number of axial increments. Must be less than or equal to 50.
2	NCMREC-Number of columns of finite elements in the rectangular section of fin. May be equal to 0 if fin is triangular only.
3	NCMTRI-Number of columns of finite elements in triangular section of fin. May be equal to 0 if only rectangular fin.
4	NRWFIN-Number of rows of finite elements in the fin section of model. May be equal to 0 if a smooth condenser. Note: If triangular or trapezoidal fin, NRWFIN must equal NCMTRI.
5	NRWTRF-Number of rows of finite elements in wall section of the model.
6	NCMTRF-Number of columns of finite elements in the trough section of finned model. Set equal to 0 if a smooth condenser.
7	NCOL---Total number of columns of finite elements. Must be equal to NCMREC+NCMTRI+NCMTRF for a finned model.
8	NPRNT--Print control number. If equal to 1 correspondence table and major elements of finite element model will be printed out. If equal to 0, output will be suppressed.

DATA BLOCK B

DESCRIPTION: FLUID, FIN MATERIAL SELECTION PARAMETERS

FORMAT: 2I5

```
  1    2    3    4    5    6    7    8
-----------------------------------------
IFLUID  IFIN
-----------------------------------------

-----------------------------------------
```

FIELD	CONTENTS
1	IFLUID-If equal to 0, working fluid is water. If equal to 1, working fluid is freon.
2	IFIN---If equal to 0, condenser wall material is copper. If equal to 1, condenser wall material is stainless steel.

DATA BLOCK C

DESCRIPTION: CONDENSER GEOMETRY

FORMAT: 6G10.5

1	2	3	4	5	6	7	8
CLI	REASEI	THICKI	BFINI	CANGL	FNWTHI		

FIELD	CONTENTS
1	CLI----Condenser length (inches).
2	RBASEI-Inside radius to wall of condenser at condenser end (inches).
3	THICKI-Wall thickness (inches).
4	BFINI--Fin height (inches). Must be set equal to 0.0 if smooth condenser.
5	CANGL--Condenser half angle for tapered condenser (degrees). Must be set equal to 0.0 for cylindrical condenser.
6	FNWTHI-Width of rectangular portion of trapezoidal or rectangular fin (inches). If triangular fin, set equal to 0.0.

DATA BLOCK D

DESCRIPTION: INTERNAL FIN GEOMETRY

FORMAT: 2G10.5

1	2	3	4	5	6	7	8
FANGL	ETOEO						

FIELD	CONTENTS
1	FANGL--Fin half angle (degrees). Set equal to 0.0 for smooth condenser or rectangular fin.
2	ETOEO--Ratio of trough width to fin base width. Determines spacing between fins.

DATA BLOCK E

DESCRIPTION: CONVERGENCE CRITERIA

FORMAT: 2G10.5

1	2	3	4	5	6	7	8
CRIT	CRITDL						

FIELD	CONTENTS
1	CRIT---Temperature convergence criterion. Used to determine solution of two dimensional steady state heat conduction problem.
2	CRITDL-Mass flow convergence criterion. Used only in cylindrical condenser analysis for mass flow convergence test.

DATA BLOCK F

DESCRIPTION: OPERATING PARAMETERS

FORMAT: 5G10.5

1	2	3	4	5	6	7	8
RPM	HINF	TINTL	TSAT	TINF			

FIELD	CONTENTS
1	RPM----Rotational speed (revolutions per minute).
2	HINF---External heat transfer coefficient (Btu/hr-ft -deg F).
3	TINTL--Initial temperature estimate (degrees F).
4	TSAT---Saturation temperature (degrees F).
5	TINF---Ambient temperature (degrees F).

DATA BLOCK G

DESCRIPTION: OUTPUT PRINT CONTROL

FORMAT: 4I5

```
    1       2       3       4       5       6       7       8
--------------------------------------------------------------
IUNITS  NFLAG1  NFLAG2  NFLAG3
--------------------------------------------------------------

--------------------------------------------------------------
```

<u>FIELD</u> <u>CONTENTS</u>

1 IUNITS-Output units control number.
If IUNITS = 0, calculated results will be provided in English units.
If IUNITS = 1, input parameters will be repeated in SI units and calculated results will be provided in SI units.
If IUNITS = 2, input parameters will be repeated in SI units and output results will be provided in both English and SI units.

2 NFLAG1-The first axial increment at which the parameters listed under remarks will be provided as output.

3 NFLAG2-The final increment at which the parameters listed under remarks will be provided.

4 NFLAG3-The step change in increments between NFLAG1.

<u>REMARKS</u>

1) No matter what value of IUNITS is used, input parameters will always first appear in English units.
2) For increments indicated by NFLAG values, the following parameters will appear: a) x-coordinate, y-coordinate and temperature at each nodal point, b) length of element, heat transfer coefficient and heat rate per unit length for each convective boundary element.
3) As a minimum, the values of 1, 1, 1 must be provided as input for NFLAG1, NFLAG2, and NFLAG3 respectively.

DATA BLOCK H

DESCRIPTION: TAPERED CONDENSER SOLUTION METHOD

FORMAT: 1I5

```
    1       2       3       4       5       6       7       8
-------------------------------------------------------------
NSOLVE
-------------------------------------------------------------

-------------------------------------------------------------
```

FIELD	CONTENTS
1	NSOLVE-Tapered solution control number. For tapered, axially finned condenser, set NSOLVE=1. For tapered smooth condenser, NSOLVE must be set to one of the following three values: Set NSOLVE = 2 if solution of film thickness is to be based on Ballback's [Ref. 1] equation. Set NSOLVE = 3 if solution of film thickness is to be based on Daniels and Al-Jumaily [Ref. 13] equation, neglecting drag terms. Set NSOLVE = 4 if solution of film thickness is to be based on Daniel's and Al-Jumaily [Ref. 13] equation, with drag effects included.

REMARKS

1) NSOLVE is only used in tapered condenser analysis.

DATA BLOCK I

DESCRIPTION: CYLINDRICAL CONDENSER ANALYSIS PARAMETERS

FORMAT: 3I5,2G15.10

1	2	3	4	5	6	7	8
NONCE	ITERMX	ITRPRT	RELAX	DELMAX			

FIELD **CONTENTS**

1 NONCE--Single iteration parameter.
If NONCE = 1, only one iteration will be permitted.
If NONCE = 0, iterations will continue until convergence or maximum number of iterations is reached.

2 ITERMX-Maximum number of iterations permitted in analysis.

3 ITRPRT-Iteration print control parameter.
If ITRPRT = 1, mass flow convergence test results will be provided for each iteration.
If ITRPRT = 0, mass flow convergence test results will only be provided on final iteration.

4 RELAX--Relaxation variable used in finite element solution of cylindrical finned film profile.

5 DELMAX-Initial estimate of maximum film thickness used in solution of cylindrical finned film profile.

REMARKS

1) Above parameters are only used in cylindrical condenser analysis.
2) Recommended value of RELAX is 0.80. It is sometimes necessary to adjust this value plus or minus 0.05 to reach film profile convergence at small film thickness values.
3) Input value of DELMAX is only used in cylindrical finned analysis. For the cylindrical smooth condenser, DELMAX is internally generated.

