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NATURAL CIRCULATION IN HORIZONTAL PIPES

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Abstract—Natural circulation in a horizontal pipe with one end connected to the hot main pipe and one end closed can create a substantial temperature gradient axially and circumferentially. The temperature distribution around such a pipe can be predicted by solving the conservation equations inside the pipe and energy equation around the pipe. The line SOR (successive over relaxation) method has been employed to solve these equations. This numerical solution has been confirmed by comparing it to an exact solution for the simplified case where the heat transfer coefficient around the pipe is uniform.

Pipe temperature variations predicted by this analysis have been characterized by five dimensionless parameters. Of these five, the parameter λ, which is related to the Biot number, is the controlling parameter. The maximum temperature gradient around the tube increases sharply as λ increases. Applications of this solution are discussed and a set of recommendations to reduce temperature gradient are presented.

NOMENCLATURE

$\mathbf{A}$, matrix;
a, tube radius [m];
a_i, b_i, c_i, d_i, elements of vector;
Bi, Biot number, $\dfrac{ha}{K_w}$;
C, axial pressure gradient, $\dfrac{a^2}{\mu W}\dfrac{\partial P}{\partial Z}$, [dimensionless];
C_1, constant axial temperature drop, $Re_a Pr a \dfrac{\Delta T^*}{\Delta Z}$ [K];
C_3, dimensionless wall parameter, $\dfrac{K_f a}{K_w t}$;
C_4, dimensionless heat loss parameter, $\dfrac{a^2}{K_w t}\dfrac{q_0''}{C_1}$;
C_p, constant pressure specific heat [J/kg K];
D, inside tuber diameter [m];
f, dummy variable;
h, heat-transfer coefficient [W/m^2 K];
K_f, fluid thermal conductivity [W/m K];
K_w, tube wall thermal conductivity [W/m K];
M, number of divisions in R-direction;
N, number of divisions in θ-direction;
Nu, local circumferential Nusselt number, $(ha)/K_f$;
P, pressure [N/m^2];
Pe, Peclet number $Re_a Pr$;
Pr, Prandtl number $(\mu C_p/K_f)$;
q'', rate of heat transfer per unit area [W/m^2];
q_0'', amplitude of rate of heat loss through outside of the tube wall [W/m^2];
R, θ, Z, dimensional cylindrical coordinates;
r, θ, z, dimensionless cylindrical coordinates;
Re_a, Reynolds number based on radius, $\dfrac{\rho \overline{W} a}{\mu}$;
T, local fluid or wall temperature [K];
T_1, top wall temperature [K];
T_2, bottom wall temperature [K];
T_b, bulk average temperature [K];
T_r, reference temperature [K];
ΔT, circumferential temperature drop between the top and bottom of the wall $T_1 - T_2$ [K];
ΔT^*, axial temperature drop between hot and cold ends, or constant axial temperature drop in fully developed region [K];
t, tube wall thickness [m];
W, dimensional axial velocity [m/s];
w, dimensionless axial velocity, $W/\overline{W}$;
$\overline{W}$, average axial velocity in upper or bottom portion of the tube [m/s];
X, dimensional circumferential direction;
Y, dimensionless temperature $(T-T_r)/\Delta T_r$ defined in equation (13).

Greek symbols

θ, angle measured from top of the tube, degree;
θ_0, boundary for regions 1 and 2 shown in Fig. 5;
λ, dimensionless parameter, $\left[\dfrac{ha}{K_w}\dfrac{a}{t}\right]^{1/2}$ or $\left[Bi\dfrac{a}{t}\right]^{1/2}$;
μ, viscosity [Ns/m^2];
ρ, density [kg/m^3];
ϕ, dimensionless temperature, $\dfrac{T-T_r}{C_1}$ or $\dfrac{T-T_2}{T_1-T_2}$;
∇, two-dimensional Laplacian operator $\dfrac{\partial^2}{\partial r^2}+\dfrac{1}{r}\dfrac{\partial}{\partial r}+\dfrac{1}{r^2}\dfrac{\partial^2}{\partial \theta^2}$.

Subscripts

b, bulk mean value;
f, property for fluid;
i, j, space subscripts of grid points in R and θ direction;
w, property for wall.

