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THESIS

THE EFFECT OF CIRCUMFERENTIAL TUBE WALL HEAT
CONDUCTION UPON LAMINAR FILMWISE CONDENSATION
ON THE OUTSIDE OF CONDENSER TUBES

by

Howard Michael Holland

December 1980

Thesis Advisor:

R. H. Nunn

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CONDUCTION UPON LAMINAR FILMWISE CONDENSATION
ON THE OUTSIDE OF CONDENSER TUBES

by

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Submitted in partial fulfillment of the
requirements for the degrees of

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ABSTRACT

This thesis describes the results of a theoretical study to predict the thermal behavior of an internally-cooled tube in a condensing vapor.

The analysis constitutes a unique application of the finite element method and provides new insights into the effects of circumferential conduction upon condenser tube performance. Comparisons are made between the present analysis and the theoretical and experimental works of others. The inclusion of circumferential conduction leads to an improvement in the predictive capabilities of the analytical model.

TABLE OF CONTENTS

I.	INTRODUCTION -----	5
II.	A SPECIFIC PROBLEM FOR STUDY -----	15
III.	RESULTS -----	23
IV.	CONCLUSIONS -----	26
V.	RECOMMENDATIONS -----	27
APPENDIX A:	LOCAL RATE OF CHANGE OF CONDENSATE FILM THICKNESS -----	28
APPENDIX B:	PROGRAM LISTING -----	30
APPENDIX C:	FIGURES -----	45
	LIST OF REFERENCES -----	52
	INITIAL DISTRIBUTION LIST -----	55

I. INTRODUCTION

A. BACKGROUND INFORMATION

As the size, complexity, and hence the cost of modern warships have grown in the last thirty years, advances in basic research have been made and the computer has become a useful tool to the researcher and designer. The application of up-to-date research and computer technology to the design of modern warships is common but there remain major, costly components relatively unchanged and unreviewed in decades. It is in the interest of the Navy to ferret out these components and re-examine them.

One such component is the naval condenser. The basic engineering developments currently used to design naval condensers are from the Heat Exchanger Institute (H.E.I.) Standards for Steam Surface Condensers [1] and the standards of the Tubular Exchanger Manufacturers Association (T.E.M.A.) [2]. The Navy has further utilized the H.E.I. information in the Design Data Sheet (D.D.S.) [3]. While these standards have proven reliable, their performance margins and extensive use of averaged values may result in a larger, costlier condenser at no increase in performance and with an unnecessary margin of reliability. These performance margins, once necessary to insure reliability, are now called into question as the mysteries of condensation are unravelled.

It is for the purpose of reducing the size of the condenser that many researchers are trying to increase the heat transfer rate per unit of condensing surface area.

B. CONDENSATION IN A NAVAL CONDENSER

A typical naval condenser uses the heat transfer mechanism of filmwise condensation to the outside of horizontal, cylindrical tubes. Dropwise condensation is therefore not considered in this thesis.

The inside of the tube has single-phase turbulent forced convection heat transfer. The correlations associated with this phenomenon are accurate enough to justify confidence in their use. The fouling of the inside of the tube is poorly understood, and varies greatly, so that a necessary design allowance is made and a performance margin created. The heat transfer through the tube wall itself is by conduction, a classical problem in cylindrical coordinates, and can be calculated as accurately as the properties of the tube material and the surrounding heat transfer coefficients are known.

Aside from a greater knowledge of coolant-side fouling and how to control it, it is largely on the steam side of the tube that advances of knowledge can be reasonably expected. Here there are three important phenomena. The first is the transfer of heat through a thin layer of condensate from a vapor, either quiescent or flowing, which causes that vapor to lose its latent heat of vaporization and join the film of

condensate on the tube. As condensing particles join this liquid layer they retain some, none, or all of the momentum they possessed as a vapor. The dynamic interaction of the vapor and the liquid under both forced flow and quiescent vapor conditions presents a formidable problem to the researcher. A second phenomenon on the steam side of the tube is the presence of noncondensable gases which complicates and usually degrades the performance of the condenser. The third phenomenon is the inundation of a condenser tube by "condensate rain," or condensate condensed elsewhere in the condenser which, as it makes its way to the hotwell, impinges upon the condenser tube.

C. IMPROVING THE HEAT TRANSFER

If the rate of heat transfer for a given area of condensing surface were to be increased, then the total size of a condenser could be reduced. The result would be savings in space, weight, and possibly cost. Various methods of increasing the heat transfer in a naval condenser have been proposed. Among these are:

1. Reducing the Internal Fouling

The reduction of internal fouling would certainly increase the rate of heat transfer. While various tube metals and chemical treatments to the coolant side of the condenser can reduce fouling, operational and other considerations have prevented their use in naval condensers. Aside from mechanical cleaning, no acceptable method of significantly

reducing marine fouling in a naval condenser has presented itself.

2. Eliminating Noncondensable Gases

The elimination or reduction of noncondensable gases in the condensing steam would be expected to increase the heat transfer rate. It is a fact of realistic operation in a marine environment that noncondensable gases cannot be eliminated. There is room for improvement in their continual removal from the condenser by air ejector devices. As long as complicated systems with mechanical seals are used in the shipboard environment, some amount of noncondensable gases will be in the condenser.

3. Enhancing Heat Transfer

a. Shaping of Tubes

Various ingenious shapes of tubes have been demonstrated to increase heat transfer rate for a given amount of tube surface area. Complications arise from these tubes, however, not the least being a sharp increase in the pumping power required to maintain an adequate coolant flow rate.

b. Promoting Dropwise Condensation

The use of various promoters upon the tube outer surface to prevent the forming of a liquid film, thus keeping the condensate in droplet form, has been proposed and demonstrated by many authors. Since it is not the intention of this thesis to examine the effect of this phenomenon, suffice it to say that until now there remains a question as to whether

any promoter has demonstrated the robustness to withstand continued, long-term use.

4. Eliminating or Reducing Inundation

Since condensers are formed with banks of tubes, it is inevitable that condensate draining from one tube should impinge upon some other lower tube. The effect of this in filmwise condensation is normally to thicken the condensate film and to reduce the rate of heat transfer. The proper placement of collection trays or baffles is of great importance in channelling falling condensate away from lower tubes. The number and placement of these collection trays is not examined in this thesis.

5. Utilizing High Vapor Velocities

In many regions of a naval condenser, vapor velocities of 300 ft/sec or more may occur. These high velocities create a forced flow situation on the steam side, thin the film of condensate, and generally increase the heat transfer coefficients on the steam side. The higher velocities also cause complications, one of which is the separation of the vapor boundary layer causing the condensate film to thicken suddenly. The heat transfer in the region of the tube which is after the vapor phase separation point is relatively unknown.

D. REVIEW OF GENERAL RESEARCH IN CONDENSATION

In analyzing the condensation of steam on tubes, the cornerstone of research was laid by Nusselt [4] who analyzed

a quiescent vapor condensing on a vertical flat plate and on a horizontal cylinder. Nusselt's analysis allowed for no interfacial shear stress between the flowing condensate and the condensing vapor. He also used a constant tube wall surface temperature.

Rohsenow, et al., [5] later expanded this work to include the effect of vapor shear at the vapor-liquid interface upon the condensation rates. This analysis was for a vertical flat plate of uniform temperature with either turbulent or laminar flow of condensate.

Sparrow and Gregg [6] conducted the first analysis of surface condensation using boundary layer theory. They included momentum terms in the condensate film analysis to account for acceleration of the flowing condensate. They showed that the momentum terms were insignificant under most conditions.

Koh, et al., [7] did further work allowing the falling condensate to drag the vapor along with it, while still using a constant wall surface temperature.

Shekriladze and Gomelauro [8] expanded upon the earlier work of Rohsenow, et al., [5], analyzing the effect of a flowing vapor upon laminar filmwise condensation in the absence of significant gravity effects. Their model used a constant wall surface temperature.

Denney and Mills [9] expanded this area of analysis to include the influence of gravity upon the condensate film, holding to a constant wall surface temperature.

E. RECENT WORK CONCERNING CONDENSATION ON TUBES

1. Vapor-Condensate Interfacial Shear Models

One of the areas of continual interest in condensation of steam on a tube is the question of the effect of the shear or drag of the condensing vapor upon the flowing condensate. Theories which attempt to quantify the interaction abound, but recent ones take one of two general forms.

The first type of model says that total shear is the result of the summation of dry friction shear of the vapor upon a bare tube and a shear caused by the condensing vapor retaining some, none, or all of its vapor phase momentum. The deceleration of this vapor as it joins the liquid layer causes this shear. Various authors [8, 10, 11, 12, 13,] subscribe to some combination of these two shears.

Other researchers [14, 15] subscribe to a theory which reaches more deeply into the molecular level interactions and in reality appears to be a more minute examination of the same idea. Using a Reynolds Flux concept, these authors state that for a flowing fluid there is a transport of momentum in a direction normal to the mass flow which is on the molecular level in laminar flow but involves larger amounts of fluid in turbulent flow. This transport is a function of temperature in laminar flow and of temperature and free stream turbulence in turbulent flow. As one of these cross-flowing particles approaches a solid wall, it is decelerated to a standstill. As it rebounds from the wall, it is reaccelerated. Thus,

an overall rate of condensation can be represented by superimposing a velocity perpendicular to the wall, similar to suction through a dry wall.

The latter theory includes within it a relationship between the rate of condensation and temperature drop that must necessarily occur in the process [14]. The former frictional shear force theories commonly assume that the outer surface of the condensate film is at the steam saturation temperature.

2. Wall Temperature Models

Fujii, et al., [16] display from their experimental data that tube temperature varies around the outside surface of the tube. In their calculations, however, they use averaged values for tube wall temperature and for the temperature-dependent heat transfer coefficients to calculate heat transfer.¹

Nobbs and Mayhew [17] show that for steam flowing vertically downward, a variable temperature analysis results in a marked overall change in heat transfer. They state that earlier isothermal tube wall surface models were optimistic in their predicted heat transfer.

¹Since the writing of this thesis, new results have been obtained by Fujii and his coworkers. The reader is referred to the Proceedings of the Workshop on Modern Developments in Marine Condensers, Naval Postgraduate School, 26-28 March, 1980.