APPENDIX C

SOURCE CODE LISTING

```
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION DIFF(200),TI(200),TJ(200),T1(200),TAVG(100),A(200,50),F(
     1200,1),TPLT(025,100),XPLT(025,100),YPLT(025,100),DLNGTH(50)
      COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,ETOEO,FANGL,HINF,QBTOT,RBASEI,R
     1PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
     1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
     2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
     3(100,50),QFMTOT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTO
     4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
     5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TTROF(100),THICK
     6,UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRT,NBSF
     7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
     8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
      COMMON/CEL1/RELAX,DELMAX,ITRPRT
      COMMON/RECT/FNWDTH,NFNTIP
C
C*********************** INPUT MODE ***********************************
C
C        PRINT HEADER FOR FINITE ELEMENT PARAMETERS
C
      WRITE (6,740)
C
C        INPUT MAJOR FINITE ELEMENT PARAMETERS
C
C     NDIV---------NUMBER OF INCREMENTS ALONG CONDENSER LENGTH
C     NCMREC-------NUMBER OF COLUMNS IN RECTANGULAR PORTION OF FIN
C                  (MAY BE EQUAL TO ZERO IF ONLY TRIANGULAR FIN)
C     NCMTRI-------NUMBER OF COLUMNS IN THE TRIANGULAR PORTION OF FIN
C                  (MAY BE EQUAL TO ZERO IF ONLY RECTANGULAR FIN)
C                  THE GEOMETRY IS SUCH THAT THE NUMBER OF COLUMNS IN
C                  THE TRIANGULAR PORTION OF THE FIN WILL BE EQUAL
C                  TO THE NUMBER OF ROWS IN THE FIN SECTION
C     NRWFIN-------NUMBER OF ROWS IN THE FIN SECTION-GEOMETRY IS SUCH
C                  THAT THE NUMBER OF ROWS IN THE RECTAGULAR SECTION
C                  WILL BE THE SAME AS THE NUMBER OF ROWS IN THE TRI-
C                  ANGULAR SECTION OF FIN
C     NRWTRF-------NUMBER OF ROWS IN THE TROUGH SECTION
C     NCMTRF-------NUMBER OF COLUMNS IN THE TROUGH SECTION
C     NCOL---------TOTAL NUMBER OF COLUMNS IN THE FINITE ELEMENT MODEL
C     NPRNT--------FINITE ELEMENT PRINT CONTROL NUMBER
C                  IF NPRNT=0,ONLY FINITE ELEMENT PARAMETERS LISTED A-
C                  BOVE WILL BE PROVIDED IN OUTPUT
C                  IF NPRNT=1,NOT ONLY ABOVE PARMETERS, BUT ALL ELEMENTS
C                  WITH CORRESPONDING NODAL POINTS AS WELL AS MAJOR ELE-
C                  MENT AND NODAL POINT NUMBERS WILL BE LISTED-USEFUL IN
C                  TROUBLESHOOTING
C
```

```
C
C     NOTE:FINITE ELEMENT GEOMETRY IS SUCH THAT NUMBER OF COLUMNS
C          IN THE TRIANGULAR FIN SECTION IS EQUAL TO THE NUMBER OF ROWS
C          IN THE TRIANGULAR FIN SECTION
C
      READ (5,650) NDIV,NCMREC,NCMTRI,NRWFIN,NRWTRF,NCMTRF,NCOL,NPRNT
      WRITE (6,750) NDIV,NCMREC,NCMTRI,NRWFIN,NRWTRF,NCMTRF,NCOL
C
C           INPUT WORKING FLUID AND FIN MATERIAL SELECTION CONSTANT
C
C     IFLUID--------0 IF WORKING FLUID IS WATER
C                   1 IF WORKING FLUID IS FREON
C     IFIN----------0 IF FIN MATERIAL IS COPPER
C                   1 IF FIN MATERIAL IS STAINLESS STEEL
C
      READ (5,660) IFLUID,IFIN
      IF (IFLUID.EQ.1) GO TO 20
      IF (IFIN.EQ.1) GO TO 10
      WRITE (6,760)
      GO TO 40
   10 WRITE (6,770)
      GO TO 40
   20 IF (IFIN.EQ.1) GO TO 30
      WRITE (6,780)
      GO TO 40
   30 WRITE (6,790)
   40 CONTINUE
C
C
C             PRINT HEADER FOR INPUT VARIABLES
C
      WRITE (6,800)
C
C
C             INPUT CONDENSER GEOMETRY
C
C     CLI-----------CONDENSER LENGTH(INCHES)
C     RBASEI--------INSIDE RADIUS OF CONDENSER AT BASE OF FIN(INCHES)
C     THICKI--------CONDENSER WALL THICKNESS AT TROUGH(INCHES)
C     BFINI---------HEIGHT OF FIN(INCHES)
C     CANGL---------CONE HALF ANGLE(DEGREES)
C     FNWTHI-------WIDTH OF RECTANGULAR PORTION OF FIN IF FIN IS
C                  TRAPEZOIDAL OR RECTANGULAR(INCHES
C
      READ (5,670) CLI,RBASEI,THICKI,BFINI,CANGL,FNWTHI
      WRITE (6,810) CLI,RBASEI,THICKI,BFINI,FNWTHI,CANGL
C
C
```

```
C
C          INPLT INTERNAL FIN GEOMETRY
C
C     FANGL---------FIN HALF ANGLE(DEGREES)
C     ETOEO---------RATIO OF TROUGH WIDTH TO FIN BASE WIDTH
C
      READ (5,680) FANGL,ETOEO
      WRITE (6,820) FANGL,ETOEO
C
C
C        INPUT CONVERGENCE CRITERION
C
C     CRIT----------CONVERGENCE CRITERION
C     CRITDL--------MASS FLOW CONVERGENCE CRITERION
C
      READ (5,690) CRIT,CRITDL
      WRITE (6,830) CRIT,CRITDL
C
C
C          INPUT TEMPERATURES,ROTATIONAL SPEED,AND EXTERNAL HEAT TRANS-
C          FER CCEFFICIENT
C
C     TINTL---------INITIAL TEMPERATURE ESTIMATE(DEGREES F)
C     TSAT----------SATURATION TEMPERATURE OF WORKING FLUID(DEGREES F)
C     TINF----------EXTERNAL AMBIENT TEMPERATURE(DEGREES F)
C     HINF----------EXTERNAL HEAT TRANSFER COEFFICIENT(BTU/HR-FT2-F)
C     RPM-----------ROTATIONAL SPEED(REVOLUTIONS PER MINUTE)
C
      READ (5,700) RPM,HINF,TINTL,TSAT,TINF
      WRITE (6,840) RPM,TINTL,TSAT,TINF,HINF
C
C
C          READ IN UNIT DETERMINATION VARIABLE(UNITS) AND OUTPUT FLAGS
C
C        IF IUNITS = 2 INPUT AND OUTPUT WILL BE IN BOTH ENGLISH AND
C                    SI UNITS
C                  = 1 INPUT WILL BE REPEATED IN SI UNITS
C                    AND OUTPUT WILL BE IN SI UNITS
C                  = 0 OUTPUT WILL BE IN ENGLISH UNITS
C        NFLAG1------FIRST INCREMENT AT WHICH NODAL POINT COORDINATES,
C                    TEMPERATURES AND CONVECTIVE BOUNDARY HEAT TRANSFER
C                    PARAMETERS WILL BE OUTPUTTED
C        NFLAG2------FINAL INCREMENT AT WHICH NODAL POINT COORDINATES,
C                    TEMPEARATURES AND CONVECTIVE BOUNDARY HEAT TRANSFER
C                    PARAMETERS WILL BE OUTPUTTED
C        NFLAG3------CHANGE IN INCREMENTS BETWEEN INTERMEDIATE INCRE-
C                    MENTS
C
      READ (5,710) IUNITS,NFLAG1,NFLAG2,NFLAG3
```

```
C
C
C     INPUT SCLUTION METHOD VARIABLE FOR TAPERED SMOOTH CASE
C
C     NSOLVE-------SOLUTION METHOD VARIABLE
C                  NSOLVE=1 IF FINNED TAPERED HEAT PIPE
C                  NSOLVE=2 BALLBACK'S EQUATION FOR FILM THICKNESS IS
C                           USED
C                  NSOLVE=3 DANIEL'S AND AL-JUMAILY EQUATION, WITHOUT
C                        DRAG IS USED TO DETERMINE FILM THICKNESS
C                  NSOLVE=4 DANIEL'S AND AL-JUMAILY EQUATION, WITH DRAG
C                        IS USED TO DETERMINE FILM THICKNESS
C
      READ (5,720)NSOLVE
C
C
C     INPUT CYLINDRICAL ANALYSIS PARAMETERS
C
C     NONCE-------IF EQUAL TO 1, ONLY ONE ITERATION WILL BE ACCOMPLISHED
C                 IF EQUAL TO 0, ITERATIONS WILL CONTINUE UNTIL MASS
C                 FLOW RATE CONVERGENCE IS REACHED
C     ITERMX------MAXIMUM NUMBER OF MASS FLOW ITERATIONS PERMITTED
C     ITRPRT------IF EQUAL TO 1, RESULTS OF MASS FLOW TEST WILL BE
C                 PRINTED FOR EACH ITERATION
C                 IF EQUAL TO 0,RESULTS OF MASS FLOW TEST WILL BE PRINT-
C                 ED ONLY ON FINAL ITERATION
C     RELAX-------RELAXATION VARIABLE USED TO ADJUST APPROXIMATE SOLU-
C                 TION OF FILM THICKNESS IN FINITE ELEMENT SOLUTION OF
C                 FILM THICKNESS PROFILE
C     DELMAX------MAXIMUM FILM THICKNESS INITIAL ESTIMATE. WILL BE AD-
C                 AUTOMATICALLY ADJUSTED AFTER FIRST ITERATION. INPUT
C                 VALUE IS ONLY USED IN CYLINDRICAL FINNED CONDENSER
C                 ANALYSIS. FOR SMOOTH CYLINDRICAL CONDENSER ANALYSIS,
C                 DELMAX IS A CALCULATED VALUE.
C
      READ(5,730)NONCE,ITERMX,ITRPRT,RELAX,DELMAX
C
C************************** EXECUTION MODE **************************
C
C           *****     BEGIN EXECUTION MODE     *****
C
C
C
C           ESTABLISH CORRESPONDENCE BETWEEN NODAL POINTS AND ELEMENTS
C           AND DEFINE OTHER FINITE ELEMENT PARAMETERS
C
      CALL CORRES(NPRNT,NBAN,NEXTM,NFINM)
C
C
```

```
C
C     CONVERT UNITS OF ALL DIMENSIONAL PARAMETERS FROM INCHES TO FEET
C     CONVERT UNITS OF ANGLES FROM DEGREES TO RADIANS.CALCULATE ADDI-
C     TIONAL CONDENSER GEOMETRIC VARIABLES.
C
C     CL-----------CONDENSER LENGTH(FEET)
C     RBASE--------INSIDE RADIUS OF CONDENSER AT BASE (FEET)
C     R2-----------AVERAGE CONDENSER RADIUS(FEET)
C     BFIN---------FIN HEIGHT(FEET)
C     FNWDTH-------WIDTH OF RECTANGULAR PORTION OF FIN(FEET)
C                  NOTE:MAY BE EQUAL TO ZERO IF ONLY TRIANGULAR FIN
C     PHI----------CONE HALF ANGLE OF CONDENSER (RADIANS)
C     SPHI---------SINE OF PHI
C     CPHI---------COSINE OF PHI
C     TPHI---------TANGENT OF PHI
C     DELX---------INCREMENT WIDTH(FEET)
C     CBASE--------BASE CIRCUMFERENCE(FEET)
C     REXIT--------CONDENSER RADIUS AT EXIT(EVAPORATOR SECTION)(FEET)
C     CEXIT--------BASE CIRCUMFERENCE AT EXIT(FEET)
C     THICK--------CONDENSER WALL THICKNESS AT TROUGH(FEET)
C     ALFA---------FIN HALF ANGLE(RADIANS)
C     SALFA--------SINE OF ALFA
C     CALFA--------COSINE OF ALFA
C     TALFA--------TANGENT OF ALFA
C     EZERO--------WIDTH OF FIN BASE(FEET)
C     EPSO---------TROUGH WIDTH(FEET)
C     ZFIN---------NUMBER OF FINS
C     SURFAR-------SURFACE AREA OF FIN AND TROUGH PER UNIT LENGTH(FT2)
C     EPSEX--------TROUGH WIDTH AT EXIT(FEET)
C     BETA---------CHANGE IN TROUGH WIDTH PER INCREMENTAL LENGTH
C     ZZERO--------FIN SURFACE LENGTH FROM APEX TO BASE(FEET)
C     AFOVAS-------RATIO OF SURFACE AREA WITH FIN PER UNIT LENGTH TO
C                  SURFACE AREA WITHOUT FIN PER UNIT LENGTH
C     BOA----------RATIO OF FIN HEIGHT TO BASE OF FIN(COTANGENT ALFA)
C     OMEGA--------ANGULAR VELOCITY(RADIANS PER HOUR)
C     SECLNG-------IF NO EXTENDED SURFACE, THE LENGTH OF THE SECTION OF
C                  INTEREST(FEET)
C
C     FOLLOWING CYLINDRICAL ANALYSIS CONTROL PARAMETERS ARE ALSO
C     INITIALIZED
C     ITER---------NUMBER OF MASS FLOW ITERATIONS
C     NDEL---------IS SET EQUAL TO 1 AFTER TEMPERATURE CONVERGENCE
C                  IS REACHED AT ALL INCREMENTS. IT ALLOWS FILM PROFILE
C                  SOLUTION TO ACCOUNT FOR TEMPERATURE VARIATION AXIALLY
C     NDELFN-------IS SET EQUAL TO 1 AFTER FINAL FILM PROFILE OF A MASS
C                  FLOW ITERATION IS DETERMINED PRIOR TO MASS FLOW CON-
C                  VERGENCE TEST.
C     NSTOP--------IS SET EQUAL TO 1 TO STOP MASS FLOW ITERATION
```

```fortran
C
C
      CL=CLI/12.0D0
      RBASE=RBASEI/12.0D0
      BFIN=BFINI/12.0D0
      FNWDTH=FNWTHI/12.0D0
      R2=RBASE-BFIN/2.0D0
      PI=3.14159265358979D0
      PHI=2.0D0*CANGL*PI/360.0D0
      SPHI=DSIN(PHI)
      CPHI=DCOS(PHI)
      TPHI=DTAN(PHI)
      DIV=DFLOAT(NDIV)
      DELX=CL/DIV
      CBASE=2.0D0*PI*RBASE
      SECLNG=CBASE/360.0D0
      REXIT=RBASE+CL*TPHI
      CEXIT=2.0D0*PI*REXIT
      THICK=THICKI/12.0D0
      ALFA=FANGL*2.0D0*PI/360.0D0
      SALFA=DSIN(ALFA)
      CALFA=DCOS(ALFA)
      TALFA=DTAN(ALFA)
      OMEGA=RPM*2.0D0*PI*60.0D0
      ITER=1
      NDEL=0
      NDELFN=0
      IF (BFIN.NE.0) GO TO 60
      BETA=((CEXIT-CBASE)/360.0D0)/DIV
      SURFAR=CBASE
      AFOVAS=1.0D0
      BOA=0.0D0
      GO TO 90
60    IF(NCMREC.EQ.0)GO TO 70
      EZERO=FNWDTH+2.0D0*BFIN*TALFA
      EPSO=ETOEO*EZERO
      ZFIN=CBASE/(EZERO+EPSO)
      SURFAR=ZFIN*(FNWDTH+(2.0D0*(BFIN/CALFA)+EPSO))
      AFOVAS=(FNWDTH+2.0D0*(BFIN/CALFA)+EPSO)/(EZERO*(1.0D0+ETOEO))
      BOA=BFIN/(EZERO/2.0D0)
      GO TO 80
70    CONTINUE
      EZERO=2.0D0*BFIN*TALFA
      EPSO=ETOEO*EZERO
      ZFIN=CBASE/(EZERO+EPSO)
      SURFAR=ZFIN*(2.0D0*(BFIN/CALFA)+EPSO)
      AFOVAS=(ETOEO+(1./SALFA))/(1.+ETOEO)
80    CONTINUE
```

```
      BOA=BFIN/(EZERO/2.0D0)
      EPSEX=(CEXIT-(ZFIN*EZERO))/ZFIN
      BETA=(EPSEX-EPSO)/DIV
      ZZERO=BFIN/CALFA
90    CONTINUE
C
C
C           TEMPERATURE ESTIMATES ALONG INTERNAL CONVECTIVE BOUNDARY AND
C              AVERAGE TEMPERATURES
C
C     T------------TEMPERATURE AT A NODAL POINT(DEG F)
C     TSOLID-------AVERAGE TEMPERATURE OF SOLID SECTION (DEG F)
C
      DO 100 IGT=NENTIP,NENTRF
      NP=ICOR(IGT,2)
      T(NP)=TINTL
100   CONTINUE
      NP=ICOR(NENTRF,1)
      T(NP)=TINTL
110    CONTINUE
      TSOLID=(TSAT+TINF)/2.0D0
C
C
C     *****      BEGIN MAIN ITERATIVE LOOP      *****
C
C     QFNTOT------TOTAL HEAT RATE INTO THE FIN SECTIONS(BTU/HR)
C     QTFTOT------TOTAL HEAT RATE INTO TROUGH SECTIONS(BTU/HR)
C     QBTOT-------TOTAL HEAT RATE OUT FROM BOTTOM OF ALL SECTIONS(BTU/HR
C     QSMTOT------TOTAL HEAT RATE INTO SMOOTH SECTION(BTU/HR)-NO EXTEND-
C                 ED SURFACE
C     DMTOT-------TOTAL CONDENSATE MASS FLOW RATE (LBM/HR)
C     R-----------MINIMUM RADIUS FOR A GIVEN INCREMENTAL SECTION(FEET)
C     SLNGTH------LENGTH OF SMOOTH SECTION(FEET)
C     EPS---------TROUGH WIDTH FOR A GIVEN INCREMENTAL SECTION(FEET)
C     DLNGTH------DISTANCE FROM CONDENSER END (X=0) TO MIDPOINT OF
C                 INCREMENT.
C
C     NOTE:ALL CALCUALTIONS ARE FOR THE MIDPOINT OF AN INCREMENT
C
C
      QFNTOT=0.0D0
      QTFTOT=0.0D0
      QBTOT=0.0D0
      QSMTOT=0.0D0
      DMTOT=0.0D0
      DMDOT=0.0D0
      DO 400 NI=1,NDIV
      IF (NI.GT.1) GO TO 120
```

```
      R(NI)=R2+DELX*SPHI/2.0D0
      EPS(NI)=EPSC+BETA/2.0D0
      SLNGTH(NI)=SECLNG+BETA/2.0D0
      DLNGTH(NI)=DELX/2.0D0
      GO TO 130
120   CONTINUE
      NI1=NI-1
      R(NI)=R(NI1)+DELX*SPHI
      EPS(NI)=EPS(NI1)+BETA
      SLNGTH(NI)=SLNGTH(NI1)+BETA
      DLNGTH(NI)=DLNGTH(NI1)+DELX
130   CONTINUE
C
C
C   ******      TEMPERATURE DISTRIBUTION ALONG FIN      ******
C         NODAL POINT COORDINATES
C
C     NOTE: SYMMETRY BOUNDARY IS VERTICAL FROM BOTTOM EDGE OF SECTION
C           (I.E. OUTSIDE WALL OF CONDENSER) TO APEX OF FIN BISECTING
C           FIN INTO TWO EQUAL PARTS IF FINNED CONDENSER.
C           IF SMOOTH CONDENSER, SYMMETRIC SECTIO IS DEGREE ARC OF
C           ROTATION OF CIRCUMFERENCE, I.E. 360 EQUAL SECTIONS.
C
C     X------------COORDINATE AXIS PERPENDICULAR TO SYMMETRY BOUNDARY
C                  ALONG BOTTOM EDGE OF SECTION
C                  (X=0 AT SYMMETRY BOUNDARY)
C     Y------------COORDINATE AXIS ALONG SYMMETRY BOUNDARY(Y=0 AT BOTTOM
C                  EDGE OF SECTION)
C     Z------------COORDINATE AXIS ALONG FIN SURFACE MEASURED FROM APEX
C                  OF FIN(Z=0 AT APEX)
C     ZB-----------DISTANCE FROM APEX TO LOWER NODAL POINT OF MIDDLE
C                  ELEMENT ALONG CONVECTIVE BOUNDARY OF FIN
C
C        DETERMINE X AND Y COORDINATES OF NODAL POINTS(USED FOR TEMPER-
C        ATURE DISTRIBUTION DETERMINATION)
      CALL COORD
C
C        ESTABLISH CONVERGENCE COUNTER
      IM=1
C
      INP=1
      Z(1)=0.0D0
      DO 140 IZEL=NFNTIP,NBSFIN
      NA=ICOR(IZEL,1)
      NB=ICOR(IZEL,2)
      XE=X(NA)-X(NB)
      YE=Y(NA)-Y(NB)
      ELZ=DSQRT(XE**2+YE**2)
```

```
      Z(INP+1)=Z(INP)+ELZ
      I2=INP+1
      INP=INP+1
140   CONTINUE
      XZB=X(ICOR(NFINM,1))-X(ICOR(NFNTIP,2))
      YZB=Y(ICOR(NFINM,1))-Y(ICOR(NFNTIP,2))
      ZB=DSQRT(XZB**2+YZB**2)
      ZC=ZZERC
C
C        PARABOLIC TEMPERATURE DISTRIBUTICN ALONG FIN CONVECTIVE BOUN-
C        DARY USING LAGRANGE INTERPOLATION
C
C     TP1----------TEMPERATURE AT APEX
C     TP2----------TEMPERATURE AT LOWER NODAL POINT OF MIDDLE FIN CONV-
C                  VECTIVE ELEMENT
C     TP3----------TEMPERATURE AT LOWER NODAL POINT OF ELEMENT AT BASE
C                  OF FIN
C     TC-----------SUM OF NODAL POINT TEMPERATURES FOR I NODAL POINTS
C     TFILM--------AVERAGE FILM TEMPERATURE OF CONDENSATE FILM ON FIN
C
150   CONTINUE
      IF (BFIN.EQ.0.0D0) GO TO 160
      TP1=T(ICOR(NFNTIP,2))
      TP2=T(ICOR(NFINM,1))
      TP3=T(ICOR(NBSFIN,1))
      AP1=TP1/(ZC*ZB)
      AP2=TP2/(ZB*(ZB-ZC))
      AP3=TP3/(ZC*(ZC-ZB))
      BP1=-(ZB+ZC)*AP1
      BP2=-ZC*AP2
      BP3=-ZB*AP3
      AA1=AP1+AP2+AP3
      BB1=BP1+BP2+BP3
C