INTRODUCTION

NATURAL circulation in a horizontal pipe occurs when one end is closed and the other end is connected with another pipe where the hot fluid is passing. The hot fluid from the flowing pipe moves into the upper portion of the closed pipe toward the closed cold end and cooler fluid flows out the lower portion of the closed pipe toward the flowing pipe. In a typical application, the closed end of the pipe is usually the location of a valve. The frequent opening and closing of the valve can create a significant thermal stress in the valve as well as in the pipe owing to the temperature difference between the main pipe and the closed end.

There will also be top-to-bottom temperature differences found in these horizontal, dead-leg pipes. This temperature gradient is also produced by the natural circulation flow. The detailed temperature profiles around such a pipe and the temperature differences between the main pipe and the cold ends are required for pipe design, thermal stress analysis and valve failure analysis.

It was observed in a plastic model test [1] that the upper fluid which is warmer tends to move toward the dead leg and the colder, bottom portion of the fluid moves to the main pipe. A pressure drop between the two ends is created by a temperature-induced density gradient along the pipe. There is, of course, an entrance length which allows the natural circulation flow to develop and a region close to the end of the dead leg where the temperature variations around the tube wall are nearly uniform. The flow pattern observed in these regions is three dimensional and the solution is complicated. The successful handling of such a flow is difficult by considering the natural convection in the vertical direction so that flow becomes three dimensional. The present analysis will be concerned with the region where the greatest temperature gradients are observed and the entrance effect will be ignored.

The velocity component in the vertical direction observed is much smaller than the velocity component in the horizontal direction generated by the natural circulation. The free convection effect in the vertical direction will be ignored and the problem is simplified as forced convection in a horizontal pipe.

Since the top portion and the bottom portion of the fluid flow in opposite directions, one can imagine that there is a solid wall across the horizontal centerline, and the flow in the pipe functions as a heat exchanger. It is noted from both visual observation and velocity measurements that the fluid flows in these two portions are laminar. One can thus simulate the flow pattern as laminar flow in a semi-circular tube.

Thermally and hydrodynamically fully developed flow in semi-circular tubes has been solved by Eckert *et al.* [2] and Sparrow and Haji-Sheikh [3]. Extensive investigation of laminar flow heat transfer in ducts of various shapes has been considered by Shah and London [4] for different kinds of thermal boundary conditions. Recently, Hong and Bergles [5] considered the entrance effect of semi-circular tubes with uniform heat flux. The three-dimensional thermal entrance region problem in the circular tubes with secondary flow effect has been solved numerically by Hong *et al.* [6, 7], for uniform wall heat flux. However, none of these existing solutions can be applied directly to the present problem. For the particular problem at hand, the results of two-dimensional analysis lead one to a further simplification of the problem. The final solution is obtained by solving an equation identical to that obtained for the one-dimensional conduction fin problem.

FORMULATION OF THE PROBLEM

The system of coordinates of steady laminar flow in the fully developed region of a horizontal tube is shown in Fig. 1. To facilitate analysis, the following assumptions are made:

1. Physical properties of the fluid and tube wall are considered to be constant so that neither free convection in the vertical direction nor temperature-dependent viscosity effects are taken into account.
2. The axial pressure gradient is constant.
3. The axial temperature gradients in the top and bottom portions of fluid flow are the same in magnitude and are constant.

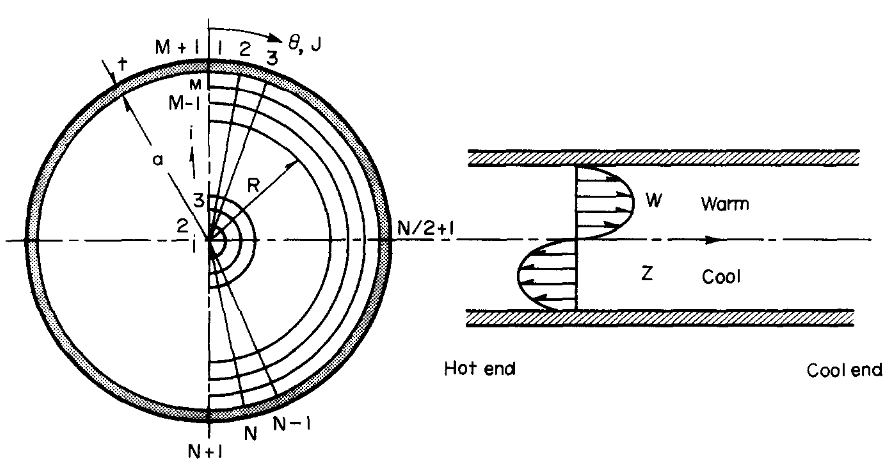

FIG. 1. Coordinate system and numerical grid for horizontal circular pipe.