Nicol and Wallace [13] show that local heat transfer coefficients are indeed a function of local tube surface temperature and recognize the importance of the circumferential path in heat conduction. In one of the first analyses to include this path of heat transfer, these authors apply a relaxation technique and iterative scheme, changing the temperature distribution and then the local heat transfer coefficients. It is interesting to note that their results show that when the circumferential heat conduction path is included, the average heat transfer coefficient rises about twenty-five percent.

Nicol, et al., [18] analyze steam crossflow upon condenser tube (Fig. 1) using a formulation wherein local tube surface temperatures are used to calculate local heat transfer coefficients. Their results show that the variable wall temperature formulation predicts less heat transfer than older isothermal wall models. The effect of circumferential conduction around the tube wall is not included in their analysis. A close look at Fig. 7 of their work, partially reproduced here as Fig. 2, shows a theoretical wall temperature profile and a few measurements of surface temperature around the tube wall. In their discussion of this particular figure they state:

"The profiles are seen to agree quite well, the main differences being in the rounding off of the more extreme peaks of the theoretical profile because of the circumferential conduction in the tube wall, which is of course more pronounced in those regions where the temperature gradient is high."

Nicol, et al., [18] also present theoretical curves of local heat transfer coefficients, local heat flux, and local tube surface temperature. These cases are presented for both an older isothermal wall model and an anisothermal wall model where heat can flow only radially in the tube wall.

Fujii, et al., [19] use the results of Nicol, et al., [18] for variable wall surface temperature, coupled with earlier work [20] and develop another model based upon the assumption of uniform heat flux around the tube. These authors also recognize the path of circumferential heat conduction as an important area for further work.

F. Purpose of Research

It is in this final area that the author has analyzed, with the computer, a single tube of a condenser and has included in his analysis the circumferential conduction of heat in the tube wall, a phenomenon not customarily included in condenser heat transfer analysis.

II. A SPECIFIC PROBLEM FOR STUDY

A. THE SINGLE TUBE PROBLEM

A review of prior research has indicated ideas and trends. One basic idea was that local heat transfer coefficients depend upon local temperature differences and that the shearing force depends upon these heat transfer coefficients. The trend was away from older isothermal wall models to newer variable wall temperature (anisothermal) models. The logical continuation of the trend would indicate that a wall fully capable of two dimensional heat conduction would have been better still.

Since actual tube walls can conduct heat in the circumferential direction, then a model which could account for this would be an improvement over previous models. In order to test the plethora of shear force and condensation rate theories, a model including the effect of circumferential conduction is desirable, and may prove to be essential.

B. THE PROBLEM STATEMENT

The problem was posed for the upper half of a cylindrical condenser tube with a cross-flowing vapor, that is, where the direction of the gravitational force and the direction of vapor flow were perpendicular, (See Fig. 1a). Symmetry with the other half of the tube was assumed, allowing the simplification that the ends of the half tube, where it would normally

join the other half, were adiabatic walls. The problem for solution was set up so that θ , the local temperature difference, was defined thusly:

$$\theta(r, \phi) = T_s - T(r, \phi) \quad (1)$$

While T_s may vary with pressure changes around the tube, it is assumed constant herein. The governing equation for heat transfer within the tube wall is the Laplace Equation:

$$\frac{\partial^2 \theta}{\partial r^2} + \frac{1}{r} \frac{\partial \theta}{\partial r} + \frac{1}{r^2} \frac{\partial^2 \theta}{\partial \phi^2} = 0 \quad , \quad (2)$$

with the following boundary conditions:

$$q_i = k_w \left(\frac{\partial \theta}{\partial r} \right)_{r_i, \phi} = h_i [\theta(r_i, \phi) - \theta_c]$$

$$q_o = -k_w \left(\frac{\partial \theta}{\partial r} \right)_{r_o, \phi} = h_l \theta_o .$$

On the inside of the tube turbulent forced convection was assumed and the Dittus-Boelter correlation [21] was used to provide a heat transfer coefficient.

On the outside of the tube, the condensate flow was taken to be laminar, and heat transfer through the thin condensate layer was assumed to be purely by conduction. For such conditions the film heat transfer coefficient is given by:

$$h_l = \frac{k_l}{\delta} \quad (3)$$

The film thickness δ was calculated from the formulation due to Nicol, et al., [18] (see also Appendix A). This analysis yields the following first-order differential equation:

$$\frac{d\delta}{d\phi} = \frac{\frac{3\mu r_o k_l \theta_o}{gh_f \rho_l (\rho_l - \rho_v)} - \delta^4 \sin \phi - \frac{3\delta^3}{2g(\rho_l - \rho_v)} \frac{d\tau_v}{d\phi}}{\frac{3\delta^2 \tau_v}{g(\rho_l - \rho_v)} - 3\delta^3 \cos \phi} \quad (4)$$

The shear term, τ_v , used by Nicol, et al., [18] was one given by Schlichting [22] and is of the following form:

$$\tau_v = \frac{\frac{1}{2}\rho_v U_v^2}{\left(\frac{U_v r_o \rho_v}{\mu_v}\right)^{1/2}} (6.973\phi - 2.732\phi^3 + 0.292\phi^5 - 0.0183\phi^7 + 0.000043\phi^9 - 0.000115\phi^{11}). \quad (5)$$

For the sake of comparison this author used the same shear stress model. This model calculates vapor boundary layer separation at about 108 degrees from the forward stagnation point. Thereafter, the shear stress was set to zero.

In order to integrate Eq. (4) a starting value of condensate film thickness at the stagnation point was needed. Because high velocity steam flowing horizontally normal to a horizontal tube was to be studied, $\frac{d\delta}{d\phi}$ was set equal to zero for the stagnation point ($\phi = 0$), and an approximate starting value for the condensate film thickness, δ , was found from Eqs. (4) and (5).

Once this starting value was in hand, Eq. (4) was integrated, in its entirety, around the tube using an estimate of the tube surface temperature difference, $[\theta_o = \theta(r_o, \phi)]$. The integration resulted in the local film thickness around the tube wall. Local heat transfer coefficients for the steam side were then calculated according to Eq. (3).

Using these local heat transfer coefficients Eq. (2) could then be integrated, resulting in a temperature distribution throughout the solid tube wall. With a new set of tube-wall outer surface temperatures in hand, a new set of local heat

transfer coefficients could be calculated and applied for another iteration of the Laplace equation. If temperatures converged and heat transfer balanced, then a solution was in hand.

C. THE GALERKIN FINITE ELEMENT METHOD

The purpose of this application of the Galerkin Finite Element method is to solve a steady state two dimensional heat conduction problem by converting the governing partial differential equation, the Laplace equations in this case, to a series of algebraic equations. Each of these algebraic equations represents the partial differential equation over a small sub-region of the total domain called an element. The result of the simultaneous solution of these algebraic equations is to drive the integral square error of the approximate solution to zero over each element. Elements may be of different sizes and may be clustered in areas where greater accuracy is desired.

The Galerkin Finite Element method differs from the Galerkin method in that, since the algebraic equation representing the approximate solution exists over only one small element, the resultant matrix of algebraic equations for solution takes on a banded form. That is, it has zero upper and lower triangles of significant size. The resultant savings in computer storage space and computational time by using special matrix equation solvers is significant.

In this discreet approximation method the boundary conditions are applied locally to each element on the boundary of the domain. In this case linear triangular elements were used. The algebraic approximations to the solution, called basis or shape functions, were linear. The resultant approximate solution amounts to approximating a smooth three dimensional surface by a mosaic containing a large number of small triangles. Triangles size varied, small ones being used in such areas of interest as the forward stagnation point and anywhere a steep temperature gradient could be expected to exist. Larger elements were used in areas where the temperature profile was considered to be relatively unchanging. As solutions progressed, refinement of the triangular meshes highlighted and concentrated upon areas of interest.

For greater detail on both the Galerkin method and the Galerkin finite element method, the interested reader is referred to Ozisik [23] and Fairweather [24].

The formulation and computer program were tested using classical two-dimensional problems of heat conduction from Carslaw and Jaeger [25]. The first of these was a rectangle with one side at a given temperature and the other three at a specified lower temperature. The second was a rectangle with two adjoining sides adiabatic, a third side at a given temperature, and the fourth having a given heat transfer

coefficient to a medium at some lower temperature. In both cases the finite element solutions converged to exact solutions.

A further test of the program and formulation was the problem of a rectangle with two opposite sides adiabatic, a third side at a given temperature, and the fourth side having a local heat transfer coefficient to some medium at a lower temperature. This local heat transfer coefficient was dependent upon both the local surface temperature and the position along the wall. Since this problem cannot be compared to a classical solution, any solution could only be given credence by detailed examination. It was seen that the solution converged for smaller and smaller triangle sizes, that plots of isotherms internal to the rectangle were smooth and not obviously in error, and that heat transfer into and out of the rectangle was in balance. While certainly no proof, these comparisons led to confidence in the program.

D. APPLICATION OF THE INNER BOUNDARY CONDITION

For the inside of the tube, the boundary condition of convection heat transfer, Eq. (2), can be simply represented by noting that the inside film coefficient, h_i , is given by the Dittus-Boelter correlation [21], and is a function of coolant bulk temperature and flow rate. Thus, since the thermal resistance at this interface is independent of local tube wall temperature, the resistance may be represented by a fictitious

thickness of insulating material according to the relationship:

$$R = \frac{l}{h_i A} = \frac{r_i \ln \left(\frac{r_i}{r_i - t_f} \right)}{K_f A} \quad (6)$$

From this, the thickness of the fictitious ring t_f , was determined and the equivalent boundary condition, that the coolant temperature existed at the inside of the fictitious ring, was imposed.

E. SOLUTION PROCEDURE

On the outside of the tube, the local heat transfer coefficients were computed all around the tube for the current value of the calculated local outside wall temperature. These local heat transfer coefficients were applied to elements having sides on the outer tube wall.

In order to converge to the proper wall temperatures and temperature-dependent heat transfer coefficients, an iterative scheme was followed wherein a new set of local heat transfer coefficients for the outside of the tube were computed from new solution temperatures. The process was repeated until temperatures at all nodal points changed less than 0.02 degrees Celsius for consecutive iterations.

Having stabilized the local temperatures throughout the tube wall, a check was made to insure that the heat transfer balanced. The total heat deposited upon the condenser tube outer wall by the condensing steam was compared to the total heat conducted into the tube wall at the outer surface and to the total heat conducted from the tube at its inner surface.