160   CONTINUE
      TC=0.0D0
      DO 170 NY=NFNTIP,NENTRF
      NZ=ICOR(NY,2)
      TC=TC+T(NZ)
170   CONTINUE
      RNY=DFLOAT(NENTRF+1)
      NTRF1=ICOR(NENTRF,1)
      TFILM=(TC+T(NTRF1)+RNY*TSAT)/(2.0D0*RNY)
C
C
C  *****      SOLID AND FLUID PROPERTIES      ******
C     HFG--------LATENT HEAT OF VAPORIZATION(BTU/LBM)
C     RHOF-------DENSITY OF FLUID(LBM/FT3)
```

```
C       UF---------DYNAMIC VISCOSITY OF LIQUID(LBM/FT-HR)
C       CF---------THERMAL CONDUCTIVITY OF CONDENSATE FILM(BTU/HR-FT-DEGF)
C       CW---------THERMAL CONDUCTIVITY OF CONDENSER WALL(BTU/HR-FT-DEGF)
C       CP---------SPECIFIC HEAT OF FLUID(BTU/LBM-DEG F)
C       RHOV-------DENSITY OF VAPOR(LBM/FT3)
C       UVAP-------DYNAMIC VISCOSITY OF VAPOR(LBM/FT-HR)
C
C           WATER PROPERTIES
C
        IF (IFLUID.EQ.1) GO TO 180
        HFG=1093.88D0-0.57030D0*TSAT+0.00012819D0*(TSAT**2)-0.000000882400*
       1(TSAT**3)
        RHOF(NI)=62.774D0-0.0025569800*TFILM-0.0000535720D0*TFILM**2
        CF(NI)=0.3034D0+0.00073892700*TFILM-0.0000014732100*TFILM**2
        UF(NI)=(0.0013970D0-0.0000146690D0*TFILM+0.00000006312530D0*TFILM**2-
       1.000000000976569D0*TFILM**3)*3600.0D0
        CP(NI)=-0.00000000000007D0*TFILM**3+0.0000014764D0*TFILM**2-0.000276
       &88D0*TFILM+1.0109117D0
        UVAP=(.018607450900+.00007897746800*TSAT-1.9480676E-07*TSAT**2+4.3
       &20809E-10*TSAT**3)
        RHOV=(-.011843082700+.000335368496*TSAT-3.08706926E-06*TSAT**2+1.2
       &265446E-08*TSAT**3)
C
C           FREON PROPERTIES
C
180     IF (IFLUID.EQ.0) GO TO 190
        HFG=69.5459-0.0156011*TSAT-0.000455294*(TSAT**2)+0.00000104144*(TS
       1AT**3)
        RHOF(NI)=102.059-0.025364*TFILM-0.000502649*(TFILM**2)+0.000001354
       107*(TFILM**3)
        CF(NI)=.0871592253-.000795216575*TFILM+6.5849702E-06*TFILM**2-1.85
       &886027E-08*TFILM**3
        UF(NI)=(8.4496827 47E-04-7.85856781E-06*TFILM+4.2075531E-08*TFILM**
       12-9.7346869E-11*TFILM**3)*3600.0D0
        RHOV=.0868201290D0-.0015952335600*TSAT+4.52222798E-05*TSAT**2-1.681
       &1776E-08*TSAT**3
        UVAP=(.022649974210D0+.000019956161*TSAT+1.8031152E-07*TSAT**2-7.53
       &704E-10*TSAT**3)
        CP(NI)=.2106709091+.00016205808*TFILM+3.1628785E-07*TFILM**2-8.838
       &385E-10*TFILM**3
190     CONTINUE
C
C
C       CALCULATE THERMAL CONDUCTIVITY OF WALL MATERIAL
        IF (IFIN.EQ.1) GO TO 200
C           FOR COPPER WALL MATERIAL
C
        CW(NI)=231.7772D0-0.0222D0*TSOLID
```

```
200   IF (IFIN.EQ.0) GO TO 210
C
C        FOR STAINLESS STEEL WALL MATERIAL
C
      CW(NI)=E.776+0.00265*TSOLID
210   CONTINUE
C
C *****  INITIAL FILM THICKNESS   *****
C
      IF(PHI.EQ.0.0D0.AND.NI.NE.1) GO TO 220
      CALL DLSTAR(DLNGTH,RHOV,UVAP,CPHI,DMDOT,TAVERG,NDEL,NDELFN,ITER,IT
     &ERMX,NSTOP,NONCE,TAVG,T1)
      IF(NSTOP.EQ.1) GO TO 410
220   CONTINUE
      IF(NDEL.EQ.1)NDELFN=1
C
C
C  *****   CALCULATE HEAT TRANSFER COEFFICIENTS OF     *****
C          CONVECTIVE BOUNDARY ELEMENT SURFACES
C
      CALL HTCOEF(AA1,BB1,CPHI)
C
C ******   ENTRY INTO FINITE ELEMENT SOLUTION     ******
C
      CALL FORMAF(A,F,NBAN)
      CALL BANDEC (A,F,NSNP,NBAN,1)
C
C
C *****        TEMPERATURE DISTRIBUTION      *****
C     T1-----------TEMPERATURE AT FIN TIP(IF NO EXTENDED SURFACE,THE
C                  TEMPERATURE AT THE INTERNAL LEFT CORNER OF THE UNIT
C                  SECTION
C     TBR----------TEMPERATURE AT EXTERNAL RIGHT CORNER OF SECTION
C     TBL----------TEMPERATURE AT EXTERNAL LEFT CORNER OF SECTION
C     TBM----------TEMPERATURE ON EXTERNAL SURFACE DIRECTLY BELOW BASE
C                  OF FIN,IF NO EXTENDED SURFACE,TEMPERATURE IN MIDDLE
C                  OF EXTERNAL SURFACE
C     TBSFIN-------TEMPERATURE AT THE BASE OF THE FIN,IF NO EXTENDED
C                  SURFACE,TEMPERATURE IN MIDDLE OF INTERNAL SURFACE
C     TTROF--------TEMPERATURE AT THE END OF THE TROUGH,IF NO EXTENDED
C                  SURFACE, TEMPERATUREAT INTERNAL RIGHT CORNER OF
C                  SECTION
C
      DO 230 NT=1,NSNP
      T(NT)=F(NT,1)
230   CONTINUE
      T1(NI)=T(ICOR(NFNTIP,2))
      TBR(NI)=T(ICOR(NEXTRT,2))
```

```
      TBM(NI)=T(ICOR(NEXTM,2))
      TBL(NI)=T(ICOR(NEXTLT,1))
      TTROF(NI)=T(ICOR(NENTRF,1))
      TBSFIN(NI)=T(ICCR(NBSFIN,1))
C
C
C        DETERMINE NEW VALUE OF TSOLID
C
      SUMTMP=C.ODC
      DO 240 NS=1,NSNP
      SUMTMP=SUMTMP+T(NS)
240   CONTINUE
      PN=DFLCAT(NSNP)
      TSOLID=SUMTMP/PN
C
C
C *****    CHECK CONVERGENCE OF TEMPERATURES     ******
      IF (IM.EQ.1) GO TO 260
      DO 250 I=1,NSNP
      TJ(I)=T(I)
250   CONTINUE
      GO TO 28C
260   CONTINUE
      DO 270 I=1,NSNP
      TI(I)=T(I)
270   CONTINUE
      IM=2
      GO TO 150
280   CONTINUE
      DO 290 I=1,NSNP
      DIFF(I)=DABS((TJ(I)-TI(I))/TJ(I))
      IF (DIFF(I).GT.CRIT) GO TO 300
290   CONTINUE
      GO TO 320
300   CONTINUE
      DO 310 I=1,NSNP
      TI(I)=TJ(I)
310   CONTINUE
      GO TO 150
320   CONTINUE
C
C     DEFINE THE AVERAGE WALL TEMPERATURE OF AN INCREMENT
      IF(PHI.NE.0.0D0) GO TO 350
      TAVG(NI)=T(ICOR(NENTRF,1))
      IF(NI.NE.NDIV) GO TO 340
      TAVSUM=C.ODC
      DO 330 NR=1,NDIV
         TAVSUM=TAVSUM+TAVG(NR)
```

```
330   CONTINUE
      TAVERG=TAVSUM/DFLOAT(NDIV)
340   CONTINUE
350   CONTINUE
      IF(PHI.NE.0.0D0)NDEL=1
C
C  ******     PERFORM HEAT TRANSFER CALCULATIONS      *****
      CALL HTCALC
C
C
C *****   CALCULATE INCREMENTAL AND TOTAL MASS FLOW RATES    ******
C     DMDOT--------CONDENSATE MASS FLOW RATE FOR A GIVEN INCREMENT
C                  (LBM/HR)
C     AMTOT--------CONDENSATE MASS FLOW RATES FOR NI INCREMENTS(LBM/HR)
C     DMTOT--------TOTAL CONDENSATE MASS FLOW RATE(LBM/HR)
C
      IF(BFIN.EQ.0.0D0)GO TO 360
      DMDOT=2.0D0*QBI*DELX/HFG
      DMTOT=DMTOT+DMDOT
      AMTOT(NI)=ZFIN*DMTOT
      IF(NI.EQ.NDIV)DMTOT=DMTOT*ZFIN
      GO TO 370
360   CONTINUE
      DMDOT=QBI*DELX/HFG
      DMTOT=DMTOT+DMDOT
      AMTOT(NI)=360.0D0*DMTOT
      IF(NI.EQ.NDIV)DMTOT=DMTOT*360.0D0
370   CONTINUE
C
C
C *****  DETERMINE NEXT INCREMENTAL TROUGH THICKNESS(DELTA STAR)    *****
C
C     NOTE:DLSTAR CALLED HERE TO DETERMINE FILM THICKNESS FOR NEXT IN-
C          CREMENT IN TAPERED ANALYIS. IN CYLINDRICAL ANALYSIS,DLSTAR
C          CALLED HERE FOR MASS FLOW CONVERGENCE TEST.
C
      IF (PHI.EQ.0.0D0.AND.NI.NE.NDIV.OR.NDEL.NE.1)GO TO 380
      CALL DLSTAR(DLNGTH,RHOV,UVAP,CPHI,DMDOT,TAVERG,NDEL,NDELFN,ITER,IT
     &ERMX,NSTOP,NONCE,TAVG,T1)
380   CONTINUE
C
C     STORE COORDINATES AND TEMPERATURES OF EACH INCREMENT
      DO 390 I=1,NSNP
      XPLT(NI,I)=X(I)
      YPLT(NI,I)=Y(I)
      TPLT(NI,I)=T(I)
390   CONTINUE
400   CONTINUE
```

```
      IF(PHI.NE.0.0D0)NSTOP=1
      IF(NSTOP.EQ.1) GO TO 410
      IF(NDEL.EQ.0.AND.NDELFN.EQ.1) GO TO 410
      TINTL=TAVERG
      NDEL=1
      GO TO 90
  410 CONTINUE
      NDELFN=0
      IF(NSTOP.EQ.0)GO TO 90
C
C        ********************      OUTPUT MODE      **********************
      IF (IUNITS.EQ.1) GO TO 620
C
C
      WRITE (6,850)
C
C             OUTPUT FIN GEOMETRY PARAMETERS
C
      WRITE (6,860) ZFIN,FANGL,SURFAR,ETOEO,BCA,AFOVAS
C
C             OUTPUT LATENT HEAT OF VAPORIZATION FOR A GIVEN TSAT
C
      WRITE (6,870) HFG,TSAT
C
C
      IF(BFIN.EQ.0.0D0.AND.NSOLVE.EQ.4)WRITE(6,880)UVAP,RHOV
      IF (BFIN.EQ.0.0D0) GO TO 430
C             OUTPUT HEAT RATE INTO TROUGH,HEAT RATE INTO FIN,TOTAL HEAT
C             RATE IN AND HEAT RATE OUT FOR A SINGLE SECTION FOR EACH
C             INCREMENT
C
      WRITE (6,890)
      DO 420 NR=1,NDIV
      WRITE (6,900) NR,QINC(NR),QTINC(NR),QTOTAL(NR),QBINC(NR)
  420 CONTINUE
      GO TO 470
  430 CONTINUE
      IF(PHI.EQ.0)GO TO 460
      IF (NSOLVE.NE.2) GO TO 440
      WRITE(6,910)
      GO TO 460
  440 IF (NSOLVE.NE.3) GO TO 450
      WRITE(6,920)
      GO TO 460
  450 CONTINUE
      WRITE(6,930)
  460 CONTINUE
      WRITE (6,940)
```

```
      DO 470 NR=1,NDIV
      WRITE(6,950)NR,CINCSM(NR),QBINC(NR),QX(NR)
470   CONTINUE
C
C             OUTPUT HEAT RATE INTO FIN, HEAT RATE INTO TROUGH, AND HEAT
C             RATE OUT BOTTOM AND TOTAL MASS FLOW RATE FOR A GIVEN SET
C             OF INPUT CONDITIONS
      IF (BFIN.EQ.0.0D0) GO TO 490
C
      IF(PHI.EQ.0.0D0) GO TO 480
      WRITE (6,970) QFNTOT,QTFTOT,QBTOT,DMTOT
      GO TO 510
480   CONTINUE
      WRITE(6,960)QFNTOT,QTFTOT,QBTOT,DMTOT,FLOMAS
      GO TO 510
490   IF(PHI.EQ.0.0D0)GO TO 500
      WRITE(6,980)QSMTOT,QBTOT,DMTOT
      GO TO 510
500   CONTINUE
      WRITE (6,990) QSMTOT,QBTOT,DMTOT,FLOMAS
510   CONTINUE
C
C
C             OUTPUT INCREMENTALLY VARYING PROPERTIES
C
      WRITE (6,005)
      DO 520 NR=1,NDIV
      WRITE (6,015) NR,CF(NR),CW(NR),UF(NR),RHOF(NR)
520   CONTINUE
C
C             OUTPUT INCREMENATALLY VARYING PARAMETERS
C
      IF (BFIN.EQ.0.0D0) GO TO 540
      WRITE (6,025)
      DO 530 NR=1,NDIV
      WRITE (6,035) NR,DELSTR(NR),EPS(NR),R(NR),AMTOT(NR)
530   CONTINUE
      GO TO 560
540   CONTINUE
      WRITE (6,045)
      DO 550 NR=1,NDIV
      WRITE (6,055) NR,DELSTR(NR),SLNGTH(NR),R(NR),AMTOT(NR)
550   CONTINUE
560   CONTINUE
C
C         OUTPUT MAJOR TEMPERATURES FOR EACH INCREMENT
C
      IF(BFIN.EQ.0.0D0)GO TO 570
```

```
      WRITE (6,065)
      GO TO 580
570   CONTINUE
      WRITE (6,075)
580   CONTINUE
      DO 590 NR=1,NDIV
      WRITE (6,085) NR,TBL(NR),TBM(NR),TBR(NR),T1(NR),TBSFIN(NR),TTROF(N
     &R)
590   CONTINUE
C
C        OUTPUT NODAL POINT X AND Y COORDINATES AND FINAL TEMPER-
C        TURE FOR EACH NODAL POINTFOR INCREMENTS OF INTEREST
C
      WRITE (6,095)
      DO 620 I=NFLAG1,NFLAG2,NFLAG3
      WRITE (6,105) I
      WRITE (6,115)
      DO 600 NP=1,NSNP
      WRITE (6,125) NP,XPLT(I,NP),YPLT(I,NP),TPLT(I,NP)
600   CONTINUE
C
C        OUTPUT CONVECTIVE BOUNDARY ELEMENT LENGTH,HEAT TRANSFER
C        COEFFICIENT AND HEAT RATE PER UNIT LENGTH FOR INCREMENTS OF
C        INTEREST
C
      WRITE (6,135)
      DO 610 IQEL=1,NEXTLT
      WRITE (6,900) IQEL,ELMNT(I,IQEL),HELMNT(I,IQEL),QELMNT(I,IQEL)
610   CONTINUE
620   CONTINUE
C
      IF (IUNITS.EQ.0) GO TO 640
630   CONTINUE
C
C        UNITS CONVERT ALL DIMENSIONAL QUANTITIES,INPUT AND CALCULATED,
C        TO SI UNITS AND OUTPUTS THESE QUANTITIES
C
      CALL SIUNIT(T1,TPLT,XPLT,YPLT,NFLAG1,NFLAG2,NFLAG3)
C
640   CONTINUE
C
C
C
      STOP
C     ******      INPUT FORMAT      *****
650   FORMAT (8I5)
660   FORMAT (2I5)
670   FORMAT (6G10.5)
```

```
680    FORMAT (2G10.5)
690    FORMAT (2G10.5)
700    FORMAT (5G10.5)
710    FORMAT (4I5)
720    FORMAT (1I5)
730    FORMAT(3I5,2G15.10)
C
C
C
C      *****     OUTPUT FORMAT     *****
740    FORMAT (30X,25HFINITE ELEMENT PARAMETERS,///)
750    FORMAT (1X,21HNUMBER OF INCREMENTS=,1X,I5,/01X,48HNUMBER OF COLUMN
      1S IN RECTANGULAR SECTION OF FIN=,01X,I5,/01X,47HNUMBER OF COLUMNS
      2IN TRIANGULAR SECTION OF FIN=,01X,I5,/01X,26HNUMBER OF ROWS IN THE
      3 FIN=,01X,I5,/01X,37HNUMBER OF ROWS IN THE TROUGH SECTION=,01X,I5,
      4/01X,40HNUMBER OF COLUMNS IN THE TROUGH SECTION=,01X,I5,/01X,24HTO
      5TAL NUMBER OF COLUMNS=,01X,I5,///)
760    FORMAT (1X,67HHEAT PIPE ANALYSIS FOR COPPER HEAT PIPE WITH WATER A
      1S WORKING FLUID////)
770    FORMAT (1X,63HHEAT PIPE ANALYSIS FOR STAINLESS STEEL HEAT PIPE WIT
      1H WATER AS ,/1X,13HWORKING FLUID////)
780    FORMAT (1X,67HHEAT PIPE ANALYSIS FOR COPPER HEAT PIPE WITH FREON A
      1S WORKING FLUID////)
790    FORMAT (1X,62HHEAT PIPE ANALYSIS FOR STAINLESS STEEL HEAT PIPE WIT
      1H FREON AS,/1X,13HWORKING FLUID////)
800    FORMAT (30X,16HINPUT PARAMETERS,/12X,53HALL DIMENSIONAL QUANTITIES
      1 ARE GIVEN IN ENGLISH UNITS,///)
810    FORMAT (1X,18HCONDENSER LENGTH =,5X,G10.5,3X,6HINCHES,/1X,12HBASE
      1RADIUS=,12X,G10.5,2X,7HINCHES ,/1X,16HWALL THICKNESS =,8X,G10.5,1X
      2,7H INCHES,/1X,12HFIN HEIGHT =,12X,G10.5,1X,7H INCHES,/1X,'RECTANG
      3ULAR PORTION OF FIN =',1X,G10.5,2X,'INCHES',/1X,,22HCONDENSER HALF
      4 ANGLE =,1X,G10.5,2X,8H DEGREES//)
820    FORMAT (10X,14HFIN PARAMETERS,/1X,16HFIN HALF ANGLE =,G10.5,8H DEG
      1REES,/1X,38HRATIO OF TROUGH WIDTH TO BASE OF FIN =,G10.5,//)
830    FORMAT (1X,'TEMPERATURE COVERGENCE CRITERION =',G10.5,/1X,'MASS FL
      &OW CONVERGENCE CRITERION =',G10.5,//1X,'NOTE:MASS FLOW CONVERGENCE
      & TEST IS ONLY USED IN CYLINDRICAL HEAT PIPE ANALYSIS',//)
840    FORMAT (10X,20HOPERATING PARAMETERS,/1X,5HRPM =,10X,G10.5,23H REVO
      1LUTIONS PER MINUTE,/1X,30HINITIAL TEMPERATURE ESTIMATE =,2X,G10.5,
      210H DEGREES F,/1X,24HSATURATION TEMPERATURE =,8X,G10.5,10H DEGREES
      3 F,/1X,22HEXTERNAL TEMPERATURE =,10X,G10.5,1X,9HDEGREES F,/,1X,36H
      4EXTERNAL HEAT TRANSFER COEFFICIENT =,G10.5,21H BTU/HR-FT2-DEGREES
      5F,//)
850    FORMAT (1H ,////25X,18HCALCULATED RESULTS,/23X,24HRESULTS IN ENGLI
      1SH UNITS,///)
860    FORMAT(10X,'FIN GEOMETRY PARAMETERS',/,1X,'NR FINS =',5X,G10.5,/1X
      &,'FIN HALF ANGLE',5X,G10.5,5X,'DEGREES'/,1X,'SURFACE AREA PER UNIT
      & LENGTH =',3X,G10.5,1X,'FEET2/FEET',/1X,'RATIO OF TROUGH WIDTH TO
```

```
     &FIN BASE WIDTH =',01X,G10.5,/1X,'RATIO OF FIN HEIGHT TO FIN BASE =
     &',1X,G10.5,/,1X,'RATIO OF SURFACE AREA TO SMOOTH AREA =',01X,G10.5
     &,//)
870   FORMAT (1H ,1X,29HLATENT HEAT OF VAPORIZATION =,G10.5,8H BTU/LBM,/
     11X,20HFOR SATURATION TEMP=,G10.5,9HDEGREES F,//)
880   FORMAT(1X,'VAPOR VISCOSITY AND DENSITY USED IN DRAG SOLUTION TERMS
     &',//1X,'VAPOR VISCOSITY =',2X,F15.10,2X,'LBM/FT-HR',/1X,'VAPOR DEN
     &SITY =',4X,F15.10,2X,'LBM/FT3',//)
890   FORMAT (1H ,15X,36HHEAT RATE SUMMARY FOR A UNIT SECTION,//1X,10HIN
     1CREMENT ,3X,10HHEAT RATE ,5X,10HHEAT RATE ,6X,9HHEAT RATE,5X,9HHEA
     2T RATE,/15X,8HINTO FIN,6X,11HINTO TROUGH,6X,8HTOTAL IN,6X,10HOUT B
     3OTTOM,/15X,8H(BTU/HR),7X,8H(BTU/HR),8X,8H(BTU/HR),6X,8H(BTU/HR),/)
900   FORMAT (1X,I5,10X,G10.5,3(5X,G10.5)/)
910   FORMAT(1X,'RESULTS BASED ON BALLBACK EQUATION FOR TROUGH THICKNESS
     &',///)
920   FORMAT(1X,'RESULTS BASED ON DANIELS  EQUATION FOR TROUGH THICKNESS
     &',/1X,'SIMPLIFIED FORM-FUNCTION OF SHERWOOD NUMBER ONLY',///)
930   FORMAT(1X,'RESULTS BASED ON DANIELS  EQUATION FOR TROUGH THICKNESS
     &',/1X,'FULL FORM, INCLUDES DRAG TERMS',///)
940   FORMAT (1H ,15X,36HHEAT RATE SUMMARY FOR A UNIT SECTION,//1X,10HIN
     1CREMENT ,3X,10HHEAT RATE ,5X,10HHEAT RATE ,8X,'TOTAL HEAT RATE ',/
     212X,12HINTO SECTION,4X,14HOUT OF SECTION,5X,'FOR ALL SECTIONS',/14
     3X,8H(BTU/HR),7X,8H(BTU/HR),9X,'IN INCREMENT',///)
950   FORMAT(1X,I5,10X,G10.5,5X,G10.5,9X,G10.5)
960   FORMAT (1H ,1X,25HTOTAL HEAT RATE INTO FINS,9X,F10.3,7H BTU/HR,/1X
     1,27HTOTAL HEAT RATE INTO TROUGH,8X,F10.3,7H BTU/HR,/1X,32HTOTAL HE
     2AT RATE OUT OF HEAT PIPE,3X,F10.3,7H BTU/HR,///1X,'TOTAL MASS FLOW
     3 RATE BASED ON HEAT RATE DIVIDED BY HFG',F10.3,3X,'LBM/HR',/1X,'TO
     4TAL MASS FLOW RATE AT OVERFALL',F10.3,3X,'LBM/HR',//)
970   FORMAT (1H ,1X,25HTOTAL HEAT RATE INTO FINS,9X,F10.3,7H BTU/HR,/1X
     1,27HTOTAL HEAT RATE INTO TROUGH,8X,F10.3,7H BTU/HR,/1X,32HTOTAL HE
     2AT RATE OUT OF HEAT PIPE,3X,F10.3,7H BTU/HR,///1X,20HTOTAL MASS FL
     3OW RATE,3X,G10.5,7H LBM/HR,//)
980   FORMAT (1H ,1X,31HTOTAL HEAT RATE INTO HEAT PIPE ,3X,F10.3,7H BTU/
     1HR,///1X,32HTOTAL HEAT RATE OUT OF HEAT PIPE,3X,F10.3,7H BTU/HR,//
     2/1X,'TOTAL MASS FLOW RATE BASED ON HEAT RATE DIVIDED BY HFG',F10.3
     3,3X,'LBM/HR',//)
990   FORMAT (1H ,1X,31HTOTAL HEAT RATE INTO HEAT PIPE ,3X,F10.3,7H BTU/
     1HR,///1X,32HTOTAL HEAT RATE OUT OF HEAT PIPE,3X,F10.3,7H BTU/HR,//
     2/1X,'TOTAL MASS FLOW RATE BASED ON HEAT RATE DIVIDED BY HFG',F10.3
     3,3X,'LBM/HR',/1X,'TOTAL MASS FLOW RATE AT OVERFALL',F10.3,3X,'LBM/
     4HR',/)
005   FORMAT (1H ,10X,43HFLUID AND MATERIAL PROPERTIES PER INCREMENT,/1X