4. The axial and radial conduction in the tube wall can be neglected.
5. The axial conduction in the fluid can be neglected.
6. The fluid flow in the upper and lower portions of the pipe is identical except in direction and is two dimensional. The free convection in the vertical direction is ignored.

When cylindrical coordinates (R, θ, Z) as shown in Fig. 1 are introduced and the assumptions stated above are applied, the governing differential equations are:

Momentum equation

$$\frac{\partial^2 W}{\partial R^2}+\frac{1}{R}\frac{\partial W}{\partial R}+\frac{1}{R^2}\frac{\partial^2 W}{\partial \theta^2}=\frac{1}{\mu}\frac{\partial P}{\partial Z} \quad (1)$$

Energy equation

$$\left(\frac{\rho C_p}{K_f}\right) W\frac{\partial T}{\partial Z}=\frac{\partial^2 T}{\partial R^2}+\frac{1}{R}\frac{\partial T}{\partial R}+\frac{1}{R^2}\frac{\partial^2 T}{\partial \theta^2} \quad (2)$$

Conduction equation at wall

$$\frac{tK_w}{a^2}\frac{\partial^2 T}{\partial \theta^2}=K_f\frac{\partial T}{\partial R}\bigg|_w+q''(\theta). \quad (3)$$

Since the thermal and hydrodynamic conditions are symmetric with respect to the vertical center line, one needs to consider only half of the circular geometry. To normalize the governing partial differential equations the following dimensionless variables, constants, and parameters were used:

Dimensionless radius $r = R/a$

Dimensionless velocity $w = W/\bar{W}$

Reynolds number based on radius $Re_a = \dfrac{\rho \bar{W} a}{\mu}$

Prandtl number $Pr = \dfrac{\mu C_p}{K_f}$

Peclet number $Pe = Re_a Pr$

Dimensionless pressure drop constant

$$C = \frac{a^2}{\mu \bar{W}}\frac{\partial P}{\partial Z}$$

Dimensionless temperature

$$\phi = \frac{T-T_r}{Pe\dfrac{a}{\Delta Z}\Delta T^*}=\frac{T-T_r}{C_1}$$

Axial temperature gradient parameter

$$C_1 = Pea\frac{\Delta T^*}{\Delta Z}$$

where T_r is the reference temperature and is chosen to be the overall average fluid bulk temperature at one cross section. The momentum and energy equation for the fluid and conduction equation for the pipe wall can thus be simplified to the following dimensionless forms.

$$\nabla^2 w = C \quad (4)$$

$$\nabla^2 \phi = -w \quad (5)$$

$$\frac{\partial^2 \phi}{\partial \theta^2}\bigg|_{r=1} = C_3\frac{\partial \phi}{\partial r}\bigg|_{r=1}+C_4 q(\theta) \quad (6)$$

where

$$C_3 = \frac{K_f}{K_w}\frac{a}{t}$$

$$C_4 = \frac{a^2 q_0''}{K_w t C_1} \quad (7)$$

are constants and

$$\nabla^2 = \frac{\partial^2}{\partial r^2}+\frac{1}{r}\frac{\partial}{\partial r}+\frac{1}{r^2}\frac{\partial^2}{\partial \theta^2}$$

is the Laplacian operator. $q(\theta)$ is assumed to be a given function for heat loss through the outside of the tube wall and can be estimated to be

$$q(\theta) = 1+\tfrac{1}{2}\cos\theta. \quad (8)$$

The boundary conditions for equations (4)–(6) are as follows:

at wall, $r = 1$, $w = 0$

at vertical centerline, $\theta = 0$ or π

$$\frac{\partial \phi}{\partial \theta}=\frac{\partial w}{\partial \theta}=0 \quad (9)$$

at horizontal centerline, $\theta = \pi/2$, $W = 0$.