F. COMPARISON WITH PREVIOUS SOLUTIONS

The aforementioned paper of Nicol, et al., [18] provided fertile ground for comparison, but only if the test conditions could be simulated. These authors were able to achieve a given average tube outer surface temperature by varying the flowrate of the coolant and hence the internal heat transfer coefficient. In the finite element formulation this condition is obtained by varying the thickness of the aforementioned fictitious ring. In order to preclude significant circumferential conduction in the fictitious ring, the circumferential thermal conductivity of the fictitious ring was chosen to be one one-hundredth of the thermal conductivity of the tube wall.

III. RESULTS

A. TUBE WALL TEMPERATURE

The tube wall temperature predictions of the author's finite element model are shown in Fig. (3), and are seen to support the prediction of Nicol et al., [18] that circumferential heat conduction would substantially round off the theoretical temperature profile obtained by considering radial heat conduction only. It is also worthwhile to note that the maximum and minimum temperatures are less extreme than those predicted by the radial heat flow model of Nicol et al., [18] and generally agree more closely with their experimental results. While the actual differences in temperature appear small, near the separation point of the vapor flow these differences approach thirty percent.

Nicol et al., [18] also show a theoretical tube wall surface temperature profile for another average wall temperature. This profile is reproduced in Fig. 4 along with a theoretical profile developed from the author's model. As before, the steep gradients are rounded off and the overall temperature profile is less extreme.

A plot of the isotherms within the tube wall is quite revealing and is shown in Fig. (5). The direction of vapor flow and point of vapor boundary layer separation, as indicated by the thickening of the condensate layer, are shown on the

figure. The isotherms are seen to start close together near the vapor stagnation point and to spread, reflecting lower rates of heat transfer, progressively around the tube. In the vicinity of separation of the vapor boundary layer, the isotherms are seen to deviate significantly from their concentric upstream orientation. This reflects a significant amount of heat transfer circumferentially around the tube in this general vicinity. The much wider spacing of the isotherms on the back part of the tube indicates that according to this model it is an area of relatively low heat transfer rate.

1. Local Heat Transfer Coefficients

Figure 6 shows a comparison of local heat transfer coefficients around the tube wall. The isothermal wall and radial heat flow local transfer coefficients of Nicol et al., are partially reproduced. The results of the finite element model are seen to lie generally between the other two models with the exception of the vapor-separated region. In the region prior to separation, as seen in Fig. 4, the wall temperature difference of the finite element model lies between the isothermal wall case and radial heat flow case. From Eq. (4) at the stagnation point, the initial value of the film thickness is proportional to $\theta_0^{1/3}$. Therefore, a larger temperature difference will create a larger initial film thickness, and therefore a smaller heat transfer coefficient, as given by Eq. (3). It is not unexpected that this formulation and solution should predict a greater heat

transfer coefficient in the separated region than the isothermal wall model. Since there was a higher wall temperature and therefore less steam condensed in the unseparated portion of the tube in the present model, there is a thinner condensate layer in the after section of the tube. In the separated region the greater heat transfer coefficient predicted by the author's model over the radial heat flow model is not fully explained, but is no-doubt due, in part, to the assumption of symmetry in the present formulation.

2. Local Heat Flux

In still a third examination under these same conditions, Nicol et al. in their Fig. 6 present local heat flux for both the isothermal wall and radial heat flow models. These profiles are partially reproduced here in Fig. 7 along with the author's results which again, as expected, lie between the two previous models except in the separated-vapor region of the tube. It is interesting to note that the inclusion of circumferential effects generates a heat flux which is much closer to being uniform than that which is obtained with the isothermal wall model.

IV. CONCLUSIONS

The circumferential path of heat conduction has been successfully included in the solution of laminar filmwise condensation of high velocity steam on condenser tubes. The author's anisothermal fully-conducting wall model yields closer agreement to previously measured tube wall surface temperatures than a previous anisothermal wall model which allowed only radial heat flow. The maximum and minimum values of wall temperature were less extreme for the anisothermal fully conducting wall than for the anisothermal wall with only radial heat flow.

The addition of a circumferential path for heat flow results in a complicated interplay between this heat flow and heat being deposited locally by condensation and leaving in the internal cooling water. The addition of vapor velocity effects on the outside surface of the film, and especially the sudden changes arising from separation of the vapor flow result in a highly complex sequence of thermal and fluid-dynamical interactions. The methods developed herein, though still subject to comprehensive experimental verification, are adequate to predict these complex phenomena.

V. RECOMMENDATIONS

Now that a model for two-dimensional heat conduction which includes temperature-dependent local heat transfer coefficients has been developed and demonstrated, the use of this model for further study is suggested. Areas thought fruitful for further examination are both the shear stress models and the temperature-dependent condensation rate models. For shear stress models the separated vapor region or trailing area must be given closer examination since it accounts for about forty percent of the surface area of the tube and the fluid flow characteristics in this region remain relatively unknown. In the study of temperature-dependent condensation rate models, further examination of the effect of the velocity of the flowing vapor and local temperature is expected to be fruitful.

The vigorous exercise of this model through a broad range of steam-side and coolant side conditions is suggested.

The model of condensation with constant heat flux proposed by several authors [18, 20, 26], should be re-examined since circumferential heat conduction in the tube wall, when included in the analysis, seems to yield a nearly constant heat flux distribution.

APPENDIX A

Local Rate of Change of Condensate Film Thickness

A force balance on the shaded element of Fig. 1b yields the following:

$$\tau_v r_0 d\phi - \tau_\ell r_0 d\phi - g(\rho_\ell - \rho_v) \cos\phi r_0 (\delta - y) d\phi = 0$$

so that:

$$\tau_\ell = \mu \frac{\partial u}{\partial y} = \tau_v - (\rho_\ell - \rho_v) g \cos\phi (\delta - y) .$$

Integrating:

$$u = \frac{\tau_v}{\mu} y - (\rho_\ell - \rho_v) \frac{g}{\mu} \cos\phi \left(\delta y - \frac{y^2}{2} \right) .$$

The average velocity, $\bar{u}$ is:

$$\begin{aligned} \bar{u} &= \frac{1}{\delta} \int_0^\delta u dy = \frac{1}{\delta} \frac{\tau_v}{\mu} \frac{\delta^2}{2} - \frac{g}{\mu} \frac{(\rho_\ell - \rho_v)}{\delta} \cos\phi \left[\frac{\delta^3}{2} - \frac{\delta^3}{6} \right] , \text{ or} \\ \bar{u} &= \frac{\tau_v \delta}{2\mu} - (\rho_\ell - \rho_v) \frac{g}{\mu} \cos\phi \frac{\delta^2}{3} . \end{aligned} \quad (8)$$

The mass flow rate of condensate around the tube is:

$$\dot{m}_\ell = \rho_\ell \delta \bar{u}$$

and, with Eq. (8),

$$\dot{m}_\ell = \frac{\tau_v \rho_\ell}{2\mu} \delta^2 - \frac{g \rho_\ell (\rho_\ell - \rho_v)}{3\mu} \cos\phi \delta^3 .$$

The rate of change of mass flow rate with respect to angle is:

$$\begin{aligned} \frac{d\dot{m}_\ell}{d\phi} &= \frac{\rho_\ell \delta^2}{2\mu} \frac{d\tau_v}{d\phi} + \frac{\rho_\ell \tau_v \delta}{\mu} \frac{d\delta}{d\phi} + \frac{g \rho_\ell (\rho_\ell - \rho_v) \delta^3 \sin\phi}{3\mu} \\ &\quad - \frac{g \rho_\ell (\rho_\ell - \rho_v)}{\mu} \cos\phi \delta^2 \frac{d\delta}{d\phi} \end{aligned}$$

Since $d\dot{m}/d\phi$ can also be equated to the amount of steam condensing, the following relationship applies:

$$\frac{d\dot{m}_l}{d\phi} = \frac{r_o k_l \theta_o}{\delta h'_{fg}}$$

where h'_{fg} is the latent heat of vaporization corrected for subcooling. By making this substitution and solving for $d\delta/d\phi$, the relationship given in Eq. (4) is obtained.

APPENDIX B: PROGRAM LISTING

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A FINITE ELEMENT PROGRAM USING FIRST ORDER LINEAR GALERKIN
FINITE ELEMENTS TO SOLVE TWO-DIMENSIONAL HEAT TRANSFER
PROBLEMS WITH NON-LINEAR, TEMPERATURE DEPENDENT BOUNDARY
CONDITIONS.

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THE PRESENT SET UP IS FOR A CYLINDRICAL CONDENSER TUBE
WITH FILMWISE CONDENSATION ON THE OUTSIDE AND TURBULENT
FORCED CONVECTION ON THE INSIDE.

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THE FOLLOWING IS A LIST OF
VARIABLE NAMES USED IN THE
PROGRAM.

VARIABLES OF THE GENERAL PROBLEM.

ATRI	THE AREA OF EACH TRIANGLE IN THE FINITE ELEMENT GRID.
CSUEP	VALUE OF SPECIFIC HEAT OF LIQUID WATER AT CONSTANT PRESSURE.
FNEEC	VALUES OF THE FORCING FUNCTION AT EACH ESSENTIAL BOUNDARY CONDITION.
F	A VECTOR OF RIGHT HAND SIDES FOR SOLVING THE MATRIX EQUATION RESULTING FROM THE FINITE ELEMENT FORMULATION.
FF	LOCAL HEAT TRANSFER COEFFICIENT.
FCL	LOCAL HEAT TRANSFER COEFFICIENT.
KERR	THE CORRESPONDENCE TABLE OF THE POSITIONS OF THE THREE CORNERS OF EACH TRIANGLE.
NAX	THE TOTAL BANDWIDTH OF THE BANDED GLOBAL MATRIX.
NCOLN	A COUNTER.