     1,9HINCREMENT,7X,5HFLUID,15X,4HWALL,15X,9HVISCOSITY,12X,7HDENSITY/1
     25X,12HCONDUCTIVITY,7X,12HCONDUCTIVITY,/14X,15HBTU/HR-FT-DEG F,4X,1
     35HBTU/HR-FT-DEG F,08X,9HLBM/FT-HR,12X,7HLBM/FT3//)
015   FORMAT (1X,I5,4(10X,G10.5))
025   FORMAT (1H1,10X,32HVARIOUS PARAMETERS PER INCREMENT,/1X,10HINCREME
```

```
      1NT ,6X,8HDEL STAR,10X,12HTROUGH WIDTH,8X,14HMINIMUM RADIUS,6X,14HM
      2ASS FLOW RATE,/18X,4HFEET,16X,4HFEET,16X,4HFEET,16X,6HLBM/HR,/)
035   FORMAT (1X,I5,4(10X,G10.5))
045   FORMAT (//,10X,32HVARIOUS PARAMETERS PER INCREMENT,/1X,10HINCREME
      1NT ,6X,8HDEL STAR,10X,14HSECTION LENGTH,6X,14HMINIMUM RADIUS,6X,14
      2HMASS FLOW RATE,/18X,4HFEET,16X,4HFEET,14X,4HFEET,16X,6HLBM/HR,/)
055   FORMAT (1X,I5,4(10X,G10.5))
065   FORMAT(//,17X,'MAJOR TEMPERATURES PER UNIT SECTION FOR ALL INCREME
      &NTS',/,7X,'EXTERNAL',07X,'EXTERNAL',07X,'EXTERNAL',07X,'INTERNAL',
      &6X,'INTERNAL',7X,'INTERNAL'/,8X,'LEFT',08X,'BELOW BASE',08X,'RIGHT
      &',09X,'FIN TIP',07X,'FIN BASE',07X,'TROUGH END',/,7X,'DEGREES F',0
      &6X,'DEGREES F',06X,'DEGREES F',06X,'DEGREES F',05X,'DEGREES F',06X
      &,'DEGREES F',//)
075   FORMAT(//,17X,'MAJOR TEMPERATURES PER UNIT SECTION FOR ALL INCREME
      &NTS',/,7X,'EXTERNAL',07X,'EXTERNAL',07X,'EXTERNAL',07X,'INTERNAL',
      &6X,'INTERNAL',7X,'INTERNAL'/,8X,'LEFT',08X,'  MIDDLE  ',08X,'RIGHT
      &',09X,' LEFT  ',07X,' MIDDLE ',07X,'   RIGHT  ',/,7X,'DEGREES F',0
      &6X,'DEGREES F',06X,'DEGREES F',06X,'DEGREES F',05X,'DEGREES F',06X
      &,'DEGREES F',//)
085   FORMAT (1X,I2,4(5X,G10.5),2(4X,G10.5))
095   FORMAT (1H0,///,10X,63HNODAL POINT COORDINATES AND TEMPERATURE AT
      1A SPECIFIC INCREMENT,/,10X,58HHEAT TRANSFER COEFFICIENT,ELEMENT LE
      2NGTH AND HEAT RATE AT ,/,10X,55HCONVECTIVE BOUNDARY ELEMENTS AT TH
      3IS SPECIFIC INCREMENT,///)
105   FORMAT (1H0,10X,17HINCREMENT NUMBER=,I5,//)
115   FORMAT (10X,39HNODAL POINT COORDINATES AND TEMPERATURE,/,1X,11HNOD
      1AL POINT,5X,8HX-COORD ,09X,7HY-COORD,9X,11HTEMPERATURE,/18X,4HFEET
      2,13X,4HFEET,12X,9HDEGREES F,//)
125   FORMAT (1X,I5,4X,3(07X,G10.5))
135   FORMAT (1H0,10X,38HCONVECTIVE BOUNDARY ELEMENT PARAMETERS,//01X,11
      1HELEMENT NR.,10X,6HLENGTH,10X,13HHEAT TRANSFER,04X,25HHEAT RATE PE
      2R UNIT LENGTH,/23X,4HFEET,12X,11HCOEFFICIENT,13X,9HBTU/HR-FT,/39X,
      312HBTU/HR-FT2-F,//)
145   FORMAT (01X,I5,3(12X,G10.5))
      END
C               SUBROUTINE CORRES ESTABLISHES CORRESPONDENCE BETWEEN ELE-
C               MENTS AND NODAL POINTS AND DEFINES MAJOR FINITE ELEMENT
C               PARAMETERS
C
      SUBROUTINE CORRES(NPRNT,NBAN,NEXTM,NFINM)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,ETOEO,FANGL,HINF,QBTOT,RBASEI,R
     &PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
     1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
     2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
     3(100,50),QFNTOT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTO
     4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
```

```
     5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TTROF(100),THICK
     6,UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRT,NBSF
     7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
     8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
      COMMON/FECT/FNWDTH,NFNTIP
C
C     NOTE:ELEMENTS ARE NUMBERED IN THER FOLLOWING ORDER:
C          1)STARTING WITH APEX OF FIN
C          2)ALONG INSIDE CONVECTIVE BOUNDARY FROM TOP TO BASE OF FIN
C            AT TROUGH FROM BASE OF FIN TO END OF TROUGH
C          3)ALONG OUTSIDE CONVECTIVE BOUNDARY FROM RIGHT TO LEFT
C          4)ALL OTHERS FROM TOP TO BOTTOM,LEFT TO RIGHT
C
C          NODAL POINTS OF AN ELEMENT ARE NUMBERED IN THE FOLLOWING OR
C          -DER:
C          1)NODAL POINTS ARE NUMBERED COUNTERCLOCKWISE
C          2)ELEMENTS WITH A CONVECTIVE BOUNDARY HAVE ELEMENT NODAL
C            POINTS 1 AND 2 LOCATED ON THE CONVECTIVE BOUNDARY
C          3)ELEMENTS NOT ON A CONVECTIVE BOUNDARY HAVE ELEMENT NODAL
C            POINT NR. 1 IS THE LOWER LEFT NODAL POINT OF THE ELEMENT
C
C     NEL----------NUMBER OF ELEMENTS
C     NSNP---------NUMBER OF SYSTEM NODAL POINTS
C     NBAN---------SYSTEM BANDWIDTH
C     NENTRF-------NUMBER OF THE ELEMENT OF A SECTION, WITH INTERNAL
C                  CONVECTIVE BOUNDARY,LOCATED AT END OF TROUGH
C                  IF THERE IS NO EXTENDED SURFACE,NENTRF IS THE RIGHT
C                  HAND ELEMENT ON THE INTERNAL CONVECTIVE BOUNDARY
C     NBSFIN-------NUMBER OF THE ELEMENT OF A SECTION WITH INTERNAL
C                  CONVECTION BOUNDARY LOCATED AT BASE OF FIN
C                  IF NO EXTENDED SURFACE,NBSFIN IS THE MIDDLE ELEMENT
C                  ON THE INTERNAL CONVECTIVE BOUNDARY
C     NEXTRT-------NUMBER OF THE ELEMENT WITH EXTERNAL CONVECTIVE
C                  BOUNDARY LOCATED AT RIGHT BOTTOM CORNER OF SECTION
C     NEXTLT-------NUMBER OF THE ELEMENT WITH EXTERNAL CONVECTIVE
C                  BOUNDARY LOCATED AT LEFT BOTTOM CORNER OF SECTION
C     NEXTF--------NEXTLT PLUS 1
C     NEXTM--------NUMBER OF INTERMEDIATE ELEMENT ON BOTTOM WITH SECOND
C                  NODAL POINT LOCATED DIRECTLY BELOW FIRST NODAL POINT
C                  OF ELEMENT AT BASE OF FIN (NBSFIN)
C     NFINM--------NUMBER OF THE INTERMEDIATE ELEMENT OF FIN ON THE IN-
C                  TERNAL CONVECTIVE BOUNDARY WHOSE FIRST NODAL POINT
C                  IS IN THE MIDDLE OR ADJACENT TO THE MIDDLE OF FIN
C                  ALONG THE Z AXIS
C     NFNTIP-------NUMBER OF ELEMENT AT THE TIP OF THE FIN ON THE VERTI-
C                  CAL CONVECTIVE SURFACE.NFNTIP WILL BE EQUAL TO 1 IF
C                  ONLY A TRIANGULAR FIN
C     NRWFIN-------NUMBER OF ROWS IN THE FINSECTION
```

```
C       NCMTRI-------NUMBER OF COLUMNS IN THE TRIANGULAR PORTION OF THE
C                    FIN(MAY BE EQUAL TO 0 IF ONLY RECTANGULAR FIN)
C       NRWTRF-------NUMBER OF ROWS IN THE TROUGH SECTION
C       NCMTRF-------NUMBER OF COLUMNS IN THE TROUGH SECTION
C       NCMREC-------NUMBER OF COLUMNS IN THE RECTANGULAR PORTION OF THE
C                    FIN(MAY BE EQUAL TO 0 IF ONLY A TRIANGULAR FIN)
C       NRFIN--------NUMBER OF ROWS IN THE FIN SECTION MINUS 1
C       NCFIN--------NUMBER OF COLUMNS IN THE TRIANGULAR SECTION OF
C                    FIN PLUS 1
C       NCOL---------TOTAL NUMBER OF COLUMNS IN THE FINITE ELEMENT MODEL
C                    IF AN EXTENDED SURFACE EXISTS, NCOL IS EQUAL TO THE
C                    SUM OF THE NUMBER OF COLUMNS IN THE FIN AND THE
C                    TROUGH
C       NPDIFF-------NUMERICAL DIFFERENCE BETWEEN TWO ADJACENT VERTICAL
C                    SYSTEM NODAL POINTS IN TROUGH SECTION
C       NPFSYM-------SYSTEM NODAL POINTS LOCATED ON SYMMETRY BOUNDARY
C       NPFCNV-------SYSTEM NODAL POINTS LOCATED ALONG FIN CONVECTIVE
C                    BOUNDARY
C       NPORIG-------SYSTEM NODAL POINT LOCATED AT ORIGIN OF COORDINATE
C                    SYSTEM
C       NPSMBS-------SYSTEM NODAL POINT LOCATED AT JUNCTION OF SYMMETRY
C                    BOUNDARY AND LINE OF INTERSECTION OF BASE OF FIN
C       ICOR---------SYSTEM NODAL POINT CORRESPONDING TO ITH NODAL POINT
C                    OF ELEMENT IEL
C
C
C          IDENTIFY MAJOR ELEMENTS NUMBERS ALONG CONVECTIVE BOUNDARY
C
C
        NEL=2*NRWTRF*NCOL+2*NRWFIN*NCMREC+NCMTRI**2
        NFNTIP=NCMREC+1
        IF (NRWFIN.NE.0) GO TO 10
        NBSFIN=NCOL/2
        IF (DFLOAT(NCOL)/2.0D0.GT.DFLOAT(NCOL/2)) NBSFIN=NBSFIN+1
        NENTRF=NCOL
        GO TO 20
10      CONTINUE
        NBSFIN=NCMREC+NRWFIN
        NENTRF=NBSFIN+NCMTRF
20      CONTINUE
        NEXTRT=NENTRF+1
        NEXTLT=NEXTRT+NCOL-1
        NFINM=NFNTIP+(NBSFIN-NFNTIP)/2
        NEXTM=NEXTRT+(NEXTLT-NEXTRT)/2
C
C          IDENTIFY OTHER MAJOR ELEMENTS THAT ARE USED AS COUNTERS
C
        NRFIN=NRWFIN-1
```

```
      NCFIN=NCMTRI-1
      NCM1=NCMREC+1
      IF (NRWFIN.NE.0) GO TO 30
      NCMREC=NCOL
      GO TO 40
C
30    IF (NCMREC.EQ.0) GO TO 90
40    CONTINUE
C
      DO 50 I=1,NCMREC
      ICOR(I,1)=I+1
      ICOR(I,2)=I
      ICOR(I,3)=I+1+NCMREC
50    CONTINUE
      IF (NRWFIN.EQ.0) GO TO 120
C
      IF (NCMTRI.EQ.0) GO TO 70
      NFN1ST=NCMREC+1
      DO 60 I=NFN1ST,NBSFIN
      ICOR(I,2)=ICOR(I-1,1)
      ICOR(I,1)=ICOR(I,2)+I+1
      ICOR(I,3)=ICOR(I,1)-1
60    CONTINUE
      GO TO 100
70    CONTINUE
C
      NEXT=NCMREC+1
      DO 80 I=NEXT,NBSFIN
      ICOR(I,2)=ICOR(I-1,1)
      ICOR(I,1)=ICOR(I,2)+NCMREC+1
      ICOR(I,3)=ICOR(I,1)-1
80    CONTINUE
      GO TO 100
90    CONTINUE
      J=1
      JJ=3
      DO 100 I=1,NRWFIN
      ICOR(I,1)=JJ
      ICOR(I,2)=J
      ICOR(I,3)=JJ-1
      J=JJ
      JJ=JJ+I+2
100   CONTINUE
C
      NN=NBSFIN+1
      JJJ=ICOR(NBSFIN,1)+1
      DO 110 IJ=NN,NENTRF
      ICOR(IJ,1)=JJJ
```

```
      ICOR(IJ,2)=JJJ-1
      ICOR(IJ,3)=ICOR(IJ,1)+NCOL
      JJJ=JJJ+1
110   CONTINUE
C
120   CONTINUE
      NJ=NCCL+1
      IF (NRWFIN.NE.0) GO TO 130
      JJ=(NRWTRF+1)*NJ
      GO TO 140
130   CONTINUE
      JJ=JJJ+NRWTRF*NJ-1
140   CONTINUE
      J=JJ-1
      DO 150 IJ=NEXTRT,NEXTLT
      ICOR(IJ,1)=J
      ICOR(IJ,2)=JJ
      ICOR(IJ,3)=JJ-NJ
      JJ=JJ-1
      J=J-1
150   CONTINUE
C
C
      IF (NCMREC.EQ.0) GC TO 190
C
      II=ICOR(1,3)
      III=ICCR(1,1)
      JJ=NEXTLT+1
      JJJ=JJ+NCMREC-1
      IF (NCMTRI.EQ.0) JJJ=JJJ-1
      DO 160 IJ=JJ,JJJ
      ICOR(IJ,1)=II
      ICOR(IJ,2)=II+1
      ICOR(IJ,3)=III
      II=II+1
      III=III+1
160   CONTINUE
      IJK=JJJ
      IF (NRWFIN.EQ.0) GO TO 240
C
C
      IF (NRWFIN.EQ.1) GO TO 210
C
      NRREC=NRWFIN-1
      IJK=JJJ
      II=ICOR(1,3)
      DO 180 I=1,NRREC
      IJ=NCMREC+I
```

```
      IF (NCMTRI.EQ.0) IJ=NCMREC
      DO 170 J=1,IJ
      IJK=IJK+1
      ICOR(IJK,3)=II+J-1
      ICOR(IJK,2)=ICOR(IJK,3)+1
      ICOR(IJK,1)=ICOR(IJK,2)+IJ
      IF (NCMTRI.EQ.0.AND.J.EQ.NCMREC) GO TO 170
      IJK=IJK+1
      ICOR(IJK,1)=ICOR(IJK-1,1)
      ICOR(IJK,2)=ICOR(IJK,1)+1
      ICOR(IJK,3)=ICOR(IJK-1,2)
170   CONTINUE
      II=II+IJ+1
180   CONTINUE
      JJ=IJK
      GO TO 210
C
190   CONTINUE
      NN=NRWFIN-1
      II=4
      JJ=NEXTLT
      DO 210 I=1,NN
      DO 200 J=1,I
      JJ=JJ+1
      ICOR(JJ,1)=II+J-1
      ICOR(JJ,2)=ICOR(JJ,1)-I
      ICOR(JJ,3)=ICOR(JJ,2)-1
      JJ=JJ+1
      ICOR(JJ,1)=II+J-1
      ICOR(JJ,2)=ICOR(JJ,1)+1
      ICOR(JJ,3)=ICOR(JJ-1,2)
200   CONTINUE
      II=II+I+2
210   CONTINUE
C
      IF (NCMREC.NE.0) GO TO 220
      II=NRWFIN*(NRWFIN+1)/2+(NCOL+2)
      GO TO 240
220   IF (NCMTRI.NE.0) GO TO 230
      II=NRWFIN*(NCMREC+1)+(NCOL+2)
      GO TO 240
230   CONTINUE
      II=(NRWFIN*NCMREC)+NRWFIN*(NRWFIN+1)/2+(NCOL+2)
240   CONTINUE
      IF (NRWFIN.NE.0) GO TO 250
      II=ICOR(1,3)
      JJ=NEXTLT
      NCMREC=0
```

```
250   CONTINUE
      IF (NCMREC.NE.0) JJ=IJK
      JJ=JJ+1
      NI=NRWFIN+1
      NF=NRWFIN+NRWTRF
      DO 290 I=NI,NF
      DO 280 J=1,NCOL
      IF (I.EQ.NI.AND.J.GT.(NCMTRI+NCMREC)) GO TO 260
      ICOR(JJ,1)=II
      ICOR(JJ,2)=II-NCOL
      ICOR(JJ,3)=ICOR(JJ,2)-1
      JJ=JJ+1
260   CONTINUE
      IF (I.EQ.NF) GO TO 270
      ICOR(JJ,1)=II
      ICOR(JJ,2)=II+1
      ICOR(JJ,3)=II-NCOL
      JJ=JJ+1
270   II=II+1
280   CONTINUE
      II=II+1
290   CONTINUE
      IF (NRWFIN.EQ.0) GO TO 360
C
C
C        DETERMINE NODAL POINTS OF FIN ON SYMMETRY BOUNDARY(NPFSYM) AND
C        NODAL POINT AT INTERSECTION OF BASE OF FIN AND SYMMETRY BOUN-
C        DARY(NPSMBS)
C
C
      NPFSYM(1)=ICOR(1,3)
      IF (NRWFIN.EQ.1) GO TO 360
      DO 320 KI=2,NRWFIN
      IF(NCMTRI.EQ.0) GO TO 300
      NPFSYM(KI)=NPFSYM(KI-1)+NCMREC+KI
      GO TO 310
300   CONTINUE
      NPFSYM(KI)=NPFSYM(KI-1)+NCMREC+1
310   CONTINUE
320   CONTINUE
      NPSMBS=NPFSYM(NRWFIN)
C
C
C        DETERMINE NODAL POINTS OF FIN ON CONVECTIVE BOUNDARY (NPFCNV)
C        AND NODAL POINT AT ORIGIN(NPORIG)
C
      NPFCNV(1)=ICOR(NCM1,1)
      DO 350 KKI=2,NRWFIN
```

```
      IF(NCMTRI.EC.0) GO TO 330
      NPFCNV(KKI)=NPFCNV(KKI-1)+NCMREC+KKI+1
      GO TO 340
330   CONTINUE
      NPFCNV(KKI)=NPFCNV(KKI-1)+NCMREC+1
340   CONTINUE
350   CONTINUE
C
360   CONTINUE
C
      NPORIG=ICOR(NEXTLT,1)
      NPDIFF=(ICOR(NEXTRT,2)-ICOR(NENTRF,1))/NRWTRF
C
C
      NSNP=1
      NBAN=1
      IF(NPRNT.EQ.1)WRITE (6,390)
C
C        PRINT ELEMENTS AND CORRESPONDING NODAL PCINTS
C
C
      DO 370 IK=1,NEL
      IF(NPRNT.EQ.1)WRITE(6,400) IK,ICOR(IK,1),ICOR(IK,2),ICOR(IK,3)
      NSNP=AMAXO(ICOR(IK,1),ICOR(IK,2),ICOR(IK,3),NSNP)
      II=IABS(ICOR(IK,1)-ICOR(IK,2))
      JJ=IABS(ICOR(IK,2)-ICOR(IK,3))
      KK=IABS(ICOR(IK,1)-ICOR(IK,3))
      NBAN=MAXO(NBAN,II,JJ,KK)
370   CONTINUE
      NBAN=NBAN+1
C
C        PRINT TOTAL NUMBER OF SYSTEM NODAL PCINTS
C
      IF(NPRNT.NE.1)GO TO 380
      WRITE (6,410) NSNP,NBAN
C
C        PRINT MAJOR ELEMENT NUMBERS
      WRITE (6,420) NBSFIN,NENTRF,NEXTRT,NEXTLT
C
C
C        PRINT NODAL POINTS FOR FOLLOWING LOCATIONS:
C            1)BOTTOM RIGHT CORNER
C            2)BOTTOM MIDDLE
C            3)BOTTOM LEFT CORNER
C            4)END OF TROUGH
C            5)BASE OF FIN
C
      WRITE (6,430) ICOR(NEXTRT,2),ICOR(NEXTM,2),ICOR(NEXTLT,1),ICOR(NEN
```

```
     1TRF,1),ICOR(NBSFIN,1)
380   CONTINUE
C
C
      RETURN
390   FORMAT (4X,11HELEMENT NR.,4X,17HNODAL PCINT NR. 1,4X,17HNODAL POIN
     1T NR. 2,4X,17HNODAL POINT NR. 3//)
400   FORMAT(6X,I3,3(18X,I3),/)
410   FORMAT (1X,30HNUMBER OF SYSTEM NODAL POINTS=,I3,/,1X,17HSYSTEM BAN
     1DWIDTH=,13X,I3,//)
420   FORMAT (05X,21HMAJOR ELEMENT NUMBERS,/1X,12HNOMENCLATURE,6X,12HELE
     1MENT  NR.,/1X,9HFIN BASE ,10X,I5,/1X,14HEND OF TROUGH ,5X,I5,/1X,1
     22HBOTTOM RIGHT,7X,I5,/1X,11HBOTTOM LEFT,8X,I5,//)
430   FORMAT (05X,39HMAJOR TEMPERATURE LOCATION NODAL POINT ,/1X,8HLOCAT
     1ION,10X,11HNODAL POINT,/1X,12HBOTTOM RIGHT,7X,I5,/1X,17HBOTTOM BEL
     2OW BASE,2X,I5,/1X,11HBOTTOM LEFT,8X,I5,/1X,13HEND OF TROUGH,6X,I5,
     3/1X,10HFIN BASE  ,9X,I5,//)
      END
C        SUBROUTINE COORD ESTABLISHES X AND Y COORDINATES FOR NODAL
C        POINTS
C
      SUBROUTINE COORD
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,ETOEO,FANGL,HINF,QBTOT,RBASEI,R
     &PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
     1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
     2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
     3(100,50),QFNTOT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTO
     4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
     5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TTROF(100),THICK
     6,UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRT,NBSF
     7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
     8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
      COMMON/FECT/FNWDTH,NFNTIP
C
      IF (NRWFIN.NE.0) DELH=BFIN/DFLOAT(NRWFIN)
      X(1)=0.0D0
      Y(1)=THICK+BFIN
      IF (NRWFIN.EQ.0) GO TO 180
C
      IF (NCMREC.EQ.0) GO TO 60
      DELTAX=FNWDTH/(DFLOAT(NCMREC)*2.0D0)
      NCREC1=NCMREC+1
      DO 10 I=2,NCREC1
      X(I)=X(I-1)+DELTAX
      Y(I)=Y(1)
10    CONTINUE
```

```
C
C
      IF (NRWFIN.EQ.1) GO TO 80
C
      DO 50 I=1,NRFIN
      ICA=NPFSYM(I)
      ICB=NPFCNV(I)
C
      X(ICA)=0.0D0
      Y(ICA)=Y(1)-DFLOAT(I)*DELH
      NCOUNT=1
C
      ICC=ICA+1
      DO 40 II=ICC,ICB
      IF (NCOUNT.GT.NCMREC) GO TO 20
      X(II)=X(II-1)+DELTAX
      GO TO 30
20    CONTINUE
      X(II)=X(II-1)+DELH*TALFA
30    CONTINUE
      Y(II)=Y(ICA)
      NCOUNT=NCOUNT+1
40    CONTINUE
50    CONTINUE
      GO TO 80
C
60    CONTINUE
C
      DO 80 I=1,NRFIN
      ICA1=NPFSYM(I)
      ICB1=NPFCNV(I)
      X(ICA1)=0.0D0
      Y(ICA1)=Y(1)-DFLOAT(I)*DELH
      ICC1=ICA1+1
      DO 70 IIJ=ICC1,ICB1
      X(IIJ)=X(IIJ-1)+DELH*TALFA