It is noted that equations (4) and (5) are Poisson's equation and can be solved by the Successive Over-Relaxation (SOR) method. The details of the SOR method are given in [7] and [8].

The heat loss term indicated in equation (7) is assumed and can be estimated as:

$$q_0'' = h(T_0-T_a) = 0.25\left(\frac{T_0-T_a}{D}\right)^{1/4}(T_0-T_a) \quad (10)$$

where T_0 and T_a are the wall and atmospheric temperature, respectively. It is clear that $C_4 = 0$ corresponds to a perfect insulated wall. A computer code was written to perform the calculation by using a Honeywell 6000 computer. A mesh size of 20 by 20 was employed.

The local Nusselt number can be obtained by considering the temperature gradient at the wall and the local heat transfer coefficient.

The result is:

$$Nu = \frac{\dfrac{\partial \phi}{\partial r}\bigg|_{r=1}}{\phi_w}. \quad (11)$$

NUMERICAL RESULTS

(a) *Velocity and temperature profiles*

For fully developed flow with uniform axial temperature gradient, the thermal and hydraulic fields depend on parameters C_1, C_3, and C_4 [as demonstrated in governing equations (4)–(6)]. The controlling parameters for dimensionless temperature profiles are C_3 and C_4. C_1 is a scale factor to determine actual temperature profile and is determined by the temperature difference between hot and cold ends.

The dimensionless velocity profiles do not depend on any of the parameters stated above. However, as pointed out by Klepfer *et al.* [1], the maximum velocity at the upper or lower portion of the fluid flow depends on the temperature drop between the hot and cold ends.

(b) *Wall temperature*

As indicated in equation (6), the wall temperature is obtained by considering circumferential heat conduction in the pipe wall, heat transfer through the fluid inside the pipe, and heat loss outside of the pipe wall. The dimensionless wall temperature profile around the wall circumference with C_3 as a parameter is shown in Fig. 2 for the case when the outside wall is insulated

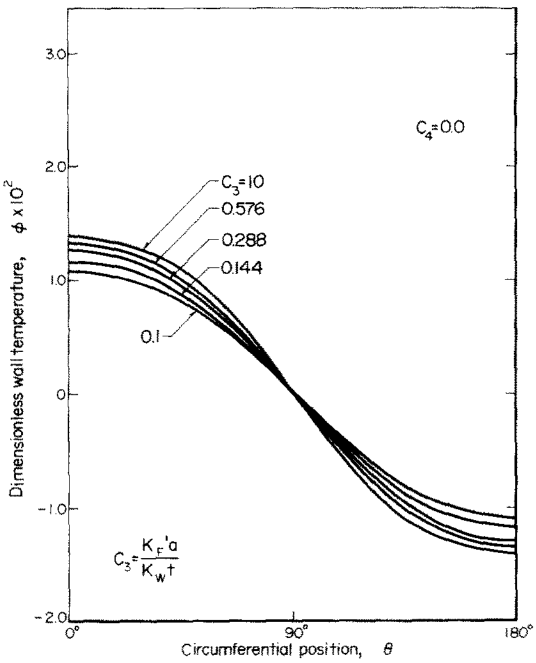

FIG. 2. C_3 effects on dimensionless wall temperature profiles.

($C_4 = 0$). Increasing C_3 tends to increase the temperature gradient around the wall. There is no great change in wall temperature profile when C_3 is larger than 10. Those profiles are seen to be symmetric with respect to $\theta = 90°$ and are similar to the cosine functions. The top and bottom wall temperature difference increases as C_3 increases. The reference temperature T_r appears at the horizontal centerline or at the wall located at $\theta = \pi/2$.

The heat loss, associated with C_4 parameters, effect on the wall temperature profile is shown in Fig. 3. Increasing heat loss tends to reduce the top and bottom temperature difference. It is seen that wall temperature profiles appear to be similar for different values of C_4. The location of the average fluid temperature T_r moves toward the top as C_4 increases.