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VAT000100
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VAT000148

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VAT000490
VAT000500
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VAT000520
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VAT000600
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VAT000690
VAT000700
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VAT000890
VAT000900
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VAT000920
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VAT000940
VAT000950
VAT000960

NCELI A VECTOR CONTAINING THE NUMBER OF EACH CONVECTION
NCELC ELEMENT IN THE INSIDE OF THE DOMAIN.
NCHK A VECTOR CONTAINING THE NUMBER OF EACH CONVECTION
NCELC ELEMENT IN THE OUTSIDE OF THE DOMAIN.
NCELC A VALUE USED TO CHECK AGREEMENT FROM ITERATION TO
ITERATION.
NEBC THE NUMBER OF EACH OF THE NODAL POINTS WHERE A
TEMPERATURE IS SPECIFIED.
NEL THE TOTAL NUMBER OF ELEMENTS.
NLC THE NUMBER OF LOWER CO-DIAGONALS OF THE GLOBAL MATRIX
IN RANGL STORAGE FORM.
NNP THE TOTAL NUMBER OF NODAL POINTS.
NSDC FOR EVERY ELEMENT HAVING A CONVECTION BOUNDARY, THIS
IS THE NUMBER OF SIDES ON THE BOUNDARY.
NUC THE NUMBER OF UPPER CO-DIAGONALS OF THE GLOBAL MATRIX
IN RANGL STORAGE FORM.
RHOL THE DENSITY OF LIQUID WATER AT SATURATION CONDITIONS.
RHOU THE DENSITY OF WATER VAPOR AT SATURATION CONDITIONS.
X THE HORIZONTAL COORDINATES IN THE CARTESIAN COORDINATE
SYSTEM.
Y THE VERTICAL COORDINATE IN THE CARTESIAN COORDINATE
SYSTEM.
XLO12 THE LENGTH OF THE SIDE OF A CONVECTION ELEMENT FROM
LOCAL POINT 1 TO LOCAL NODAL POINT 2.
XLO23 THE LENGTH OF A SIDE OF A CONVECTION ELEMENT
FROM LOCAL NODAL POINT 2 TO LOCAL NODAL POINT 3.
XL SPACE FROM THE LINEAR EQUATION SOLVER.
TEAR THE AVERAGE OF LOCAL TEMPERATURE ALONG THE CONVECTIVE
BOUNDARY.
ZA THE GLOBAL MATRIX FORMED IN THE FINITE ELEMENT METHOD,
STORED IN RANGL STORAGE MCCE.
ZAST A STORAGE PLACE FOR THE GLOBAL MATRIX SO THAT ONCE IT IS
INITIALLY FORMED IT NEED NOT BE FORMED AGAIN FOR
EACH SUCCESSIVE ITERATION.
XKLIQ THE THERMAL CONDUCTIVITY OF LIQUID WATER AT SATURATION
CONDITIONS.
XKSOL THE THERMAL CONDUCTIVITY OF THE SOLID TUBE MATERIAL.
XMUL THE DYNAMIC VISCOSITY OF LIQUID WATER AT THE SATURATION
CONDITIONS.
XMUV THE DYNAMIC VISCOSITY OF WATER VAPOR AT THE SATURATION
CONDITIONS.
XHFG THE LATENT HEAT OF VAPORIZATION OF WATER AT THE
SATURATION CONDITIONS.
VELST THE FREE STREAM VELOCITY OF THE FLOWING STEAM.

CC

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120
142
148
CCCCCCCC

CVERT IS A SUBROUTINE WHICH READS IN VALUES OF ANGLE
 FROM THE FORWARD STAGNATION POINT AND RADII FROM THE
 CENTER OF THE TLEE AND CONVERTS THESE TO X AND Y CARTESIAN
 COORDINATES.

VAT0145C
VAT0146C
VAT0147C
VAT0148C
VAT0149C
VAT0150C
VAT0151C
VAT0152C
VAT0153C
VAT0154C
VAT0155C
VAT0156C
VAT0157C
VAT0158C
VAT0159C
VAT0160C
VAT0161C
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VAT0180C
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VAT0184C
VAT0185C
VAT0186C
VAT0187C
VAT0188C
VAT0189C
VAT0190C
VAT0191C
VAT0192C

CALL CVERT(X,Y,R,PTEE,NPF)

KCRF IS A SUBROUTINE USED TO DEVELOP THE CORRESPONDENCE
TABLE IN ORDER TO CONNECT THE NODAL POINTS IN THE PROPER ORDER
TO FORM THE PROPER TRIANGULAR GRID.

CALL KCRF(KCRR,NEL)

CECM IS A SUBROUTINE USED TO FIGURE THE AREA OF EACH
TRIANGULAR ELEMENT AND THE LENGTH OF ANY SIDE ALONG A CON-
VECTION BOUNDARY CONDITION.

CALL CECM(NNF,NEL,X,Y,KCRR,ATRI,NCELO,XLO12,XLO23,
INCC,NSEC)

FCRMB IS A SUBROUTINE WHICH TAKES THE X AND Y COORDINATES
AND THE AREA OF EACH TRIANGLE AND CONSTRUCTS THE LOCAL THREE BY
THREE MATRIX FOR EACH ELEMENT. IT THEN SUPERIMPOSES EACH OF
THE VALUES FROM THE LOCAL THREE BY THREE MATRIX UPON THE PROPERLY
NUMBERED NODAL POINTS ACCORDING TO THE CORRESPONDENCE TABLE.

CALL FCFME (ATRI,NEL,NNP,X,Y,ZA,KORR,NLC,NLC)

ZASTC IS A SUBROUTINE WHICH STORES THE GLOBAL MATRIX, ZA,
SO THAT IT NEED NOT BE REFORMED FOR EACH SUCCESSIVE ITERATION.

CALL ZASTC(NNP,ZA,ZAST,NLC,NLC)

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VATC153C
VATC154C
VATC155C
VATC156C
VATC157C
VATC158C
VATC159C
VATC200C
VATC201C
VATC202C
VATC203C
VATC204C
VATC205C
VATC206C
VATC207C
VATC208C
VATC209C
VATC210C
VATC211C
VATC212C
VATC213C
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VATC218C
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VATC227C
VATC228C
VATC229C
VATC230C
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VATC232C
VATC233C
VATC234C
VATC235C
VATC236C
VATC237C
VATC238C
VATC239C
VATC240C

TEMDS IS A SUBROUTINE WHICH DISTRIBUTES INITIAL VALUES OF
TEMPERATURE DIFFERENCE (FIRST GUESSES) AT THE CONVECTION
ASSOCIATED NODAL POINTS SO THAT THE LOCAL TEMPERATURE DEPENDENT
HEAT TRANSFER COEFFICIENTS CAN BE CALCULATED THE FIRST TIME.

CALL TEMDS(TEM,NBNPC,NCANPO)

FLCCC IS A SUBROUTINE WHICH FIGURES THE LOCAL HEAT TRANSFER
COEFFICIENTS ON THE OUTSIDE OF THE TUBE. THIS SUBROUTINE IS
INCLUDED IN ALL ITERATIONS.

CALL FLCCC(X+FG,R+CL,R+CV,XMUL,XMOV,XKLIQ,XKSCL,PHEE,R,TEM,HOK,
INCELC,KCRR,NGO,TPAR,FF,FEEBAR,VELST,CSLBF)

TFACT=C.
CO=3.24
IF(L=1,NGO
TFACT=TFACT+TBAR(JU)*XLC12(JU)
AR=AR+XLC12(JU)
CONTINUE
TAV=TFACT/AF
WRITE(6,165)TAV
WRITE(7,165)TAV
FORMAT(10,AVERAGE TEMP IS',F10.5)

FORMF IS A SUBROUTINE WHICH PLACES ZEROS IN THE RIGHT HAND
SIDE OF THE MATRIX EQUATION.

CALL FORMF(F,NNP)

FORMCD IS A SUBROUTINE WHICH MODIFIES THE GLOBAL MATRIX FOR
THE CONVECTION BOUNDARY CONDITIONS.

CALL FORMCD (NNP,ZA,KCRR,ACELC,NSDO,XLC12,XLC23,FCK,NGO
1,NLC,NLC)

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765

CCCCCCCCC CCCCCCCCCC

VAT02241C
VAT02242C
VAT02243C
VAT02244C
VAT02245C
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VAT02251C
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VAT02258C
VAT02259C
VAT02260C
VAT02261C
VAT02262C
VAT02263C
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VAT02277C
VAT02278C
VAT02279C
VAT02280C
VAT02281C
VAT02282C
VAT02283C
VAT02284C
VAT02285C
VAT02286C
VAT02287C
VAT02288C

PCCBCE IS A SUBROUTINE WHICH MODIFIES THE GLOBAL MATRIX AND
THE FORCING FUNCTION FOR THE SPECIFIED TEMPERATURES (ESSENTIAL
BOUNDARY CONDITIONS.)

CALL PCCBCE(ZA,F,NNP,NCCUN,NEBC,NBC,FNEBC,NLC,NUC)
MAX=NLC+1

LECTIE IS A SINGLE PRECISION LINEAR EQUATION SOLVER
SPECIFICALLY PROGRAMMED TO WORK UPON BANDED MATRICES.

CALL LECTIE(ZA,NNP,NLC,NUC,320,F,1,1,0,XL,IER)

CLOSE IS A SUBROUTINE WHICH CHECKS AFTER EACH ITERATION TO
SEE HOW CLOSE THE NEW TEMPERATURE DIFFERENCES AGREE WITH THE OLD
ONES. IF AGREEMENT IS NOT SUFFICIENT, IT REPLACES THE PRESENT
GLOBAL MATRIX WITH THE OLD, UNMODIFIED ONE AND CAUSES
THE ITERATIONS TO BEGIN AGAIN, BUT WITH THE NEW VALUES
OF LOCAL TEMPERATURE DIFFERENCE.