      Y(IIJ)=Y(ICA1)
70    CONTINUE
C
80    CONTINUE
C
      AN=0.0D0
      ICD=NCMREC+NCMTRI
      DO 130 J=NPSMBS,NPORIG,NPDIFF
      X(J)=X(1)
      Y(J)=(1.0D0-AN/DFLOAT(NRWTRF))*THICK
      IF (NCMREC.EQ.0) GO TO 140
C
```

```
C
      DO 110 JJ=1,ICD
      IF (JJ.GT.NCMREC) GO TO 90
      X(J+JJ)=X(J+JJ-1)+DELTAX
      GO TO 100
90    CONTINUE
      X(J+JJ)=X(J+JJ-1)+(BFIN*TALFA)/DFLOAT(NCMTRI)
100   CONTINUE
      Y(J+JJ)=Y(J)
110   CONTINUE
C
      DO 120 K=1,NCMTRF
      X(J+JJ-1+K)=X(J+JJ-1)+DFLOAT(K)*EPS(NI)/(2.0D0*DFLOAT(NCMTRF))
      Y(J+JJ-1+K)=Y(J)
120   CONTINUE
      AN=AN+1.0D0
130   CONTINUE
      GO TO 170
140   CONTINUE
      CBA=DFLOAT(ICB1-ICA1)
      AN=0.0D0
      ICD1=ICB1-ICA1+1
      DO 170 J=NPSMBS,NPORIG,NPDIFF
      X(J)=X(1)
      Y(J)=(1.0D0-AN/DFLOAT(NRWTRF))*THICK
      DO 150 JJ=1,ICD1
      X(J+JJ)=X(J)+JJ*EZERO/(2.0D0*(CBA+1.0D0))
      Y(J+JJ)=Y(J)
150   CONTINUE
      JJ=ICD
      DO 160 K=1,NCMTRF
      X(J+JJ+K)=X(J+JJ)+DFLOAT(K)*EPS(NI)/(2.0D0*DFLOAT(NCMTRF))
      Y(J+JJ+K)=Y(J)
160   CONTINUE
      AN=AN+1.0D0
170   CONTINUE
      GO TO 200
C
C        DETERMINE GEOMETRY FOR A SMOOTH INTERNAL SURFACE
C
C
180   CONTINUE
      DELTAX=SLNGTH(NI)/DFLOAT(NCOL)
      DELY=THICK/DFLOAT(NRWTRF)
      NCOL1=NCOL+1
      NROW1=NRWTRF+1
      NROW=NROW1-1
      JJ=0
```

```
      DO 200 I=1,NROW1
      DO 190 J=2,NCOL1
      X(J+JJ)=X(J+JJ-1)+DELTAX
      Y(J+JJ)=Y(J+JJ-1)
190   CONTINUE
      JJ=I*NCOL1
      IF (I.EQ.NROW1) GO TO 200
      X(JJ+1)=X(1)
      Y(JJ+1)=Y(JJ)-DELY
      IF(I.EQ.NROW)Y(JJ+1)=0.0D0
200   CONTINUE
      RETURN
      END
C        SUBROUTINE DLSTAR DETERMINES THE FILM THICKNESS ON THE  SMOOTH
C        SURFACE OF THE HEAT PIPE
C
C
      SUBROUTINE DLSTAR(DLNGTH,RHOV,UVAP,CPHI,DMDOT,TAVERG,NDEL,NDELFN,I
     &TER,ITERMX,NSTOP,NONCE,TAVG,T1)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION XCOF(5),ROOTR(4),ROOTI(4),COF(5),HFGPRM(100),REYN(100)
      DIMENSION VEL(100),TAVG(100),T1(100),DLNGTH(100)
      COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,ETOEO,FANGL,HINF,QBTOT,RBASEI,R
     &PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
     1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
     2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
     3(100,50),QFNTOT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTO
     4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
     5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TTROF(100),THICK
     6,UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRT,NBSF
     7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
     8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
      COMMON/DEL1/RELAX,DELMAX,ITRPRT
C
C     HFGPRM------CORRECTED LATENT HEAT OF VAPORIZATION
C     SHER--------SHERWOOD NUMBER
C
      HFGPRM(NI)=HFG+0.35D0*CP(NI)*(TSAT-T(ICOR(NENTRF,2)))
      SHER=(RHOF(NI)**2*(OMEGA**2*R(NI)-(32.174*3600.0**2))*SPHI*
     %HFGPRM(NI)*DLNGTH(NI)**3)/(4.0D0*UF(NI)*CF(NI)*(TSAT-T(ICOR(NENTRF
     %,2))))
      IF (PHI.EQ.0.0D0) GO TO 200
C
C     DELSTR-------CONDENSATE THICKNESS IN TROUGH
C
C
C     THIS PORTION OF DLSTAR DETERMINES TROUGH FILM THICKNESS FOR TRUNC-
C     ATED HEAT PIPE
```

```
      IF(NDEL.EQ.1) GO TO 10
      IF (NI.NE.1) GO TO 440
      DELSTR(1)=1.107*(DABS((TSAT-TINF)*CF(NI)/(UF(NI)*HFG))**.25)*(DABS
     1(UF(NI)/(RHOF(NI)*OMEGA))**0.5)
      GO TO 440
10    CONTINUE
C
C
C     DELTA STAR IS SMALLEST POSITIVE REAL ROOT OF FOURTH DEGREE POLY-
C     NOMIAL EQUATION IN THE CASE OF THE FINNED TAPERED HEAT PIPE
C     IF NSOLVE IS EQUAL TO 1, THIS ROOT WILL BE FOUND
C
C     A POLYNOMIAL ROOTFINDER ROUTINE IS USED TO FIND ROOTS
C     XCOF---------COEFFICIENT OF POLYNOMIAL
C     C1-----------CONSTANT OF TWO COEFICIENTS
      IF(BFIN.NE.0.0D0)GO TO 20
      IF (NSOLVE.NE.1) GO TO 90
20    CONTINUE
      C1=RHOF(NI)**2*OMEGA**2*R(NI)/(3.0D0*UF(NI))*SPHI
      XCOF(1)=-DMTOT
      XCOF(2)=0.0D0
      XCOF(3)=0.0D0
      XCOF(4)=C1*EPS(NI)
      XCOF(5)=C1*TALFA
      M=4
      IF(CALFA.EQ.1.0D0)M=3
      CALL DPOLRT (XCOF,COF,M,ROOTR,ROOTI,IER)
      IF (NI.GT.1) GO TO 30
C     WRITE (6,150)
30    CONTINUE
C     WRITE (6,160) (ROOTR(I),I=1,4)
      IF (ROOTR(1).GT.0.0D0) GO TO 40
      IF (ROOTR(2).GT.0.0D0) GO TO 50
      IF (ROOTR(3).GT.0.0D0) GO TO 60
      IF (ROOTR(4).GT.0.0D0) GO TO 70
C
C        DEFINE NEW TROUGH CONDENSATE THICKNESS
C
40    DELSTR(NI+1)=ROOTR(1)
      GO TO 80
50    DELSTR(NI+1)=ROOTR(2)
      GO TO 80
60    DELSTR(NI+1)=ROOTR(3)
      GO TO 80
70    DELSTR(NI+1)=ROOTR(4)
80    CONTINUE
      NDEL=0
      GO TO 440
```

```
C
C
C     IF NSOLVE EQUALS 2, BALLBACKS EQUATION IS USED TO DETERMINE DELTA
C     STAR FOR EACH INCREMENT IN THE SMOOTH TAPERED HEAT PIPE ANALYSIS
90    IF(NSOLVE.NE.2)GO TO 100
      A1=(RBASEI/12.0D0)/((RBASEI/12.0D0)+DLNGTH(NI)*SPHI)
      A2=CF(NI)*UF(NI)*(TSAT-T(3))/(RHOF(NI)**2*OMEGA**2*SPHI**2*HFG)
      A22=(3.0D0/2.0D0)*A2
      C=8.0D0/3.0D0
      DELSTR(NI+1)=DABS(A22*(1-DABS(A1)**C))**.25
      NDEL=0
      GO TO 440
C
C
C     IF NSOLVE EQUALS 3,THE DANIELS AND AL-JUMAILY SOLUTION NEGLECTING
C     DRAG IS USED TO SOLVE FOR DELTA STAR AT EACH INCREMENT OF THE
C     SMOOTH TAPERED HEAT PIPE
100   IF (NSOLVE.NE.3) GO TO 110
      DELST=DABS(1.0D0/SHER)**.25
      DELSTR(NI+1)=DELST*DLNGTH(NI)
      NDEL=0
      GO TO 440
C
C
C     IF NSOLVE EQUALS 4, THE DANIELS AND AL-JUMAILY SOLUTION INCLUDING
C     DRAG DUE TO VAPOR FRICTION TO DETERMINE DELTA STAR FOR THE SMOOTH
C     TAPER HEAT PIPE
110   CONTINUE
C     VEL-----------VELOCITY(FT/HR)
C     VELO----------VELOCITY(FT/SEC)
C     REYN----------REYNOLDS NUMBER
C     FRICT---------FRICTION FACTOR
C     TAUV----------SHEAR STRESS VAPOR-LIQUID INTERFACE
C     DRX-----------DRAG NUMBER
C     REVX----------TWO PHASE REYNOLDS NUMBER
C
      VEL(NI)=(3600.0D0*DMTOT/(RHOV*PI*R(NI)**2))
      VELO=VEL(NI)/3600.0D0
      REYN(NI)=(DMTOT*3600.0D0/UVAP)*(4.0D0/(PI*2.0D0*R(NI)))
      IF(REYN(NI).GT.2000.0) GO TO 120
      FRICT=16.0D0/REYN(NI)
      GO TO 130
120   CONTINUE
      FRICT=.0791/(DABS(REYN(NI))**0.25)
130   CONTINUE
      TAUV=0.5D0*FRICT*RHOV*VEL(NI)**2
      DRX=RHOF(NI)*TAUV*HFGPRM(NI)*DLNGTH(NI)**2*CPHI/(UF(NI)*CF(NI)*
     &(TSAT-T(3)))
```

```
      REVX=RHCF(NI)*VEL(NI)*DLNGTH(NI)*CPHI/UF(NI)
      XCOF(1)=-1.CDO
      XCOF(2)=0.0CO
      XCOF(3)=((-1.0DC/4.0DO)*REVX)
      XCOF(4)=(-  .0DO/3.0DO)*DRX
      XCOF(5)=SH
      M=4
C
      CALL DPCLRT (XCCF,COF,M,ROOTR,ROOTI,IER)
      IF (NI.GT.1) GO TO 140
C     WRITE (6,45C)
140   CONTINUE
C     WRITE (6,46C) (ROOTR(I),I=1,4)
      IF (ROCTR(1).GT.0.0DO) GO TO 150
      IF (ROOTR(2).GT.0.0DO) GO TO 160
      IF (ROOTR(3).GT.0.0DO) GO TO 170
      IF (ROCTR(4).GT.0.0DO) GO TO 180
C
C        DEFINE NEW TROUGH CONDENSATE THICKNESS
C
150   DELSTR(NI+1)=ROCTR(1)
      GO TO 190
160   DELSTR(NI+1)=ROCTR(2)
      GO TO 190
170   DELSTR(NI+1)=ROOTR(3)
      GO TO 190
180   DELSTR(NI+1)=ROOTR(4)
190   CONTINUE
      IF(BFIN.EQ.C.ODO.AND.ROOTR(4).GT.ROOTR(3))DELSTR(NI+1)=ROOTR(4)
      DELSTR(NI+1)=DELSTR(NI+1)*DLNGTH(NI)
      NDEL=0
      GO TO 440
C
C        CALCULATE FILM THICKNESS FOR CYLINDRICAL HEAT PIPE
C
200   CONTINUE
C
C
C     SUBROUTINE DELCRV SOLVES FOR FILM PROFILE OF CYLINDRICAL HEAT
C     PIPE. DELCRV WILL DETERMINE THE FILM PROFILE ALONG THE HEAT PIPE
C     AND THE DERIVATIVE AT THE OVERFALL.THE PROGRAM, AS WRITTEN,WILL
C     DETERMINE A NEW PROFILE FOR EACH ITERATION WHEN NI EQUALS 1 UN-
C     TIL TEMPERATURE CONVERGENCE IS REACHED AT NI EQUALS 1.THIS PRO-
C     FILE WILL BE MAINTAIN FOR NI EQUALS 2-NDIV.A NEW PROFILE WILL
C     THEN BE DETERMINED USING THE AVERAGE TEMPERATURE AT EACH ITERA-
C     TION.THIS PROFILE WILL BE MAINTAINED FOR NI EQUALS 1=NDIV UNTIL
C     MASS FLOW RATES ARE COMPARED.IF CONVERGENCE IS NOT MET, THE PRO-
C     CESS WILL START ANEW.
```

```
C      NOTE:NDEL IS A CONTROL NUMBER, IT IS SET EQUAL TO 1 AFTER NI HAS
C      ITERATED TO NDIV.WHEN NDEL IS EQUAL TO ONE, THE AVERAGE TEMPERA-
C      TURE FROM THE FIRST COMPLETED ITERATION IS USED TO DETERMINE THE
C      CONSTANTS IN DELCRV
C      NOTE:NDELFN IS A CONTROL NUMBER, IT IS SET EQUAL TO 1 FOR THE FI-
C      NAL ITERATION PRIOR TO COMPARISON OF MASS FLOW RATES
       IF(NI.GT.1.AND.NI.LT.NDIV) GO TO 440
       IF(NI.EQ.NDIV) GO TO 230
       IF(NDEL.EQ.1) GO TO 210
       TAVERG=T(ICOR(NENTRF,2))
210    CONTINUE
       IF(NDELFN.EQ.1) GO TO 220
C
C      DELMAX-------FILM THICKNESS AT CONDENSER END OF PIPE(X=0.0)
C      DERIV--------DERIVATIVE OF FILM THICKNESS WITH RESPECT TO X AT
C                   OVERFALL
C
       CALL DELCRV(NDIV,NI,ITER,IFLUID,CLI,TSAT,TAVERG,RBASEI,OMEGA,DELMA
     &X,DELSTR,DERIV,TAVG,NDEL,TALFA,CALFA,ZZERO,BFIN,EPS,T,CRITDL
     &,T1,TBSFIN,NTERM)
       IF(NTERM.EQ.1)GO TO 240
220    CONTINUE
       GO TO 440
230    CONTINUE
       IF(NDEL.NE.1)GO TO 440
C
C      OVRDEL-------FILM THICKNESS AT THE OVERFALL
C      FLOMAS-------MASS FLOW RATE DETERMINED AT THE OVERFALL
       OVRDEL=0.25D0*DELMAX
       FLOMAS=((-2.0D0*PI*RHOF(NDIV)**2*OMEGA**2*R(NDIV)**2)/UF(NDIV))*
     &DERIV*(0.25D0*DELMAX)**3/3.0D0
       FLOW1=(RHOF(NDIV)**2*OMEGA**2*R(NDIV))/(3.0D0*UF(NDIV))
       FLOW=FLOW1*(-1.0D0*OVRDEL**2*DERIV)*(EPS(NI)*OVRDEL+OVRDEL**2*
     &TALFA)*ZFIN
       IF(BFIN.NE.0.0D0)FLOMAS=FLOW
C
C
C
C      COMPARE MASS FLOW RATE AT OVERFALL TO CONDENSING MASS FLOW RATE,
C      IF NOT EQUAL, ADJUST DELMAX AND ITERATE AGAIN
       DMFSAV=DELMF
       DELSAV=DELMAX
       DELMF=(DMTOT-FLOMAS)
       DEFM=DABS(DELMF/DMTOT)
       IF(NONCE.EQ.1)GO TO 240
       IF(DEFM.GT.CRITDL) GO TO 250
240    CONTINUE
       NSTOP=1
```

```
      WRITE(6,470)ITER
      WRITE(6,480)ITER,FLOMAS,DELMF,DELSAV,DERIV,GVRDEL
      GO TO 440
250   CONTINUE
      IF(ITER.GT.1) GO TO 260
      DEFMSV=DEFM
      DLMXSV=DELMAX
260   CONTINUE
      IF(DEFM.GE.DEFMSV)GO TO 270
      DEFMSV=DEFM
      DLMXSV=DELMAX
      ITERSV=ITER
270   CONTINUE
      IF(DELMF.GT.0.0D0) GO TO 330
      IF(BFIN.GT.0.0D0)GO TO 290
      IF(DEFM.LT.1.0D0)GO TO 280
      DELMAX=0.965D0*DELMAX
      GO TO 320
280   IF(DEFM.LT.0.10D0) GO TO 290
      DELMAX=0.9825D0*DELMAX
      GO TO 320
290   IF(DEFM.LT.0.05D0) GO TO 300
      DELMAX=0.99125D0*DELMAX
      GO TO 320
300   IF(DEFM.LT.0.025D0) GO TO 310
      DELMAX=0.995625D0*DELMAX
      GO TO 320
310   CONTINUE
      DELMAX=0.99785D0*DELMAX
320   CONTINUE
      GO TO 390
330   CONTINUE
      IF(BFIN.GT.0.0D0)GO TO 350
      IF(DEFM.LT.1.0D0)GO TO 340
      DELMAX=1.020D0*DELMAX
      GO TO 380
340   IF(DEFM.LT.0.10D0) GO TO 350
      DELMAX=1.0150D0*DELMAX
      GO TO 380
350   IF(DEFM.LT.0.05D0)GO TO 360
      DELMAX=1.001875D0*DELMAX
      GO TO 380
360   IF(DEFM.LT.0.025D0) GO TO 370
      DELMAX=1.000937D0*DELMAX
      GO TO 380
370   CONTINUE
      DELMAX=1.00046875D0*DELMAX
380   CONTINUE
```

```
390   CONTINUE
      IF((DELMF.GT.0.0D0.AND.DMFSAV.GT.0.0D0).OR.
     &(DELMF.LT.0.0D0.AND.DMFSAV.LT.0.0D0)) GO TO 420
      IF(DELSAV.GT.DELMAX) GO TO 400
      DELMAX=DELSAV+0.5D0*(DELMAX-DELSAV)
      GO TO 410
400   CONTINUE
      DELMAX=DELMAX+0.5D0*(DELSAV-DELMAX)
410   CONTINUE
420   CONTINUE
      IF(ITRPRT.EQ.0)GO TO 430
      WRITE(6,490)ITER,FLOMAS,DELMF,DELSAV,DERIV,OVRDEL
430   CONTINUE
      ITER=ITER+1
      NSTOP=0
      NDEL=0
      IF(ITER.EQ.ITERMX) NSTOP=1
      IF(ITER.EQ.ITERMX)WRITE(6,500)ITERMX,ITERSV,DEFMSV,DLMXSV
440   CONTINUE
      RETURN
450   FORMAT (1X,29HROOTS OF 4TH ORDER POLYNOMIAL,//6X,6HROOT 1,10X,6HRO
     1OT 2,07X,6HROOT 3,10X,6HROOT 4,/)
460   FORMAT (4G15.7)
470   FORMAT(10X,'CONVERGENCE MET ON ITERATION NUMBER =',I5,/)
480   FORMAT(1X,'ON ITERATION NUMBER',I5,2X,'THE FOLLOWING CONDITIONS EX
     &IST',/1X,'MASS FLOW RATE AT THE OVERFALL =',F15.5,2X,'LBM/HR',/1X,
     &'THE DIFFERENCE BETWEEN THE TWO CALCULATED MASS FLOW RATES IS',
     &F15.8,/1X,'THE MAXIMUM FILM THICKNESS IS=',F15.8,/1X,'THE DERIVATI
     &VE AT THE OVERFALL IS=',G15.8,/1X,'FILM THICKNESS AT OVERFALL=',
     &F15.8,/)
490   FORMAT(1X,'ON ITERATION NUMBER',I5,2X,'THE FOLLOWING CONDITIONS EX
     &IST',/1X,'MASS FLOW RATE AT THE OVERFALL =',F15.5,2X,'LBM/HR',/1X,
     &'THE DIFFERENCE BETWEEN THE TWO CALCULATED MASS FLOW RATES IS',
     &F15.8,/1X,'THE MAXIMUM FILM THICKNESS IS=',F15.8,/1X,'THE DERIVATI
     &VE AT THE OVERFALL IS=',G15.8,/1X,'FILM THICKNESS AT OVERFALL=',
     &F15.8,/)
500   FORMAT(1X,'CONVERGENCE WAS NOT MET AFTER ',I5,2X,'ITERATIONS',/1X,
     &'HOWEVER, ON THE ',I5,2X,'ITERATION THE MINIMUM REALTIVE DIFFERENC
     &E WAS ACHIEVED',/1X,'THIS MINIMUM DIFFERENCE WAS EQUAL TO',D20.12,
     &/1X,'THE DELMAX TO ACHIEVE THIS MINIMUM DIFFERENCE WAS',D20.12,/)
      END
C             SUBROUTINE FORMAF FORMS STIFFNESS(A) AND FORCING(F)
C             MATRICES
C
      SUBROUTINE FORMAF(A,F,NBAN)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION A(200,50),F(200,1),B(3),C(3),EA(3,3)
      COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,ETOEC,FANGL,HINF,QBTOT,RBASEI,R
```

```
     &PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
     1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
     2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
     3(100,50),QFNTOT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTO
     4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
     5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TIROF(100),THICK
     6,UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRT,NBSF
     7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
     8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
      DO 20 N=1,NSNP
      F(N,1)=0.0D0
      DO 10 MA=1,NBAN
      A(N,MA)=0.0D0
10    CONTINUE
20    CONTINUE
      DO 120 IEL=1,NEL
      IA=ICOR(IEL,1)
      IB=ICOR(IEL,2)
      IC=ICOR(IEL,3)
      B(1)=Y(IB)-Y(IC)
      B(2)=Y(IC)-Y(IA)
      B(3)=Y(IA)-Y(IB)
      C(1)=X(IC)-X(IB)
      C(2)=X(IA)-X(IC)
      C(3)=X(IB)-X(IA)
C
C
C        CALCULATE LENGTH DOMAIN (EL)
      EL=DSQRT(C(3)**2+B(3)**2)
C
C        CALCULATE AREA DOMAIN
      AS=DABS((B(1)*C(2)-B(2)*C(1))/2.0D0)
C        ESTABLISH HEAT TRANSFER COEFFICIENT
      IF (IEL.LE.NEXTLT) GO TO 30
      HC=0.0D0
      GO TO 40
30    CONTINUE
      HC=H(IEL)/CW(NI)
40    CONTINUE
C
C
C        FORM STIFFNESS (A) MATRIX
      DO 80 J=1,3
      JJ=ICOR(IEL,J)
      DO 70 K=1,3
      KK=ICOR(IEL,K)
      EA(J,K)=(B(J)*B(K)+C(J)*C(K))/(4*AS)
```

```
      IF (HC.EQ.0.0D0) GO TO 60
      HEL=HC*EL/6.0D0
      IF (J.EQ.3) GO TO 60
      IF (K.EQ.3) GO TO 60
      IF (J.EQ.K) GO TO 50
      EA(J,K)=EA(J,K)+HEL
      GO TO 60
50    CONTINUE
      EA(J,K)=EA(J,K)+2*HEL
60    IF (KK.LT.JJ) GO TO 70
      NW=KK-JJ+1
      A(JJ,NW)=A(JJ,NW)+EA(J,K)
70    CONTINUE
80    CONTINUE
C
C        FORM FORCING MATRIX(F)
C
      IF (IEL.GT.NENTRF) GO TO 90
      FE=HC*TSAT*EL/2.0D0
      GO TO 110
90    IF (IEL.GT.NEXTLT) GO TO 100
      FE=HC*TINF*EL/2.0D0
      GO TO 110
100   CONTINUE
      FE=0.0D0
110   CONTINUE
      F(IA,1)=F(IA,1)+FE
      F(IB,1)=F(IB,1)+FE
120   CONTINUE
      RETURN
      END
C             SUBROUTINE BANDEC SOLVES FOR T MATRIX
C
      SUBROUTINE BANDEC (A,F,NEQ,MAXB,NVEC)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION A(200,50),F(200,1)
      COMMON /GLOB1/ BOA,BFINI,CANGL,CL1,ETOEO,FANGL,HINF,QBTOT,RBASEI,R
     &PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
     1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
     2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
     3(100,50),QFNTOT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTU
     4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