The local heat-transfer coefficients associated with Nusselt number can be obtained from equation (11). Figure 4 shows the variation of the local Nusselt number with C_3 as the parameter. The local Nusselt number increases as C_3 decreases and has a maximum value at $\theta = \pi/2$. For a larger C_3 (greater than 10), the heat-transfer coefficient is uniform around the tube

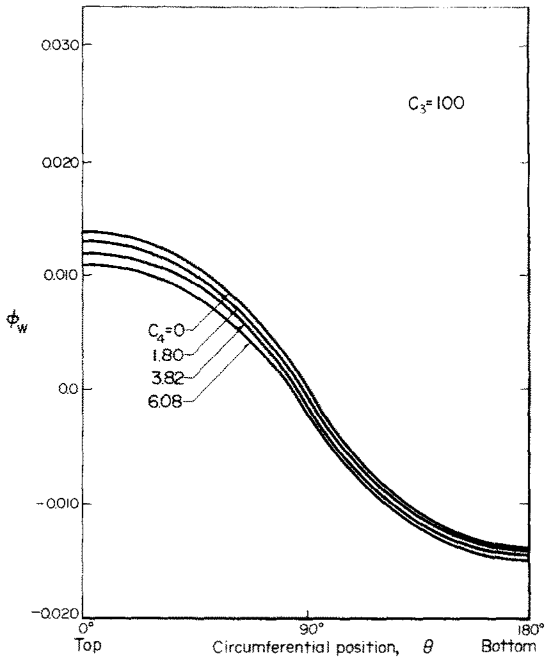

FIG. 3. Heat loss effects on dimensionless wall temperature profiles.

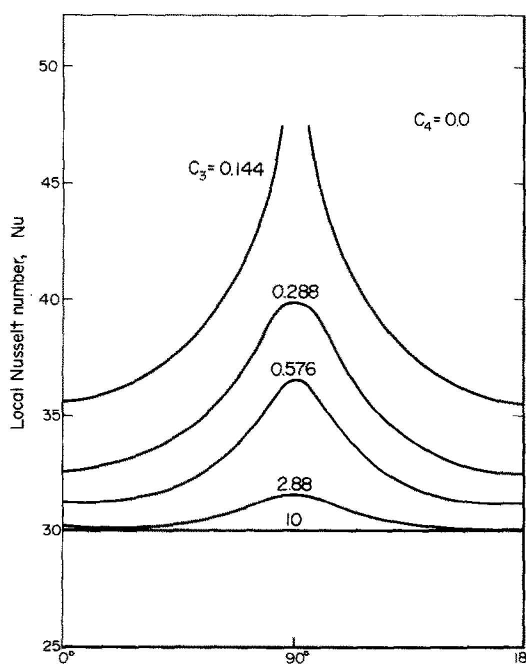

FIG. 4. C_3 effects on local Nusselt number.

circumference. This is an important result which leads one to further simplify the problem to the point where an exact solution can be obtained. A detailed discussion of this solution will be given in the next section.

EXACT SOLUTION FOR THE TEMPERATURE VARIATION AROUND THE PIPES

(a) *Background*

The problems involved in predicting the wall temperature variation around a pipe has been well defined and solved numerically in previous sections. However, the assumptions made in previous sections may not be applicable to the real problems. The two main restrictions stated in the previous numerical solutions are: flow is laminar and the upper portion of the warm fluid blow and the bottom portion of the cold flow are separated at the horizontal centerline. These assumptions were guided by the visual observations described by Klepfer *et al.* [1]. In the real situation, the Reynolds number based on diameter may exceed the laminar flow limit due to large average velocity and low viscosity and the flow may become turbulent. In addition, there is no strong evidence nor measurement of the location of zero velocity plane inside the pipe. The actual location of this separation line between warm and cold fluid may depend on axial location, the strength of the turbulence generated in the entrance, and on the temperature drop between the hot and cold ends. It may not even be a horizontal line. In order to simplify the problem and determine which major parameter controls the wall temperature profiles, the conduction equation (6) will be modified.