CALL CLOSE(F,TEM,NMF,NENFC,NCANPO,ZA,ZAST,NCCUN,NCHK
1,NUC,NLC)
IF(NCCUN.EC.51) GC TC 103
IF(NCHK.EC.5) GC TC 3
NST=1
CALL PCCCC(XFFG,RFCL,RFV,XPUL,XMUV,XKLIC,XKSOL,PHEE,R,TEM,HOK,
1,ACELC,KCORR,NCC,TBAR,FF,FEEBAR,VELST,CSUEP)
CALL FCF(F,NMP)
FORMAT(1,NCC FOR FORMC12=,15)
CALL FCFMCD(NNP,ZA,KCORR,ACELC,NSDG,XLC12,
1,XLC23,FCF,NCC,NUC,NLC)
CALL PCCBCE(ZA,F,NNP,NCCUN,NEBC,NBC,FNEBC,NLC,NUC)
CALL LECTIE(ZA,NNP,NLC,NUC,320,F,1,1,0,XL,IER)
EC 541 NMF=1,NNP,9

CCCCCCCC CCCCCC CCCCCCCCCCCCCCCCCC

4 474

[illegible]


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1+INCELC(AL),DELC
DELC=LELAL/2.+DELC
FORMAT(15,5(F20.9,2X))
310
CCATINLE
FAVAV=FFFF/3.141592654
WRITE(6,24)HAWAV
343
FORMAT(1,1,AREA WEIGHTED AVERAGE + IS: ',F12.2)
RETURN
FORMAT(15,15)
111
END
SUBROUTINE FORMOD (ANF,ZA,KCRR,ACELO,NSCC,XLC12,
1XLC23,FCR,NCC,NUC,NLC)
DIMENSION Z(320,25),KCRF(600,3),NCELO(40),NSCO(40),HOK(600)
REAL *8 XLC12(40),XLC23(40)
FORMAT(1,1,INFORMC',4(15))
1XLC12=1,NCO
FCFMT(1X,15,2(F14.5))
XNUM=FCR(NCELC(JR),1)+XLC12(JR)/3.
IF(NSCC(JR).EQ.2) GC TC 185
ZA(KCRR(NCELC(JR),1),1)=ZA(KCRR,NCELO(JR),1),(NLC+1)+XNUM
1ZA(KCRR(NCELC(JR),1),1),(KCRR(NCELO(JR),2)-KORP(NCELC(JR),1)+NLC+1))=
1+XNUM/2
ZA(KCRR(NCELC(JR),2),(NLC+1))=ZA(KCRR(NCELC(JR),2),(NLC+1))+XNUM
1ZA(KCRR(NCELC(JR),2),(KORR(NCELO(JR),1)-KORR(NCELC(JR),2)+NLC+1))=
1+XNUM/2
CCNTINLE
RETURN
185
END
SUBROUTINE MCD BCE (ZA,F,NP,NCCUN,NEEC,NBC,FNEBC,NLC,NUC)
DIMENSION Z(320,25),F(320),NEBC(40),FNEBC(40)
IF(NCCUN.EQ.0) GC TC 801
FORMAT(1,2X,15)
NAX=NLC+NLC+1
DO 148 IT=1,NBC
CO 144 IC=1,NAX
1ZA(NEBC(IT),IC)=0.
1ZA(NEBC(IT),(NLC+1))=1.
F(NEBC(IT))=FNEBC(IT)
CCNTINLE
FORMAT(10,5)
RETURN
END
SUBROUTINE CLOSE(F,TEM,NP,NBNPO,NCANPO,ZA,ZAST,NCCUN,NCFK
1,NLC,NLC)
DIMENSION F(320),TEM(320),NBNPO(40),Z(320,25),ZAST(320,25)

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321 I=1,NMF
322 FORMAT(10F10.5)
323 I=1,NCANFC
324 FORMAT(1F1)
325 FORMAT(1E)
326 NCFK=C
327 FORMAT('IN CLOSE' )
328 GO 705 -R=1,NCANFC
329 JT=NCANFC-JR+1
330 IF(AES(F(NBNFC(JT))-TEM(NBNFC(JT))) -LT. .C2) GO TC 705
331 TEM(NBNFC(JT))=(TEM(NBNFC(JT))+F(NBNFC(JT)))/2.
332 NCFK=J
333 NCNTINLE
334 NCNTINLE C=1,NMF
335 COMB=C-F(JC)
336 TEM(JC)=COMB+1
337 NAX=ALC+ALC+1
338 LC 650 NCF=1,NMF
339 LC 651 NCF=ZACT(M1,M2)
340 ZACTINLE C=1,NMF
341 NCNTINLE C=1,NMF
342 NCNTINLE C=1,NMF
343 COMB=C-F(JC)
344 TEM(JC)=COMB+1
345 NAX=ALC+ALC+1
346 LC 650 NCF=1,NMF
347 LC 651 NCF=ZACT(M1,M2)
348 ZACTINLE C=1,NMF
349 NCNTINLE C=1,NMF
350 NCNTINLE C=1,NMF
351 COMB=C-F(JC)
352 TEM(JC)=COMB+1
353 NAX=ALC+ALC+1
354 LC 650 NCF=1,NMF
355 LC 651 NCF=ZACT(M1,M2)
356 ZACTINLE C=1,NMF
357 NCNTINLE C=1,NMF
358 NCNTINLE C=1,NMF
359 COMB=C-F(JC)
360 TEM(JC)=COMB+1
361 NAX=ALC+ALC+1
362 LC 650 NCF=1,NMF
363 LC 651 NCF=ZACT(M1,M2)
364 ZACTINLE C=1,NMF
365 NCNTINLE C=1,NMF
366 NCNTINLE C=1,NMF
367 COMB=C-F(JC)
368 TEM(JC)=COMB+1
369 NAX=ALC+ALC+1
370 LC 650 NCF=1,NMF
371 LC 651 NCF=ZACT(M1,M2)
372 ZACTINLE C=1,NMF
373 NCNTINLE C=1,NMF
374 NCNTINLE C=1,NMF
375 COMB=C-F(JC)
376 TEM(JC)=COMB+1
377 NAX=ALC+ALC+1
378 LC 650 NCF=1,NMF
379 LC 651 NCF=ZACT(M1,M2)
380 ZACTINLE C=1,NMF
381 NCNTINLE C=1,NMF
382 NCNTINLE C=1,NMF
383 COMB=C-F(JC)
384 TEM(JC)=COMB+1
385 NAX=ALC+ALC+1
386 LC 650 NCF=1,NMF
387 LC 651 NCF=ZACT(M1,M2)
388 ZACTINLE C=1,NMF
389 NCNTINLE C=1,NMF
390 NCNTINLE C=1,NMF
391 COMB=C-F(JC)
392 TEM(JC)=COMB+1
393 NAX=ALC+ALC+1
394 LC 650 NCF=1,NMF
395 LC 651 NCF=ZACT(M1,M2)
396 ZACTINLE C=1,NMF
397 NCNTINLE C=1,NMF
398 NCNTINLE C=1,NMF
399 COMB=C-F(JC)
400 TEM(JC)=COMB+1
401 NAX=ALC+ALC+1
402 LC 650 NCF=1,NMF
403 LC 651 NCF=ZACT(M1,M2)
404 ZACTINLE C=1,NMF
405 NCNTINLE C=1,NMF
406 NCNTINLE C=1,NMF
407 COMB=C-F(JC)
408 TEM(JC)=COMB+1
409 NAX=ALC+ALC+1
410 LC 650 NCF=1,NMF
411 LC 651 NCF=ZACT(M1,M2)
412 ZACTINLE C=1,NMF
413 NCNTINLE C=1,NMF
414 NCNTINLE C=1,NMF
415 COMB=C-F(JC)
416 TEM(JC)=COMB+1
417 NAX=ALC+ALC+1
418 LC 650 NCF=1,NMF
419 LC 651 NCF=ZACT(M1,M2)
420 ZACTINLE C=1,NMF
421 NCNTINLE C=1,NMF
422 NCNTINLE C=1,NMF
423 COMB=C-F(JC)
424 TEM(JC)=COMB+1
425 NAX=ALC+ALC+1
426 LC 650 NCF=1,NMF
427 LC 651 NCF=ZACT(M1,M2)
428 ZACTINLE C=1,NMF
429 NCNTINLE C=1,NMF
430 NCNTINLE C=1,NMF
431 COMB=C-F(JC)
432 TEM(JC)=COMB+1
433 NAX=ALC+ALC+1
434 LC 650 NCF=1,NMF
435 LC 651 NCF=ZACT(M1,M2)
436 ZACTINLE C=1,NMF
437 NCNTINLE C=1,NMF
438 NCNTINLE C=1,NMF
439 COMB=C-F(JC)
440 TEM(JC)=COMB+1
441 NAX=ALC+ALC+1
442 LC 650 NCF=1,NMF
443 LC 651 NCF=ZACT(M1,M2)
444 ZACTINLE C=1,NMF
445 NCNTINLE C=1,NMF
446 NCNTINLE C=1,NMF
447 COMB=C-F(JC)
448 TEM(JC)=COMB+1
449 NAX=ALC+ALC+1
450 LC 650 NCF=1,NMF
451 LC 651 NCF=ZACT(M1,M2)
452 ZACTINLE C=1,NMF
453 NCNTINLE C=1,NMF
454 NCNTINLE C=1,NMF
455 COMB=C-F(JC)
456 TEM(JC)=COMB+1
457 NAX=ALC+ALC+1
458 LC 650 NCF=1,NMF
459 LC 651 NCF=ZACT(M1,M2)
460 ZACTINLE C=1,NMF
461 NCNTINLE C=1,NMF
462 NCNTINLE C=1,NMF
463 COMB=C-F(JC)
464 TEM(JC)=COMB+1
465 NAX=ALC+ALC+1
466 LC 650 NCF=1,NMF
467 LC 651 NCF=ZACT(M1,M2)
468 ZACTINLE C=1,NMF
469 NCNTINLE C=1,NMF
470 NCNTINLE C=1,NMF
471 COMB=C-F(JC)
472 TEM(JC)=COMB+1
473 NAX=ALC+ALC+1
474 LC 650 NCF=1,NMF
475 LC 651 NCF=ZACT(M1,M2)
476 ZACTINLE C=1,NMF
477 NCNTINLE C=1,NMF
478 NCNTINLE C=1,NMF
479 COMB=C-F(JC)
480 TEM(JC)=COMB+1
481 NAX=ALC+ALC+1
482 LC 650 NCF=1,NMF
483 LC 651 NCF=ZACT(M1,M2)
484 ZACTINLE C=1,NMF
485 NCNTINLE C=1,NMF
486 NCNTINLE C=1,NMF
487 COMB=C-F(JC)
488 TEM(JC)=COMB+1
489 NAX=ALC+ALC+1
490 LC 650 NCF=1,NMF
491 LC 651 NCF=ZACT(M1,M2)
492 ZACTINLE C=1,NMF
493 NCNTINLE C=1,NMF
494 NCNTINLE C=1,NMF
495 COMB=C-F(JC)
496 TEM(JC)=COMB+1
497 NAX=ALC+ALC+1
498 LC 650 NCF=1,NMF
499 LC 651 NCF=ZACT(M1,M2)
500 ZACTINLE C=1,NMF
501 NCNTINLE C=1,NMF
502 NCNTINLE C=1,NMF
503 COMB=C-F(JC)
504 TEM(JC)=COMB+1
505 NAX=ALC+ALC+1
506 LC 650 NCF=1,NMF
507 LC 651 NCF=ZACT(M1,M2)
508 ZACTINLE C=1,NMF
509 NCNTINLE C=1,NMF
510 NCNTINLE C=1,NMF
511 COMB=C-F(JC)
512 TEM(JC)=COMB+1
513 NAX=ALC+ALC+1
514 LC 650 NCF=1,NMF
515 LC 651 NCF=ZACT(M1,M2)
516 ZACTINLE C=1,NMF
517 NCNTINLE C=1,NMF
518 NCNTINLE C=1,NMF
519 COMB=C-F(JC)
520 TEM(JC)=COMB+1
521 NAX=ALC+ALC+1
522 LC 650 NCF=1,NMF
523 LC 651 NCF=ZACT(M1,M2)
524 ZACTINLE C=1,NMF
525 NCNTINLE C=1,NMF
526 NCNTINLE C=1,NMF
527 COMB=C-F(JC)
528 TEM(JC)=COMB+1
529 NAX=ALC+ALC+1
530 LC 650 NCF=1,NMF
531 LC 651 NCF=ZACT(M1,M2)
532 ZACTINLE C=1,NMF
533 NCNTINLE C=1,NMF
534 NCNTINLE C=1,NMF
535 COMB=C-F(JC)
536 TEM(JC)=COMB+1
537 NAX=ALC+ALC+1
538 LC 650 NCF=1,NMF
539 LC 651 NCF=ZACT(M1,M2)
540 ZACTINLE C=1,NMF
541 NCNTINLE C=1,NMF
542 NCNTINLE C=1,NMF
543 COMB=C-F(JC)
544 TEM(JC)=COMB+1
545 NAX=ALC+ALC+1
546 LC 650 NCF=1,NMF
547 LC 651 NCF=ZACT(M1,M2)
548 ZACTINLE C=1,NMF
549 NCNTINLE C=1,NMF
550 NCNTINLE C=1,NMF
551 COMB=C-F(JC)
552 TEM(JC)=COMB+1
553 NAX=ALC+ALC+1
554 LC 650 NCF=1,NMF
555 LC 651 NCF=ZACT(M1,M2)
556 ZACTINLE C=1,NMF
557 NCNTINLE C=1,NMF
558 NCNTINLE C=1,NMF
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560 TEM(JC)=COMB+1
561 NAX=ALC+ALC+1
562 LC 650 NCF=1,NMF
563 LC 651 NCF=ZACT(M1,M2)
564 ZACTINLE C=1,NMF
565 NCNTINLE C=1,NMF
566 NCNTINLE C=1,NMF
567 COMB=C-F(JC)
568 TEM(JC)=COMB+1
569 NAX=ALC+ALC+1
570 LC 650 NCF=1,NMF
571 LC 651 NCF=ZACT(M1,M2)
572 ZACTINLE C=1,NMF
573 NCNTINLE C=1,NMF
574 NCNTINLE C=1,NMF
575 COMB=C-F(JC)
576 TEM(JC)=COMB+1
577 NAX=ALC+ALC+1
578 LC 650 NCF=1,NMF
579 LC 651 NCF=ZACT(M1,M2)
580 ZACTINLE C=1,NMF
581 NCNTINLE C=1,NMF
582 NCNTINLE C=1,NMF
583 COMB=C-F(JC)
584 TEM(JC)=COMB+1
585 NAX=ALC+ALC+1
586 LC 650 NCF=1,NMF
587 LC 651 NCF=ZACT(M1,M2)
588 ZACTINLE C=1,NMF
589 NCNTINLE C=1,NMF
590 NCNTINLE C=1,NMF
591 COMB=C-F(JC)
592 TEM(JC)=COMB+1
593 NAX=ALC+ALC+1
594 LC 650 NCF=1,NMF
595 LC 651 NCF=ZACT(M1,M2)
596 ZACTINLE C=1,NMF
597 NCNTINLE C=1,NMF
598 NCNTINLE C=1,NMF
599 COMB=C-F(JC)
600 TEM(JC)=COMB+1
601 NAX=ALC+ALC+1
602 LC 650 NCF=1,NMF
603 LC 651 NCF=ZACT(M1,M2)
604 ZACTINLE C=1,NMF
605 NCNTINLE C=1,NMF
606 NCNTINLE C=1,NMF
607 COMB=C-F(JC)
608 TEM(JC)=COMB+1
609 NAX=ALC+ALC+1
610 LC 650 NCF=1,NMF
611 LC 651 NCF=ZACT(M1,M2)
612 ZACTINLE C=1,NMF
613 NCNTINLE C=1,NMF
614 NCNTINLE C=1,NMF
615 COMB=C-F(JC)
616 TEM(JC)=COMB+1
617 NAX=ALC+ALC+1
618 LC 650 NCF=1
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[illegible]