     5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TTROF(100),THICK
     6,UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRT,NBSF
     7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
     8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
      LOOP=NEQ-1
```

```
      DO 20  I=1,LCOP
      MB=I+1
      NB=MINO(I+MAXB-1,NEQ)
      DO 20  J=MB,NB
      L=J+2-MB
      D=A(I,L)/A(I,1)
      DO 10  MM=1,NVEC
10    F(J,MM)=F(J,MM)-D*F(I,MM)
      MM=MINO(MAXB-L+1,NEQ-J+1)
      DO 20  K=1,MM
      NN=L+K-1
20    A(J,K)=A(J,K)-D*A(I,NN)
      DO 30  I=1,NVEC
30    F(NEQ,I)=F(NEQ,I)/A(NEQ,1)
      DO 50  I=2,NEQ
      J=NEQ-I+1
      K=MINO(NEQ-J+1,MAXB)
      DO 50  MM=1,NVEC
      DO 40  L=2,K
      MB=J+L-1
40    F(J,MM)=F(J,MM)-A(J,L)*F(MB,MM)
50    F(J,MM)=F(J,MM)/A(J,1)
      RETURN
      END
C         SUBROUTINE HTCOEF DETERMINES HEAT TRANSFER COEFFICIENTS
C
      SUBROUTINE HTCOEF(AA1,BB1,CPHI)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION H2(5)
      COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,ETDEG,FANGL,HINF,QBTOT,RBASEI,R
     &PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
     1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
     2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
     3(100,50),QFNTOT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTO
     4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
     5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TTROF(100),THICK
     6,UF(100),X(100),Y(100),Z(100),ZZERO,ICCR(200,3),NEXTLT,NEXTRT,NBSF
     7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
     8M(10),NFSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
      COMMON/FECT/NFNTIP
C
C         AVERAGE ELEMENT CONVECTIVE HEAT TRANSFER COEFFICIENT ALONG FIN
C         SURFACE
C     ZSTAR---------DISTANCE ALONG FIN SURFACE FROM APEX TO SURFACE OF
C                   TROUGH CONDENSATE
C     DELZ----------AVERAGE CONDENSATE FILM THICKNESS ON CONVECTIVE
C                   BOUNDARY
```

```
C       AZ-----------Z COORDINATE OF UPPER NODAL POINT OF ELEMENT
C       BZ-----------Z COORDINATE OF LOWER NODAL POINT OF ELEMENT
C       AK-----------SUBDIVISION OF ELEMENT LENGTH(5 FOR FIRST ELEMENT AND
C                    4 FOR OTHER ELEMENTS)
C       HAZ----------WEIGHTED THERMAL CONDUCTIVITY OF TROUGH CONDENSATE
C                         (BTU/HR-FT-DEGF)
C       HAC----------WEIGHTED THERMAL CONDUCTIVITY OF FILM CONDENSATE FOR
C                    ELEMENT IN CONTACT WITH BOTH TROUGH AND FILM CONDEN-
C                         SATE(BTU/HR-FT-DEGF)
C       CONH---------AVERAGE HEAT TRANSFER COEFFICIENT(BTU/HR-FT2-DEGF)
C       HZ-----------LOCAL HEAT TRANSFER COEFFICIENT(BTU/HR-FT2-DEGF)
C       DELA---------FILM THICKNESS AT UPPER NODAL POINT OF ELEMENT(FEET)
C       DELB---------FILM THICKNESS AT LOWER NODAL POINT OF ELEMENT(FEET)
C       H------------AVERAGE HEAT TRANSFER COEFFICIENT OF THE SURFACE OF
C                    CONVECTIVE BOUNDARY ELEMENTS(BTU/HR-FT2-DEGF)
C
C
        IF (BFIN.EQ.0.0D0) GO TO 140
C
C
C       DETERMINE HEAT TRANSFER COEFFICIENT FOR HORIZONTAL SURFACE OF
C       FIN.
C       NOTE:FOR EASE OF ANALYSIS, ASSUME SURFACE IS INSULATED,I.E.
C       H=0.0
        DO 5 IEL=1,NCMREC
           H(IEL)=0C.0D0
5       CONTINUE
C
C
C          DETERMINE HEAT TRANSFER COEFFICIENTS FOR ELEMENTS ON FIN
C          VERTICAL SURFACE
        ZSTAR=ZZERO-(DELSTR(NI)/CALFA)
        HDEN=(-1.0D0*(AA1*ZSTAR**3/3.0D0+BB1*ZSTAR**2/2.0D0))+ZSTAR*(TSAT-
       1T(1))
        CONST=RHOF(NI)**2*OMEGA**2*HFG*CPHI*CALFA*R(NI)
        DELA=DABS(4*CF(NI)*UF(NI)*HDEN/CONST)**0.25D0
        HAC=0.0D0
        INP=1
        DO 130 IEL=NFNTIP,NBSFIN
        AZ=Z(INP)
        BZ=Z(INP+1)
        IF (ZSTAR.LE.BZ) GO TO 10
        GO TO 30
10      IF (HAC.NE.0.0D0) GO TO 110
        IF (IEL.NE.NFNTIP) GO TO 20
        AK=(ZSTAR-AZ)/5.0D0
        ZZ=AK
        GO TO 50
```

```
20    CONTINUE
      AK=(ZSTAR-AZ)/4.0D0
      ZZ=AZ
      GO TO 50
30    IF (IEL.NE.NFNTIP) GO TO 40
      AK=(BZ-AZ)/5.0D0
      ZZ=AK
      GO TO 50
40    CONTINUE
      AK=(BZ-AZ)/4.0D0
      ZZ=AZ
50    CONTINUE
      ZEL=4.0D0*AK
      DO 60 NH=1,5
      HDEN=(-1.0D0*(AA1*ZZ**3/3.0D0+BB1*ZZ**2/2.0D0))+ZZ*(TSAT-T(1))
      HZ(NH)=DABS(CF(NI)**3*CONST/(4.0D0*UF(NI)*HDEN))**0.25D0
      ZZ=ZZ+AK
60    CONTINUE
      CONH=AK*(HZ(1)+4.0D0*HZ(2)+2.0D0*HZ(3)+4.0D0*HZ(4)+HZ(5))/(3.0D0*Z
     1EL)
      IF (ZSTAR.LE.BZ) GO TO 70
      H(IEL)=CONH
      GO TO 120
70    CONTINUE
      HAZ=CONH*(ZSTAR-AZ)
80    CONTINUE
      AZZ=DELSTR(NI)
      DELB=(BZ-ZSTAR)*AZZ/(ZZERO-ZSTAR)
      IF (ZSTAR.EQ.BZ) DELB=DELA
      DELZ=(DELA+DELB)/2.0D0
      IF (AZ.LT.ZSTAR) GO TO 90
      HAC=(BZ-AZ)*CF(NI)/DELZ
      GO TO 100
90    CONTINUE
      HAC=(BZ-ZSTAR)*CF(NI)/DELZ
100   CONTINUE
      H(IEL)=(HAZ+HAC)/(BZ-AZ)
      GO TO 120
110   CONTINUE
      DELA=DELB
      HAZ=0.0D0
      GO TO 80
111   CONTINUE
120   CONTINUE
      INP=INP+1
130   CONTINUE
140   CONTINUE
C
```

```
C        DETERMINE HEAT TRANSFER COEFFICIENT FOR ELEMENTS IN TROUGH
C
      NEXT=NBSFIN+1
      IF (BFIN.EQ.0.0D0) NEXT=1
      DO 150 IEL=NEXT,NENTRF
      H(IEL)=CF(NI)/DELSTR(NI)
150   CONTINUE
C
C        DEFINE HEAT TRANSFER COEFFICIENT ALONG BOTTOM(OUTSIDE) SURFACE
C
      DO 160 ICL=NEXTRT,NEXTLT
      H(ICL)=HINF
160   CONTINUE
      RETURN
      END
C              SUBROUTINE HTCALC PERFORMS HEAT TRANSFER CALCULATIONS
C
      SUBROUTINE HTCALC
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON /GLOB1/  EOA,BFINI,CANGL,CLI,ETOEG,FANGL,HINF,QBTOT,RBASEI,R
     &PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
     1ITOL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
     2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
     3(100,50),QFNTOT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTO
     4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
     5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TIRUF(100),THICK
     6,UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRT,NBSF
     7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
     8M(10),NFSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
C        HEAT RATE CALCULATIONS
C     NOTE FOR ALL HEAT CALCULATIONS THE FOLLOWING VARIABLES ARE
C        DEFINED:
C     KA-----------FIRST NODAL POINT OF ELEMENT
C     KB-----------SECOND NODAL POINT OF ELEMENT
C     XQEL---------DISTANCE BETWEEN NODAL POINTS IN X DIRECTION
C     YQEL---------DISTANCE BETWEEN NODAL POINTS IN Y DIRECTION
C     ELM----------ELEMENT LENGTH ON CONVECTIVE BOUNDARY
C     ELMNT--------ELEMENT LENGTH ON CONVECTIVE BOUNDARY
C     HELMNT-------HEAT TRANSFER COEFFICIENT FOR CONVECTIVE BOUNDARY
C                  ELEMENT
C     QELMNT-------HEAT RATE PER UNIT LENGTH FOR CONVECTIVE BOUNDARY
C                  ELEMENT
C
C
C
C        HEAT RATE INTO FIN SECTION
C
C
C     QEL----------HEAT RATE INTO ELEMENTS ON CONVECTIVE BOUNDARY PER
```

```
C                    INCREMENTAL LENGTH(BTU/HR-FT)
C         QINC---------TOTAL HEAT RATE INTO ONE FIN SECTION FOR A GIVEN IN-
C                      CREMENT(BTU/HR)
C         QET----------TOTAL HEAT RATE INTO ALL FIN SECTIONS FOR A GIVEN IN-
C                      CREMENT(BTU/HR)
C         QFNTOT-------TOTAL HEAT RATE INTO ALL FIN SECTIONS FOR NI INCRE-
C                      MENTS(BTU/HR)
C
          IF (BFIN.EQ.0.0D0) GO TO 30
          QEL=0.0DC
          DO 10 IQEL=1,NBSFIN
          KA=ICOR(IQEL,1)
          KB=ICOR(IQEL,2)
          XQEL=X(KA)-X(KB)
          YQEL=Y(KA)-Y(KB)
          ELM=DSQRT(XQEL**2+YQEL**2)
          ELMNT(NI,IQEL)=ELM
          HELMNT(NI,IQEL)=H(IQEL)
          QELMNT(NI,IQEL)=(2*TSAT-T(KA)-T(KB))*ELM*H(IQEL)/2.0D0
          QEL=QEL+(2*TSAT-T(KA)-T(KB))*ELM*H(IQEL)/2.0D0
10        CONTINUE
          QINC(NI)=QEL*DELX
          QET=QEL*DELX*ZFIN*2
          QFNTOT=QFNTCT+QET
C
C
C            HEAT RATE INTO TROUGH
C
C         QTRF---------HEAT RATE INTO TROUGH ELEMENTS IN CONVECTIVE BOUNDARY
C                      PER INCREMENTAL WIDTH (BTU/HR-FT)
C         QTINC--------TOTAL HEAT RATE INTO THE TROUGH SECTION FOR A GIVEN
C                      INCREMENT(BTU/HR)
C         QTRFT--------TOTAL HEAT RATE INTO ALL TROUGH SECTIONS OF A GIVEN
C                      INCREMENT(BTU/HR)
C         QTFTOT-------TOTAL HEAT RATE INTO ALL TROUGH SECTIONS FOR IN IN-
C                      CREMENTS(BTU/HR)
          QTRF=0.0C0
          NEXT=NBSFIN+1
          DO 20 IQEL=NEXT,NENTRF
          KA=ICOR(IQEL,1)
          KB=ICOR(IQEL,2)
          XQEL=X(KA)-X(KB)
          YQEL=Y(KA)-Y(KB)
          ELM=DSQRT(XQEL**2+YQEL**2)
          ELMNT(NI,IQEL)=ELM
          HELMNT(NI,IQEL)=H(IQEL)
          QELMNT(NI,IQEL)=(2*TSAT-T(KA)-T(KB))*ELM*H(IQEL)/2.0D0
          QTRF=QTRF+(2*TSAT-T(KA)-T(KB))*ELM*H(IQEL)/2.0D0
```

```
20    CONTINUE
      QTINC(NI)=QTRF*DELX
      QTRFT=QTRF*DELX*ZFIN*2.0D0
      QTFTOT=QTFTOT+QTRFT
C
      QINCSM(NI)=0.0D0
      GO TO 50
30    CONTINUE
      QELSM=0.0D0
      DO 40 IQEL=1,NENTRF
      KA=ICOR(IQEL,1)
      KB=ICOR(IQEL,2)
      XQEL=X(KA)-X(KB)
      YQEL=Y(KA)-Y(KB)
      ELM=DSQRT(XQEL**2+YQEL**2)
      ELMNT(NI,IQEL)=ELM
      HELMNT(NI,IQEL)=H(IQEL)
      QELMNT(NI,IQEL)=(2*TSAT-T(KA)-T(KB))*ELM*H(IQEL)/2.0D0
      QELSM=QELSM+(2*TSAT-T(KA)-T(KB))*ELM*H(IQEL)/2.0D0
40    CONTINUE
      QINCSM(NI)=QELSM*DELX
      QSMT=QELSM*DELX*360.0D0
      QSMTOT=QSMTOT+QSMT
50    CONTINUE
C
C        TOTAL HEAT RATE INTO A SECTION THROUGH FIN AND TROUGH
C     QTOTAL-------TOTAL HEAT RATE INTO A GIVEN SECTION THROUGH FIN AND
C                  TROUGH IN A GIVEN INCREMENT(BTU/HR)
      QTOTAL(NI)=QINC(NI)+QTINC(NI)+QINCSM(NI)
C
C
C        HEAT RATE FROM BOTTOM TO AMBIENT
C
C     QBI----------HEAT RATE FROM BOTTOM ELEMENTS OF SECTION PER INCRE-
C                  MENTAL LENGTH(BTU/HR-FT)
C     QBINC--------TOTAL HEAT RATE OUT OF BOTTOM OF SECTION FOR A
C                  INCREMENT(BTU/HR)
C     QBI----------TOTAL HEAT RATE OUT OF BOTTOM OF ALL SECTIONS IN
C                  A GIVEN INCREMENT(BTU/HR)
C     QBTOT--------TOTAL HEAT RATE OUT OF HEAT PIPE FOR NI INCREMENTS
C                  (BTU/HR)
C
      QBI=0.0D0
      DO 60 IQEL=NEXTRT,NEXTLT
      KA=ICOR(IQEL,1)
      KB=ICOR(IQEL,2)
      XQEL=X(KA)-X(KB)
      YQEL=Y(KA)-Y(KB)
```

```
      ELM=DSQRT(XQEL**2+YQEL**2)
      ELMNT(NI,IQEL)=ELM
      HELMNT(NI,IQEL)=H(IQEL)
      QELMNT(NI,IQEL)=(T(KA)+T(KB)-2*TINF)*ELM*H(IQEL)/2.0D0
      QBI=QBI+(T(KA)+T(KB)-2*TINF)*ELM*H(IQEL)/2.0D0
60    CONTINUE
      QBINC(NI)=QBI*DELX
      QB=QBI*DELX*ZFIN*2.0D0
      IF(BFIN.NE.0.0D0) GO TO 70
      QB=QBI*DELX*360.0D0
      QX(NI)=QBINC(NI)*360.0D0
70    CONTINUE
      QBTOT=QBTOT+QB
      RETURN
      END
C        SUBROUTINE SIUNIT CONVERTS A DIMENSIONAL QUANTITIES TO SI UNITS
C        AND OUTPUTS THESE QUANTITIES
C
C
      SUBROUTINE SIUNIT(T1,TPLT,XPLT,YPLT,NFLAG1,NFLAG2,NFLAG3)
C
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION TPLT(1,1),XPLT(1,1),YPLT(1,1),T1(1)
      COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,ETOEO,FANGL,HINF,QBTOT,RBASEI,R
     &PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
     1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
     2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
     3(100,50),QFNTOT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTO
     4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
     5100),TALFA,TBL(100),TBM(100),TBR(100),TESFIN(100),TTRUF(100),THICK
     6,UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRT,NBSF
     7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
     8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
C
C
C        *****     OUTPUT MODE     *****
C
C
C             PRINT HEADER
      WRITE (6,170)
C
C
C             CONVERT DIMENSIONAL INPUT PARAMETERS TO SI UNITS
C
      CLI=CLI/39.37007D0
      RBASEI=RBASEI/39.37007D0
      THICKI=THICKI/39.37007D0
      BFINI=BFINI/39.37007D0
```

```
      HINF=HINF/.176225D0
      TINTL=(TINTL-32.0D0)*(5.0D0/9.0D0)
      TSAT=(TSAT-32.0D0)*(5.0D0/9.0D0)
      TINF=(TINF-32.0D0)*(5.0D0/9.0D0)
C
C
C             OUTPUT INPUT DATA IN SI UNITS
C
      WRITE (6,180) CLI,RBASEI,THICKI,BFINI,CANGL
      WRITE (6,190) FANGL,ETCEO
      WRITE (6,200) CRIT,CRITDL
      WRITE (6,210) RPM,TINTL,TSAT,TINF,HINF
C
C
C             PRINT HEADER FOR CALCULATED RESULTS
      WRITE (6,220)
C
C
C             OUTPUT FIN GEOMETRY PARAMETERS
      SURFAR=SURFAR*.3048D0
      WRITE (6,330) ZFIN,FANGL,SURFAR,ETOEO,BCA,AFOVAS
C
C
C             OUTPUT LATENT HEAT OF VAPORIZATION FOR A GIVEN TSAT
C
      HFG=HFG*2.324444D0
      WRITE (6,340) HFG,TSAT
C
C             OUTPUT HEAT RATE INTO TROUGH,HEAT RATE INTO FIN,TOTAL HEAT
C             RATE IN AND HEAT RATE OUT FOR A SINGLE SECTION FOR EACH
C             INCREMENT
C
      DO 10 NR=1,NDIV
      QINC(NR)=QINC(NR)/3.412322D0
      QTINC(NR)=QTINC(NR)/3.412322D0
      QTOTAL(NR)=QTOTAL(NR)/3.412322D0
      QBINC(NR)=QBINC(NR)/3.412322D0
      QINCSM(NR)=QINCSM(NR)/3.412322D0
      QX(NR)=QX(NR)/3.412322D0
10    CONTINUE
      DMTOT=DMTOT/7936.639D0
      FLOMAS=FLOMAS/7936.639D0
      IF (BFIN.EQ.0.0D0) GO TO 30
C
C             OUTPUT HEAT RATE INTO TROUGH,HEAT RATE INTO FIN,TOTAL HEAT
C             RATE IN AND HEAT RATE OUT FOR A SINGLE SECTION FOR EACH
C             INCREMENT
C
```

```
      WRITE (6,23C)
      DO 20 NR=1,NDIV
      WRITE (6,24C) NR,QINC(NR),QTINC(NR),QTOTAL(NR),QBINC(NR)
20    CONTINUE
      GO TO 8C
30    CONTINUE
      IF (NSCLVE.NE.1) GO TO 40
      GO TO 70
40    IF (NSCLVE.NE.2) GO TO 50
      WRITE(6,260)
      GO TO 70
50    IF (NSCLVE.NE.3) GC TO 60
      WRITE(6,270)
      GO TO 70
60    CONTINUE
      WRITE(6,280)
70    CONTINUE
      WRITE (6,29C)
      DO 80 NR=1,NDIV
      WRITE(6,300)NR,CINCSM(NR),CBINC(NR),QX(NR)
80    CONTINUE
C
C            OUTFUT HEAT RATE INTO FIN, HEAT RATE INTO TROUGH, AND HEAT
C            RATE OUT BOTTOM AND TOTAL MASS FLOW RATE FOR A GIVEN SET
C            OF INPUT CONDITIONS
C
C            OUTFUT HEAT RATE INTO FIN, HEAT RATE INTO TROUGH, AND HEAT
C            RATE OUT BOTTOM AND TOTAL MASS FLOW RATE FOR A GIVEN SET
C            OF INPUT CONDITIONS
C
      QFNTOT=CFNTCT/3.41232200
      QTFTOT=CTFTCT/3.41232200
      QBTOT=QBTCT/3.41232200
      QSMTOT=CSMTCT/3.41232200
      IF (BFIN.EQ.0.0D0) GO TO  90
      WRITE (6,31C) QFNTOT,QTFTOT,QBTOT,DMTOT
      GO TO 1C0
90    IF(PHI.EQ.0.0D0)GO TU 95
      WRITE(6,315)QSMTOT,QBTCT,DMTOT
      GO TO 1C0
95    CONTINUE
      WRITE (6,32C) QSMTOT,QBTOT,DMTCT,FLCMAS
100   CONTINUE
C
C            OUTFUT INCREMENTALLY VARYING PRCPERTIES
C
      WRITE (6,35C)
      DO 110 NR=1,NDIV
```

```
      CF(NR)=CF(NR)*1.729577D0
      CW(NR)=CW(NR)*1.729577D0
      UF(NR)=UF(NR)/2419.088D0
      RHOF(NR)=RHOF(NR)/.0624279D0
      WRITE (6,360) NR,CF(NR),CW(NR),UF(NR),RHOF(NR)
110   CONTINUE
C
C           OUTPUT INCREMENATALLY VARYING PARAMETERS
C
      IF(BFIN.EQ.0.0D0) GO TO 125
      WRITE (6,370)
      DO 120 NR=1,NDIV
      DELSTR(NR)=DELSTR(NR)*.3048D0
      EPS(NR)=EPS(NR)*.3048D0
      R(NR)=R(NR)*.3048D0
      AMTOT(NR)=AMTOT(NR)/7936.639D0
      WRITE (6,380) NR,DELSTR(NR),EPS(NR),R(NR),AMTOT(NR)
120   CONTINUE
      GO TO 128
125   CONTINUE
      WRITE(6,375)
      DO 128 NR=1,NDIV
        DELSTR(NR)=DELSTR(NR)*0.3048D0
        SLNGTH(NR)=SLNGTH(NR)*0.3048D0
        R(NR)=R(NR)*0.3048D0
        AMTOT(NR)=AMTOT(NR)/7936.639D0
        WRITE(6,380)NR,DELSTR(NR),SLNGTH(NR),R(NR),AMTOT(NR)
128   CONTINUE
C
C       OUTPUT MAJOR TEMPERATURES FOR EACH INCREMENT
C
      IF(BFIN.EQ.0.0D0) GO TO 135
      WRITE (6,390)
      GO TO 138
135   CONTINUE
      WRITE(6,395)
138   CONTINUE
      DO 130 NR=1,NDIV