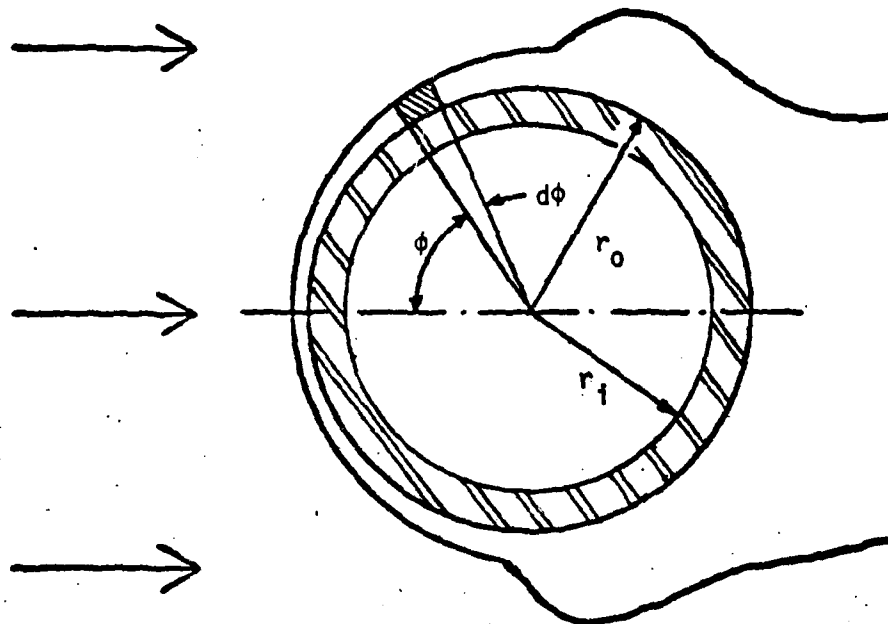
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 VVATC674C
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 VVATC680C
 VVATC681C
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 VVATC708C
 VVATC709C

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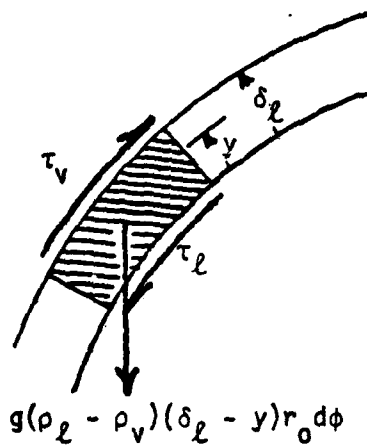
1+(Y(NCCCCF(JA))-Y(NCCCCF(JA+1)))**2))
1+5*(X(NCCCCF(JA-1))-X(NCCCCF(JA)))**2
1+(Y(NCCCCF(JA-1))-Y(NCCCCF(JA)))**2)
GC TC
AREAC(JA)=5*(DSCT((X(NCCCCF(JA-1))-X(NCCCCF(JA)))**2
1+(Y(NCCCCF(JA-1))-Y(NCCCCF(JA)))**2))
COUT(JA)=XKSCU*AREAC(JA)*CF/DIST
CTCTC=CTCTC+CCUT(JA)
ATCTC=ATCTC+AREAO(JA)
ATTCT=AREAC(JA)*TBAR(JA)+ATTCT
CONTINUE
ATCTC=
CTCTC=
CC=1,NCO
CCNAV(JE)=F*(NCELO(JE))*XL012(JB)*TBAR(JB)
CCNA(JE)=F*(NCELO(JE))*TEAR(JB)
ATCTC=ATCTC+XL012(JB)
CTCTC=CTCTC+CCNV(JE)
CONTINUE
WRITE(6,875) AREA GLT C CCND OUT LOCAL FLX ,
FCFMACTI , AREA CCNSEC
1AREA CCNV Q CCND IN Q CCND IN
CC 876 J=1,NCO
WRITE(6,877) AREA I(JC),CIN(JC),AREAO(JE),CCUT(JD),XL012(JC),QCCNV
1(JL)
CONTINUE
FORMAT(6,F12.6,1X),F12.2)
WRITE(6,882)
FCFMACTI
TCTALS , )
WRITE(6,879) ATOTI,CTCTI,ATOTO,QIOTO,ATCTC,CTCTC
TAVG=ATCTI/ATOTO
WRITE(6,880) TAVG
WRITE(6,881) TAVG
FORMAT(6,AVERAGE WALL TENF = ,F10.5)
RETURN
END
  
```

551
 952