      T1(NR)=(T1(NR)-32.0D0)*(5.0D0/9.0D0)
      TBR(NR)=(TBR(NR)-32.0D0)*(5.0D0/9.0D0)
      TBM(NR)=(TBM(NR)-32.0D0)*(5.0D0/9.0D0)
      TBL(NR)=(TBL(NR)-32.0D0)*(5.0D0/9.0D0)
      TBSFIN(NR)=(TBSFIN(NR)-32.0D0)*(5.0D0/9.0D0)
      TTROF(NR)=(TTROF(NR)-32.0D0)*(5.0D0/9.0D0)
      WRITE(6,400)NR,TBL(NR),TBM(NR),TBR(NR),T1(NR),TBSFIN(NR),TTROF(NR)
130   CONTINUE
C
C       OUTPUT NODAL POINT X AND Y COORDINATES AND FINAL TEMPER-
```

```
C         TURE FOR EACH NODAL POINT FOR INCREMENT OF INTEREST
C
      WRITE (6,410)
      DO 140 I=NFLAG1,NFLAG2,NFLAG3
      WRITE (6,420) I
      WRITE (6,430)
      DO 140 NP=1,NSNP
      XPLT(I,NP)=XPLT(I,NP)*.3048D0
      YPLT(I,NP)=YPLT(I,NP)*.3048D0
      TPLT(I,NP)=(TPLT(I,NP)-32.0D0)*(5.0D0/9.0D0)
      WRITE (6,440) NP,XPLT(I,NP),YPLT(I,NP),TPLT(I,NP)
140   CONTINUE
C
C         OUTPUT ELEMENT LENGTH, HEAT TRANSFER COEFFICIENT AND HEAT
C         RATE PER UNIT LENGTH FOR CONVECTIVE BOUNDARY ELEMENTS FOR
C         INCREMENTS OF INTEREST
C
      WRITE (6,450)
      DO 150 IQEL=1,NEXTLT
      ELMNT(I,IQEL)=ELMNT(I,IQEL)*.3048D0
      HELMNT(I,IQEL)=HELMNT(I,IQEL)*5.674561D0
      QELMNT(I,IQEL)=QELMNT(I,IQEL)/1.040076D0
      WRITE (6,460) IQEL,ELMNT(I,IQEL),HELMNT(I,IQEL),QELMNT(I,IQEL)
150   CONTINUE
160   CONTINUE
C
C
C     *****     OUTPUT FORMAT     *****
      RETURN
170   FORMAT (1H1,////30X,16HINPUT PARAMETERS,/14X,48HALL DIMENSIONAL QU
     1ANTITIES ARE GIVEN IN SI UNITS,////)
180   FORMAT (1X,18HCONDENSER LENGTH =,6X,G10.5,2X,6HMETERS,/1X,16HMINIM
     1UM RADIUS =,8X,G10.5,2X,7HMETERS ,/1X,16HWALL THICKNESS =,8X,G10.5
     2,1X,7H METERS,/1X,12HFIN HEIGHT =,12X,G10.5,1X,7H METERS,/1X,22HCO
     3NDENSER HALF ANGLE =,1X,G10.5,2X,8H DEGREES//)
190   FORMAT (10X,14HFIN PARAMETERS,/1X,16HFIN HALF ANGLE =,G10.5,8H DEG
     1REES,/1X,38HRATIO OF TROUGH WIDTH TO BASE OF FIN =,G10.5,//)
200   FORMAT (1X,'TEMPERATURE COVERGENCE CRITERION =',G10.5,/1X,'MASS FL
     &OW CONVERGENCE CRITREION =',G10.5,//1X,'NOTE:MASS FLOW CONVERGENCE
     & TEST IS ONLY USED FOR IN CYLINDRICAL HEAT PIPE ANALYSIS',//)
210   FORMAT (10X,20HOPERATING PARAMETERS,/1X,5HRPM =,10X,G10.5,23H REVO
     1LUTIONS PER MINUTE,/1X,30HINITIAL TEMPERATURE ESTIMATE =,2X,G10.5,
     210H DEGREES C,/1X,24HSATURATION TEMPERATURE =,8X,G10.5,10H DEGREES
     3 C,/1X,22HEXTERNAL TEMPERATURE =,10X,G10.5,1X,9HDEGREES C,/,1X,36H
     4EXTERNAL HEAT TRANSFER COEFFICIENT =,G10.5,10HW/M2-DEG K,/////)
220   FORMAT (1H1,25X,18HCALCULATED RESULTS,/25X,19HRESULTS IN SI UNITS,
     1////)
230   FORMAT (1H0,1X,36HHEAT RATE SUMMARY FOR A UNIT SECTION,/1X,11HELEM
```

```
     1ENT NR.,3X,10HHEAT RATE ,5X,10HHEAT RATE ,6X,9HHEAT RATE,5X,9HHEAT
     2 RATE,/15X,8HINTO FIN,6X,11HINTO TROUGH,6X,8HTOTAL IN,6X,10HOUT BO
     3TTOM,/15X,8H(WATTS) ,7X,8H(WATTS) ,8X,8H(WATTS) ,6X,8H(WATTS) ,/)
240   FORMAT (1X,I5,10X,G10.5,3(5X,G10.5)/)
260   FORMAT(1X,'RESULTS BASED ON BALLBACK EQUATION FOR TROUGH THICKNESS
     &',///)
270   FORMAT(1X,'RESULTS BASED ON DANIELS  EQUATION FOR TROUGH THICKNESS
     &',/1X,'SIMPLIFIED FORM-FUNCTION OF SHERWOOD NUMBER ONLY',///)
280   FORMAT(1X,'RESULTS BASED ON DANIELS  EQUATION FOR TROUGH THICKNESS
     &',/1X,'FULL FORM, INCLUDES DRAG TERMS',///)
290   FORMAT (1H0,15X,36HHEAT RATE SUMMARY FOR A UNIT SECTION,//1X,10HIN
     1CREMENT ,3X,10HHEAT RATE ,5X,10HHEAT RATE ,8X,'TOTAL HEAT RATE '/1
     22X,12HINTO SECTION,4X,14HOUT OF SECTION,9X,'FOR ALL SECTIONS',/14X
     3,8H(WATTS ),7X,8H(WATTS ),8X,'IN THE INCREMENT',///)
300   FORMAT(1X,I5,10X,G10.5,5X,G10.5,9X,G10.5)
310   FORMAT (1H0,27HTOTAL HEAT RATE INTO FINS  ,09X,F10.3,7H WATTS ,/1X
     1,27HTOTAL HEAT RATE INTO TROUGH,8X,F10.3,7H WATTS ,/1X,32HTOTAL HE
     2AT RATE OUT OF HEAT PIPE,3X,F10.3,7H WATTS ,///1X,20HTOTAL MASS FL
     3OW RATE,3X,G10.5,7H KG/S  ,//)
315   FORMAT (1H0,1X,31HTOTAL HEAT RATE INTO HEAT PIPE ,3X,F10.3,7H WATT
     1S ,///1X,32HTOTAL HEAT RATE OUT OF HEAT PIPE,3X,F10.3,7H WATTS ,//
     2/1X,20HTOTAL MASS FLOW RATE,3X,G10.5,7H KG/M3 ,//)
320   FORMAT (1H0,1X,31HTOTAL HEAT RATE INTO HEAT PIPE ,3X,F10.3,7H WATT
     1S ,///1X,32HTOTAL HEAT RATE OUT OF HEAT PIPE,3X,F10.3,7H WATTS ,//
     2/1X,'TOTAL MASS FLOW RATE BASED ON HEAT RATE DIVIDED BY HFG',F10.3
     3,3X,'KG/SEC',/1X,'TOTAL MASS FLOW RATE AT OVERFALL',F10.3,3X,'KG/S
     4EC',/)
330   FORMAT (10X,23HFIN GEOMETRY PARAMETERS,/1X,9HNR FINS =,5X,G10.5,/1
     1X,14HFIN HALF ANGLE,5X,G10.5,5X,8HDEGREES /,1X,32HFIN SURFACE AREA
     2 PER UNIT LENGTH,01X,G10.5,01X,12HMETER2/METER,/1X,41HRATIO OF TRO
     3UGH WIDTH TO FIN BASE WIDTH =,01X,G10.5,/,1X,32HRATIO OF FIN HEIGH
     4T TO FIN BASE=,01X,G10.5,/,1X,42HRATIO OF FIN SURFACE AREA TO SMOO
     5TH AREA =,01X,G10.5//)
340   FORMAT (1H0,1X,29HLATENT HEAT OF VAPORIZATION =,G10.5,8H  KJ/KG ,/
     11X,20HFOR SATURATION TEMP=,G10.5,9HDEGREES C,//)
350   FORMAT (1H1,10X,43HFLUID AND MATERIAL PROPERTIES PER INCREMENT,/1X
     1,9HINCREMENT,7X,5HFLUID,15X,4HWALL,15X,9HVISCOSITY,12X,7HDENSITY/1
     25X,12HCONDUCTIVITY,7X,12HCONDUCTIVITY,/16X,9HW/M-DEG K,10X,9HW/M-D
     3EG K,13X,6HN-S/M2,14X,5HKG/M3,//)
360   FORMAT (1X,I5,4(10X,G10.5))
370   FORMAT (1H1,10X,32HVARIOUS PARAMETERS PER INCREMENT,/1X,10HINCREME
     1NT ,6X,8HDEL STAR,10X,12HTROUGH WIDTH,8X,14HMINIMUM RADIUS,6X,14HM
     2ASS FLOW RATE,/16X,6HMETERS,14X,6HMETERS,14X,6HMETERS,14X,6H KG/S
     3,/)
375   FORMAT (1H1,10X,32HVARIOUS PARAMETERS PER INCREMENT,/1X,10HINCREME
     1NT ,6X,8HDEL STAR,10X,13HSECTION WIDTH,7X,14HMINIMUM RADIUS,6X,14H
     2MASS FLOW RATE,/19X,6HMETERS,14X,6HMETERS,14X,6HMETERS,14X,6H KG/S
     3 ,/)
```

```
380   FORMAT (1X,I5,4(10X,G10.5))
390   FORMAT(//,17X,'MAJOR TEMPERATURES PER UNIT SECTION FOR ALL INCREME
     &NTS',/,7X,'EXTERNAL',07X,'EXTERNAL',07X,'EXTERNAL',07X,'INTERNAL',
     &6X,'INTERNAL',7X,'INTERNAL'/,8X,'LEFT',08X,'BELOW BASE',08X,'RIGHT
     &',09X,'FIN TIP',07X,'FIN BASE',07X,'TROUGH END',/,7X,'DEGREES C',0
     &6X,'DEGREES C',06X,'DEGREES C',06X,'DEGREES C',05X,'DEGREES C',06X
     &,'DEGREES C',//)
395   FORMAT(//,17X,'MAJOR TEMPERATURES PER UNIT SECTION FOR ALL INCREME
     &NTS',/,7X,'EXTERNAL',07X,'EXTERNAL',07X,'EXTERNAL',07X,'INTERNAL',
     &6X,'INTERNAL',7X,'INTERNAL'/,8X,'LEFT',08X,'  MIDDLE  ',08X,'RIGHT
     &',09X,' LEFT  ',07X,' MIDDLE ',07X,'   RIGHT  ',/,7X,'DEGREES F',0
     &6X,'DEGREES C',06X,'DEGREES C',06X,'DEGREES C',05X,'DEGREES C',06X
     &,'DEGREES C',//)
400   FORMAT (1X,I2,4(5X,G10.5),2(4X,G10.5))
410   FORMAT (1H0,///,10X,63HNODAL POINT COORDINATES AND TEMPERATURE AT
     1A SPECIFIC INCREMENT,/,10X,58HHEAT TRANSFER COEFFICIENT,ELEMENT LE
     2NGTH AND HEAT RATE AT ,/,10X,55HCONVECTIVE BOUNDARY ELEMENTS AT TH
     3IS SPECIFIC INCREMENT,///)
420   FORMAT (1H0,10X,17HINCREMENT NUMBER=,I5,//)
430   FORMAT (10X,39HNODAL POINT COORDINATES AND TEMPERATURE,/,1X,11HNOD
     1AL POINT,5X,8HX-COORD ,09X,7HY-COORD,9X,11HTEMPERATURE,/17X,6HMETE
     2RS,11X,6HMETERS,11X,9HDEGREES C,//)
440   FORMAT (1X,I5,4X,3(07X,G10.5))
450   FORMAT (1H0,10X,38HCONVECTIVE BOUNDARY ELEMENT PARAMETERS,//01X,11
     1HELEMENT NR.,10X,6HLENGTH,10X,13HHEAT TRANSFER,04X,25HHEAT RATE PE
     2R UNIT LENGTH,/22X,6HMETERS,11X,11HCOEFFICIENT,12X,11HWATTS/METER,
     3/38X,14HWATTS/M2-DEG K,//)
460   FORMAT (01X,I5,3(12X,G10.5))
      END
C
C
      SUBROUTINE CELCRV(NDIV,NI,ITER,IFLUID,CLI,TSAT,TAVERG,RBASEI,OMEGA
     &,DELMAX,DELSTR,DRVTE,TAVG,NDEL,TALFA,CALFA,ZZERO,BFINI,EPS,T,CRIT
     &,T1,TBSFIN,NTERM)
      IMPLICIT REAL*8(A-H,L,O-Z)
      DIMENSION ICOR(110,2),DELSAV(110),DEL(110),GK(110,110),DEL1(110)
      DIMENSION GF(110,1),EK(4,4),EB(4,4),EF(4),DERIV(110),NDOF(110,4)
      DIMENSION ELNGTH(110),DELSTR(110),DIFF(110),WKAREA(13000)
      DIMENSION CNST(100),TAVG(100),DRVSAV(110),EPS(100),DERIV1(110)
      DIMENSION CNST1(100),CNST2(100),T(100),EC(4,4),T1(100),TBSFIN(100)
      COMMON/CEL1/RELAX
C     BFNTMP=BFINI
      IF(NDEL.EQ.0)GO TO 10
      NFIN=1
      GO TO 30
10    IF(ITER.EQ.1) GO TO 20
      NFIN=1
      GO TO 30
```

```
20    CONTINUE
      NFIN=0
30    CONTINUE
C     IF(NDEL.EQ.C)BFIN=0.0D0
C     IF(NDEL.EQ.1)BFIN=BFIN1
C     IF(ITER.GT.1)BFIN=BFIN1
      WRITE(6,*)NCALL,NI,NDEL,NDELFN,DELMAX
C
      NCALL=NCALL+1
      NCOUNT=1
C
      R=RBASEI/12.0D0
      CL=CLI/12.0D0
      DELX=CL/DFLCAT(NDIV)
      NEL=NDIV+1
      EPSO=EPS(NI)
C
      TAVGTM=TAVERG
      DO 100 I=1,NEL
      IF(NDEL.EQ.C.ANC.ITER.EQ.1) GO TO 60
      IF(I.EQ.1)GC TO 40
      TAVERG=(TAVG(I)+TAVG(I-1))/2.0D0
      TWALL=(T1(I)+T1(I-1)+TBSFIN(I)+TBSFIN(I-1))/4.0D0
      GO TO 60
40    IF(I.EC.NEL) GO TO 50
      TAVERG=TAVG(I)
      TWALL=(T1(I)+TBSFIN(I))/2.0D0
      GO TO 6C
50    CONTINUE
      TAVERG=TAVG(I-1)
      TWALL=(T1(I-1)+TBSFIN(I-1))/2.0D0
60    CONTINUE
      TO=TAVERG
      TFILM=(TAVERG+TSAT)/2.0D0
C         WATER PRCPERTIES
C
      IF (IFLUID.EQ.1) GO TO 70
      HFG=1093.88D0-0.5703D0*TSAT+0.00012819D0*(TSAT**2)-0.00000008824D0*
     1(TSAT**3)
      RHOF=62.774D0-0.00255698D0*TFILM-0.00005572D0*TFILM**2
      CF=0.3034D0+0.00738927D0*TFILM-0.0000147321D0*TFILM**2
      UF=(0.0C1397D0-C.00001466 9D0*TFILM+0.00000006312 53D0*TFILM**2-
     1.00000CC000976569D0*TFILM**3)*3600.0D0
      CP=-0.0C000C000007D0*TFILM**3+0.0000014764D0*TFILM**2-0.000276
     &88D0*TFILM+1.0109117D0
C
C         FRECN PRCPERTIES
C
```

```
70    IF (IFLUID.EQ.0) GO TO 80
      HFG=69.5459-0.0156011*TSAT-0.000455294*(TSAT**2)+0.00000104144*(TS
     1AT**3)
      RHOF=102.059-0.025364*TFILM-0.000502649*(TFILM**2)+0.000001354
     107*(TFILM**3)
      CF=.0871592253-.00079521657 5*TFILM+6.5849702E-06*TFILM**2-1.85
     &886027E-08*TFILM**3
      UF=(8.449682747E-04-7.85856781E-06*TFILM+4.2075531E-08*TFILM**
     12-9.7346869E-11*TFILM**3)*3600.0D0
80    CONTINUE
      CLMBDA=HFG+(3.0D0/8.0D0)*CP*(TSAT-TO)
      CNST1(I)=(-3.0D0*CF*UF*(TSAT-TO))/(OMEGA**2*R*RHOF**2*CLMBDA)
      IF(CALFA.EQ.1.0D0)GO TO 90
      IF(NFIN.EQ.I)CNST1(I)=CNST1(I)*EPSO
90    CONTINUE
      CNST2(I)=(4.0D0*CF*UF*(TSAT-TWALL))/(RHOF**2*OMEGA**2*R*CLMBDA*
     &CALFA)
      IF(NFIN.EQ.0)CNST2(I)=0.0D0
100   CONTINUE
      TAVERG=TAVGTM
C
C
      ETAMNI=0.55435167201353170 7D+00
      IF(ITER.GT.1)GO TO 120
      IF(BFIN1.NE.0.0D0) GO TO 110
      DELMAX=((-2.0D0*CNST1(1)*CL**2)/(3.0D0*ETAMNI**2))**.2
110   CONTINUE
120   CONTINUE
C
C
C         ESTABLISH CORRESPONDENCE BETWEEN GLOBAL AND LOCAL
      J=1
      JJJ=1
      DO 130 IEL=1,NEL
          ICOR(IEL,1)=J
          ICOR(IEL,2)=J+1
          NDOF(IEL,1)=JJJ
      NDOF(IEL,2)=JJJ+1
      NDOF(IEL,3)=JJJ+2
      NDOF(IEL,4)=JJJ+3
          J=J+1
          JJJ=JJJ+2
130   CONTINUE
      NSNP=NEL+1
      NDOFT=NDOF(NEL,4)
C
C
```

```
C        DEFINE LENGTH OF ELEMENTS
      ELNGTH(1)=DELX/2.0D0
      ELNGTH(NEL)=ELNGTH(1)
      NEL1=NEL-1
      DO 140 I=2,NEL1
         ELNGTH(I)=DELX
140   CONTINUE
C
C
C           INITIALIZE TROUGH THICKNESS(DEL) AND DERIVATIVE (DERIV)
C              AT EACH NODAL POINT
C
150   CONTINUE
      DEL(1)=DELMAX
      NSNP1=NSNP-1
      DO 160 NP=2,NSNP1
         DEL(NP)=DEL(NP-1)-(0.6D0*DELMAX/DFLOAT(NDIV))
         DERIV(NP)=0.0D0
160   CONTINUE
      IF(NFIN.EQ.0)GO TO 170
      DO 170 NP=1,NSNP
         DEL1(NP)=DELMAX
         DERIV1(NP)=DERIV(NP)
170    CONTINUE
C
      ITER1=1
180   CONTINUE
C
C           INITIALIZE K AND F MATRIX
C
      DO 200 I=1,NDOFT
         DO 190 J=1,NDOFT
               GK(I,J)=0.0D0
190      CONTINUE
         GF(I,1)=0.0D0
200   CONTINUE
C
C
C        FORM GLOBAL K AND F MATRICES
      JJ=0
      DO 270 IEL=1,NEL
         II=2*IEL-1
         JJJ=II+1
         MM=JJJ+1
         NN=MM+1
         I1=ICOR(IEL,1)
         I2=ICOR(IEL,2)
         DELAVG=(DEL(I2)+DEL(I1))/2.0D0
```

```
      DRVAVG=(DERIV(I2)+DERIV(I1))/2.0D0
      IF(NFIN.EQ.0)GO TO 210
      DELAVG=(DEL1(I2)+DEL1(I1))/2.0D0
      DRVAVG=(DERIV1(I2)+DERIV1(I1))/2.0D0
210   CONTINUE
      ZSTAR=ZZERO-(DELAVG/CALFA)
      EK(1,1)=-6.0D0/(5.0D0*ELNGTH(IEL))
      EK(1,2)=-11.0D0/10.0D0
      EK(1,3)=-1.0D0*EK(1,1)
      EK(1,4)=-1.0D0/10.0D0
      EK(2,1)=EK(1,4)
      EK(2,2)=(-2.0D0*ELNGTH(IEL))/15.0D0
      EK(2,3)=-1.0D0*EK(2,1)
      EK(2,4)=ELNGTH(IEL)/30.0D0
      EK(3,1)=EK(1,3)
      EK(3,2)=EK(2,3)
      EK(3,3)=EK(1,1)
      EK(3,4)=-1.0D0*EK(1,2)
      EK(4,1)=EK(2,1)
      EK(4,2)=EK(2,4)
      EK(4,3)=EK(3,2)
      EK(4,4)=EK(2,2)
      EB(1,1)=-1.0D0/2.0D0
      EB(1,2)=ELNGTH(IEL)/10.0D0
      EB(1,3)=-1.0D0*EB(1,1)
      EB(1,4)=-1.0D0*EB(1,2)
      EB(2,1)=EB(1,4)
      EB(2,2)=0.0D0
      EB(2,3)=EB(1,2)
      EB(2,4)=(-1.0D0*ELNGTH(IEL)**2)/60.0D0
      EB(3,1)=EB(1,1)
      EB(3,2)=EB(2,1)
      EB(3,3)=EB(1,3)
      EB(3,4)=EB(1,2)
      EB(4,1)=EB(3,4)
      EB(4,2)=-1.0D0*EB(2,4)
      EB(4,3)=EB(3,2)
      EB(4,4)=EB(2,2)
      EB(4,4)=EB(2,2)
      IF(NFIN.EQ.0)GO TO 220
      CDEL=DABS(CNST2(IEL)*ZSTAR)**.75*CALFA
      IF(CALFA.EQ.1.0D0.AND.BFIN1.NE.0.0D0)CDEL=CDEL/EPS0
      CNST(IEL)=CNST1(IEL)-2.0D0*CDEL*DELAVG
      CB=(3.0D0*EPS0*DELAVG**3*DRVAVG)+(4.0D0*DELAVG**4*TALFA*DRVAVG)
220   CONTINUE
      EF(1)=((CNST(IEL)*ELNGTH(IEL))/2.0D0)
      EF(2)=(CNST(IEL)*ELNGTH(IEL)**2)/12.0D0
      EF(3)=EF(1)
```

```
      EF(4)=-1.0D0*EF(2)
      DO 260 J=1,4
         JJ=NDOF(IEL,J)
         DO 250 K=1,4
            KK=NDOF(IEL,K)
               IF(NFIN.EQ.1.AND.CALFA.NE.1.0D0) GO TO 230
               EK(J,K)=(EK(J,K)*DELAVG**4)
               EB(J,K)=(EB(J,K)*3.0D0*DELAVG**3*DRVAVG)
               GO TO 240
230            CONTINUE
               EK(J,K)=EK(J,K)*((EPSO*DELAVG**4)+(DELAVG**5*TALFA))
               EB(J,K)=EB(J,K)*CB
240            CONTINUE
               GK(JJ,KK)=GK(JJ,KK)+EK(J,K)+EB(J,K)
250         CONTINUE
260      CONTINUE
      GF(II,1)=GF(II,1)+EF(1)
      GF(JJJ,1)=GF(JJJ,1)+EF(2)
      GF(MM,1)=GF(MM,1)+EF(3)
      GF(NN,1)=GF(NN,1)+EF(4)
270   CONTINUE
C
C
C        APPLY BOUNDARY CONDITIONS
      NDOFT1=NDOFT-1
      DO 280 I=1,NDOFT
         GK(1,I)=0.0D0
         GK(2,I)=0.0D0
        GK(NDOFT1,I)=0.0D0
280   CONTINUE
      GK(1,1)=1.0D0
      GK(2,2)=1.0D0
      GK(NDOFT1,NDOFT1)=1.0D0
      GF(1,1)=DELMAX
      GF(NDOFT1,1)=0.25D0*DELMAX
      GF(2,1)=0.0D0
C
C
C        SOLVE FOR NEW DEL
      CALL LECT2F(GK,1,NDOFT,110,GF,5,WKAREA,IER)
C
C
C        SAVE OLD DEL (DELSAV) FOR CONVERGENCE TEST AND DEFINE
C        SOLUTION VECTOR AS DEL AND DERIV
      NP1=1
      DO 290 NP=1,NSNP
         DELSAV(NP)=DEL(NP)
         DRVSAV(NP)=DERIV(NP)
```

```
         DEL(NP)=GF(NP1,1)
         DERIV(NP)=GF(NP1+1,1)
      NP1=NP1+2
290   CONTINUE
C
C
C        CHECK FOR CONVERGENCE
C
      NSNP1=NSNP-1
      DO 300 NP=1,NSNP1
         DIFF(NP)=DELSAV(NP)-DEL(NP)
         IF(DABS(DIFF(NP)/DEL(NP)).GT.0.0001) GO TO 310
300   CONTINUE
      GO TO 350
310   CONTINUE
      IF(NFIN.EQ.0) GO TO 320
      DO 320 NP=1,NSNP
         DEL1(NP)=DELSAV(NP)+RELAX*(DEL(NP)-DELSAV(NP))
         DERIV1(NP)=DRVSAV(NP)+RELAX*(DERIV(NP)-DRVSAV(NP))
320   CONTINUE
      IF(ITER1.GE.300)WRITE(6,650)ITER1
650   FORMAT(1X,'TOTAL NUMBER OF ITERATIONS WITHIN DELCRV IS',I5,/)
      IF(ITER1.LT.300)GO TO 330
      NCOUNT=NCOUNT+1
330   CONTINUE
      ITER1=ITER1+1
      IF(NCOUNT.LT.2) GO TO 340
      NTERM=1
      GO TO 370
340   CONTINUE
      GO TO 180
350   CONTINUE
C
C
C        CONVERGENCE HAS BEEN MET,SET TROUGH THICKNESS(DELSTR) EQUAL
C        TO DEL AND DEFINE DERIVATIVE AT OVERFALL AS DRVTE
      NSNP2=NSNP-2
      DO 360 I=1,NSNP2
      DELSTR(I)=DEL(I+1)
360   CONTINUE
      IF(DELSTR(1).GT.DELMAX)DELSTR(1)=DELMAX
      DO 370 NP=1,NSNP2
      NP1=NP+1
      IF(DELSTR(NP).LT.DELSTR(NP1))DELSTR(NP1)=DELSTR(NP)
370   CONTINUE
      DRVTE=DERIV(NSNP)
      DLMXSV=DELMAX
      RETURN
```