 544
 880
 875
 878
 877
 882
 400

APPENDIX C



(a)



(b)

Figure 1. Schematic diagrams of physical model.

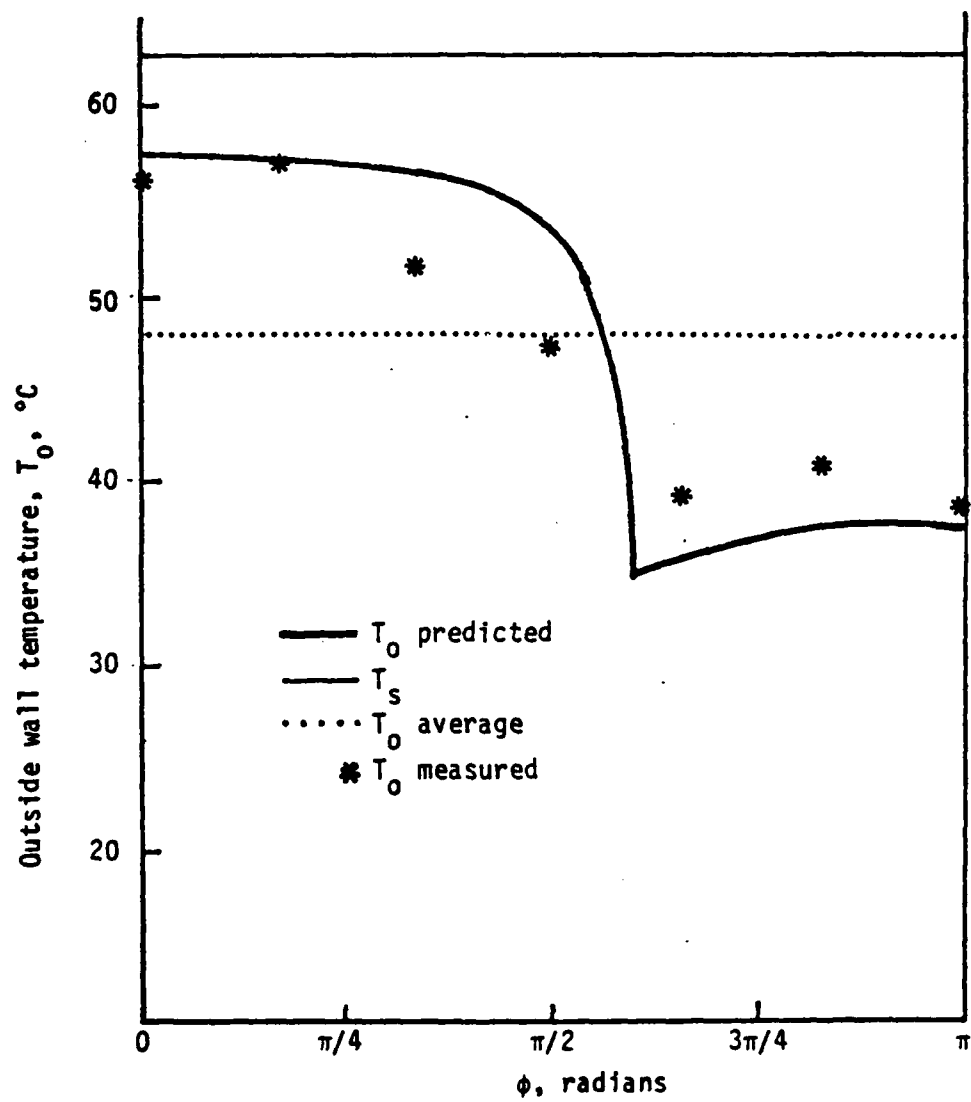


Figure 2. Predicted and measured T_o versus angle from stagnation point [18].

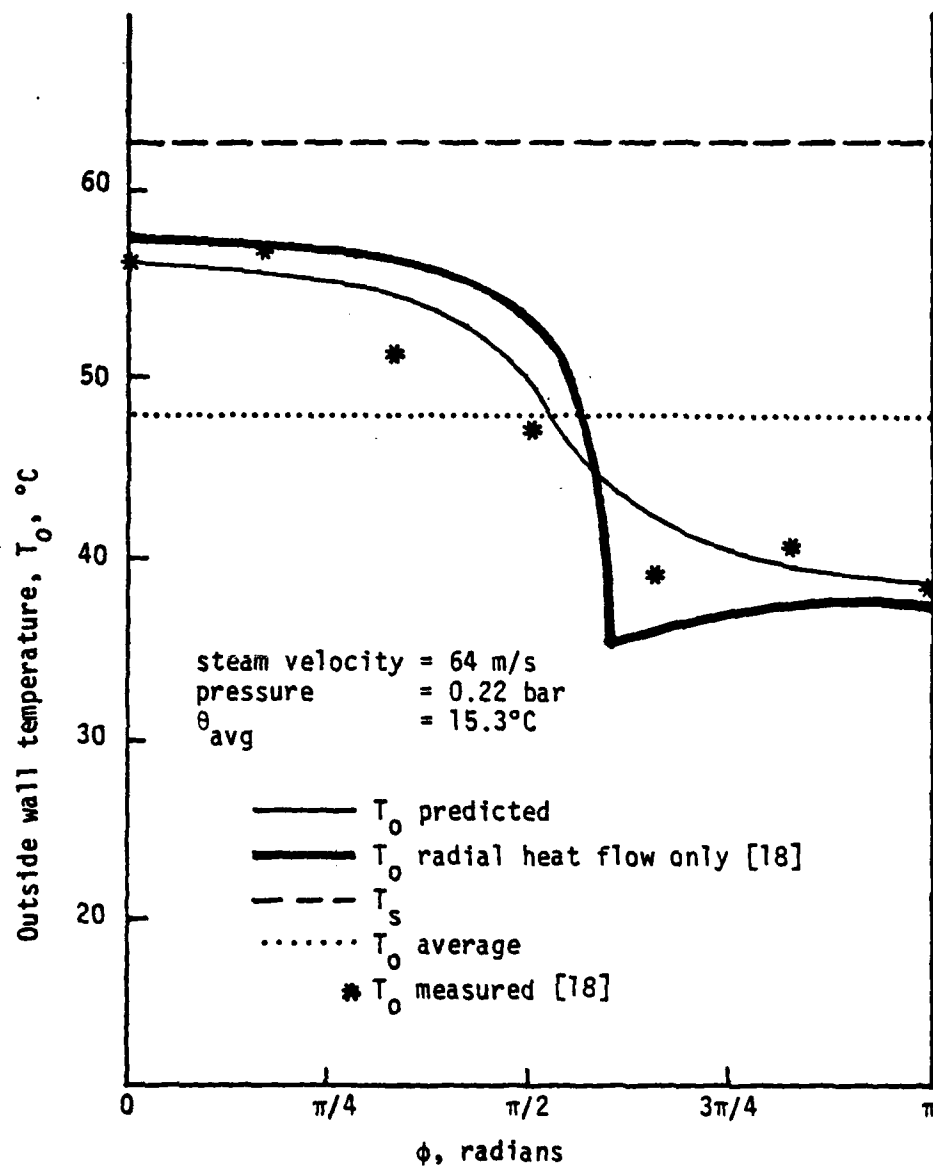


Figure 3. Predicted and measured T_o versus angle from forward stagnation point.

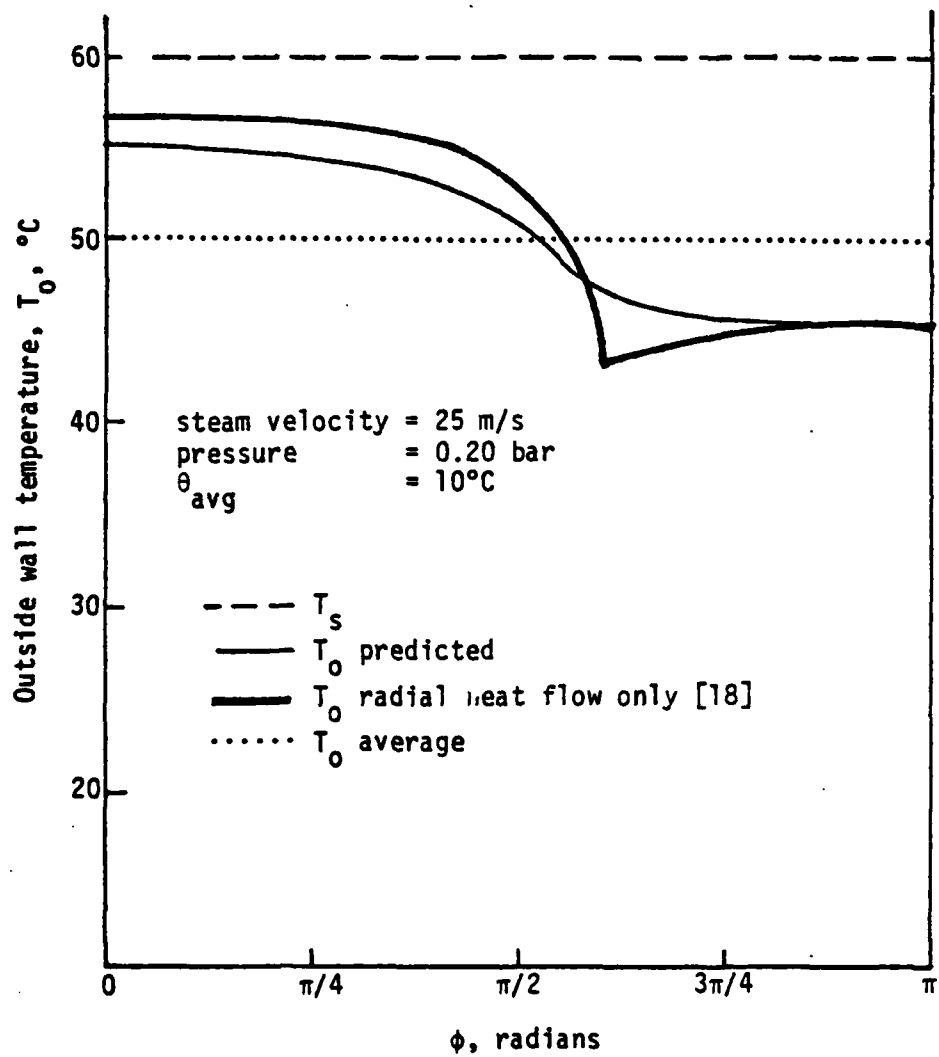


Figure 4. Predicted T_o versus angle from forward stagnation point.

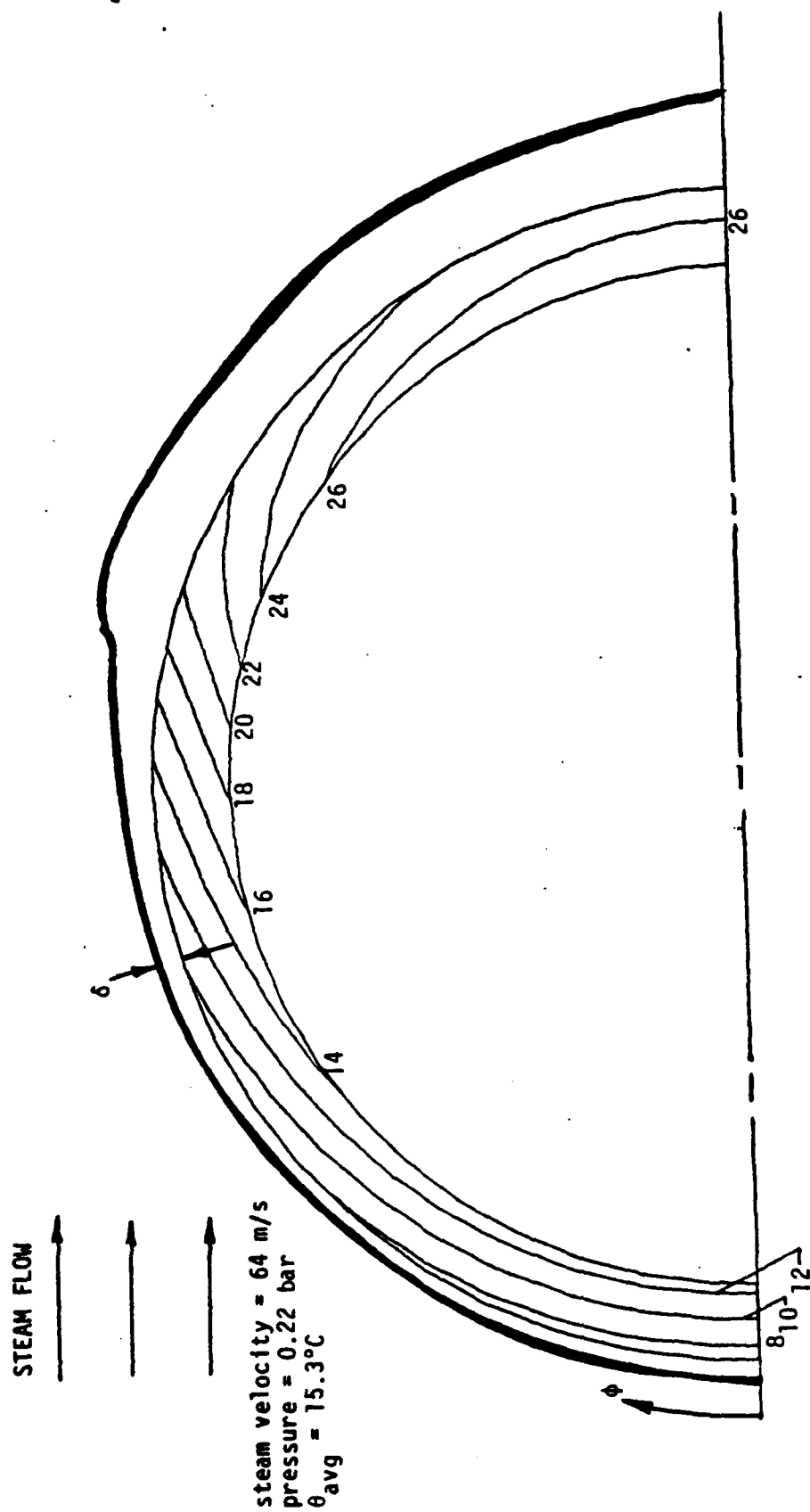


Figure 5. Isotherms (in degrees Celsius) of θ inside tube wall for top half of tube. Thickness of condensate film is shown with a magnification of 25 times.

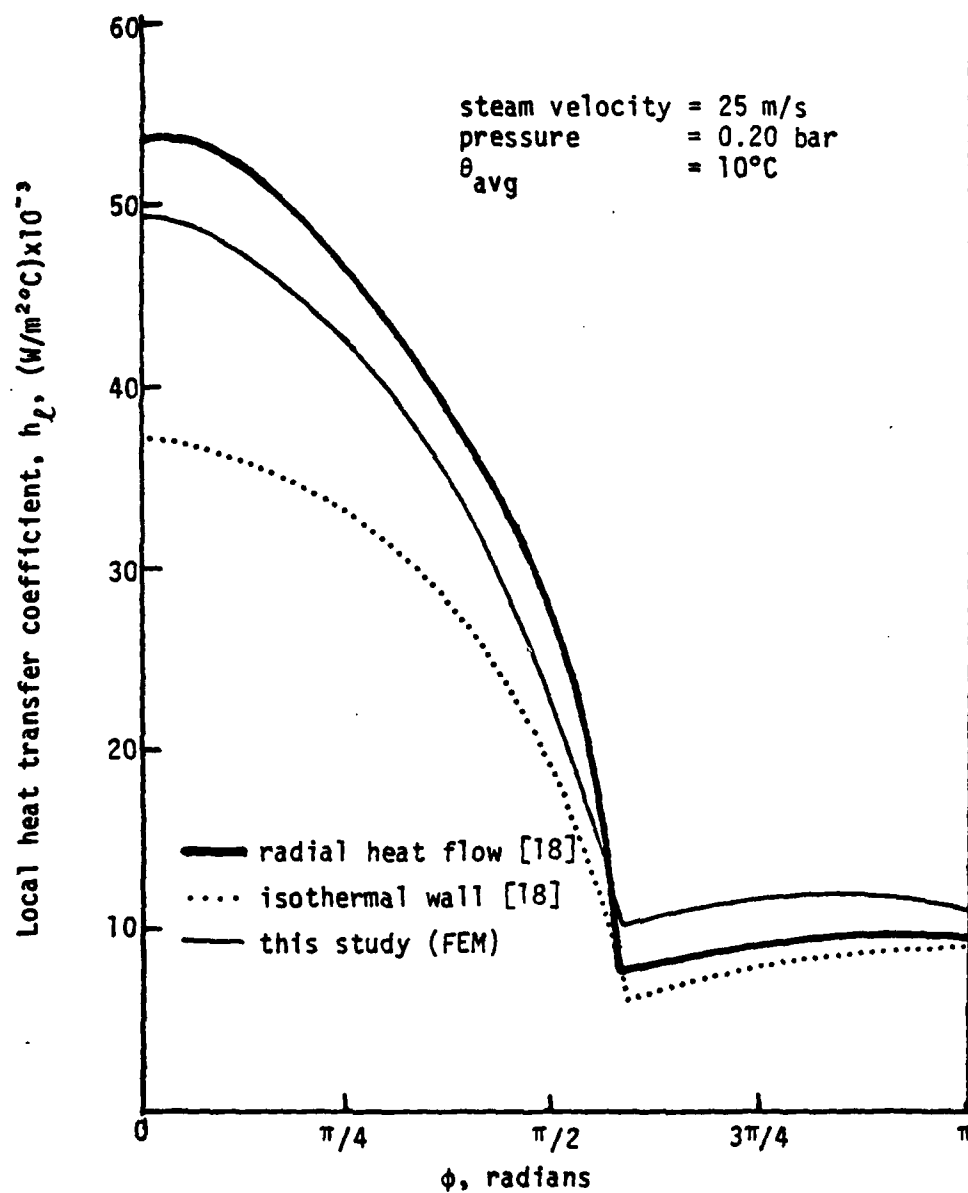


Figure 6. Predicted local heat transfer coefficient versus angle from forward stagnation point.

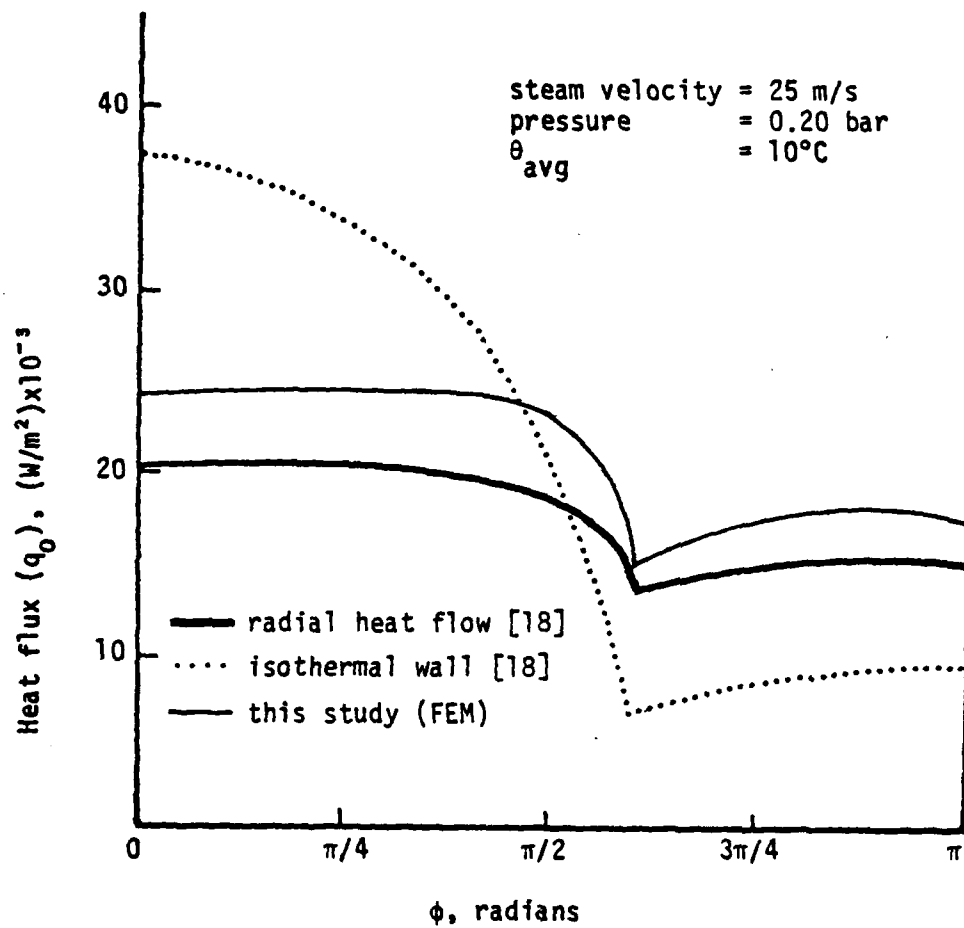


Figure 7. Predicted local heat flux versus angle from forward stagnation point.

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FILM**