LIST OF REFERENCES

1. Ballback, L. J., The Operation of a Rotating Wickless Heat Pipe, M. S. Thesis, Naval Postgraduate School, Monterey, California, December 1969.

2. Tantrakul, C., Condensation Heat Transfer Inside Rotating Heat Pipe, M. S. and M. E. Thesis, Naval Postgraduate School, Monterey, California, June 1977.

3. Schafer, C. E., Augmenting the Heat Transfer Performance of Rotating Two-Phase Thermosyphons, M. S. Thesis, Naval Postgraduate School, Monterey, California, December 1972.

4. Corley, R. D., Heat Transfer Analysis of a Rotating Heat Pipe Containing Internal Axial Fins, M. S. Thesis, Naval Postgraduate School, Monterey, California, June 1976.

5. Purnomo, S. P., The Enhancement of Heat Transfer in a Rotating Heat Pipe, M. S. and M. E. Thesis, Naval Postgraduate School, Monterey, California, June 1978.

6. Davis, W. A., Optimization of an Internally Finned Rotating Heat Pipe, M. S. Thesis, Naval Postgraduate School, Monterey, California, September 1980.

7. Vanderplaats, G. N., COPES-A Fortran Control Program for Engineering Synthesis, User's Manual, prepared for a graduate course on numerical optimization presented at the Naval Postgraduate School, Monterey, California, 1979.

8. Leppert, G., and Nimmo, B. G., "Laminar Film Condensation on Surfaces Normal to Body or Inertial Forces," Transactions of the ASME, p. 178, February 1968.

9. Nimmo, B. G., and Leppert, G., Laminar Film Condensation on Finite Horizontal Surface, Clarkson College of Technology, Potsdam, New York 1970.

10. Wagenseil, L. L., Heat Transfer Performance of Various Rotating Heat Pipes, M. S. Thesis, Naval Postgraduate School, Monterey, California, December 1976.

11. Nusselt, W., "Die Oberflachenkondensation des Wassendamphes," Zeitshcrift des Vereines deutscher Ingenieure, Vol. 60, pp. 541 and 569, 1916.

12. Sparrow, E. M., and Gregg, J. L., "A Theory of Rotating Condensation," Journal of Heat Transfer, Vol. 81, Series C, pp. 113-120, May 1959.

13. Daniels, T. C., and Al-Jumaily, F. K., "Investigations of the Factors Affecting the Performance of a Rotating Heat Pipe," International Journal of Heat and Mass Transfer, Vol. 18, pp. 961-973, 1975.

INITIAL DISTRIBUTION LIST

		No. Copies
1.	Defense Technical Information Center Cameron Station Alexandria, Virginia 22314	2
2.	Library, Code 0142 Naval Postgraduate School Monterey, California 93940	2
3.	Department Chairman, Code 69 Department of Mechanical Engineering Naval Postgraduate School Monterey, California 93940	1
4.	Dr. Paul J. Marto, Code 69 Department of Mechanical Engineering Naval Postgraduate School Monterey, California 93940	2
5.	Dr. David Salinas, Code 69Zc Department of Mechanical Engineering Naval Postgraduate School Monterey, California 93940	2
6.	LT Adam F. Kleinholz 13734 Birkhall Avenue Bellflower, California 90706	3

END

FILMED

9-83

DTIC