(b) *Formulation and solution*

Since tube wall thickness is often small compared with tube diameter, temperature variation in the radial direction across the wall can be ignored in most cases. Also, the temperature gradient in the axial direction is usually much smaller than the temperature gradient in the circumferential direction and thus axial conduction can also be neglected. Considering the tube wall as a flat plate, a simplified coordinate system for tube circumference is illustrated in Fig. 5. As previously indicated, the heat loss exterior to the tube wall does not

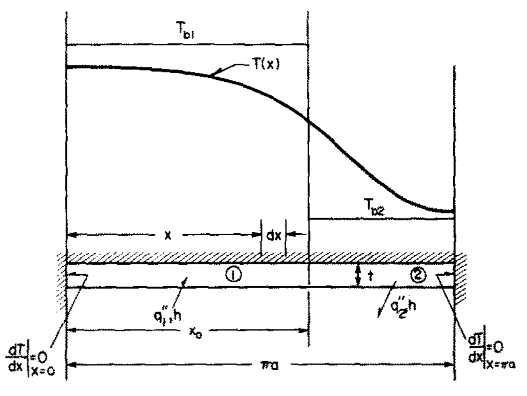

FIG. 5. Simplified coordinate system for tube circumference.

have a significant effect on wall temperature profile and can be neglected. It is further assumed that this flat plate is exposed to a fluid flow with bulk temperature of T_{b1} for $X \leqslant X_0$ and of T_{b2} for $X \geqslant X_0$; where X_0 represents the boundary of warm and cool fluid inside the tube. It has been shown earlier, Fig. 4, that the heat-transfer coefficient around the tube wall is uniform for large C_3. When the assumptions stated above are applied, the one-dimensional energy equations at the wall become:

$$
\begin{aligned}
K_w \cdot t \cdot \frac{d^2T}{dx^2} &= h(T - T_{b1}) \text{ for } X \leqslant X_0 \\
K_w t \frac{d^2T}{dx^2} &= h(T - T_{b2}) \text{ for } X \geqslant X_0.
\end{aligned} \tag{12}
$$

Equation (12) is the same as the differential equation for the fin problem [9], and is similar to equation (6). Equation (12) can be simplified by introducing the following dimensionless parameters:

$$
\begin{aligned}
y &= (T - T_r)/\Delta T_r \\
X &= a \cdot \theta \\
\lambda^2 &= \frac{ha}{K_w} \frac{a}{t}.
\end{aligned} \tag{13}
$$

The exact solution of equation (12) can be obtained by considering the symmetric condition at $\theta = 0$ and $\theta = \pi$, the continuous condition at $X = X_0$ and the dimensionless parameter shown in equation (13). The results are:

$$
\begin{aligned}
y_1 &= \frac{1}{\sinh \lambda\theta_0} [\sinh \lambda\pi - \sinh \lambda(\pi - \theta_0) \cosh \lambda\theta] \quad \theta \leqslant \theta_0 \\
y_2 &= \cosh \lambda(\pi - \theta) \text{ for } \theta \geqslant \theta_0
\end{aligned} \tag{14}
$$

where T_r is the reference temperature. ΔT_r is some reference temperature difference to be determined and y_1 and y_2 stand for $(T - T_r)/\Delta T_r$ in regions 1 and 2, respectively.

Equation (14) indicates that this dimensionless temperature profile has a maximum value at the top of the tube and a minimum at the bottom.

To actually predict the dimensional temperature profile one has to estimate the numerical value for λ and θ_0 and then obtain T_r and ΔT_r with any two measured temperatures around the tube wall. Since the top and bottom temperatures in most cases are considered to be given, it is thus convenient to normalize equation (14) with given top and bottom temperatures as follows:

$$
\phi = \frac{T - T_2}{T_1 - T_2} \tag{15}
$$

where T_1 and T_2 are wall temperatures at top and bottom, respectively. The solution for ϕ is:

$$
\phi = \frac{y - 1}{y(0) - 1} \tag{16}
$$

where y is given in equation (14) and

$$
y(0) = \frac{1}{\sinh \lambda\theta_0} [\sinh \lambda\pi - \sinh \lambda(\pi - \theta_0)].
$$

It is easily seen that ϕ varies between 0 and unity.

(c) *Analytical results*

Two major parameters are involved in the present exact solution of the simplified problem. They are λ which is related to Biot number and θ_0 which is the location where the warm and cool fluid separates inside the tube. Figure 6 shows the effect of parameter θ_0 on dimensionless wall temperature distribution for

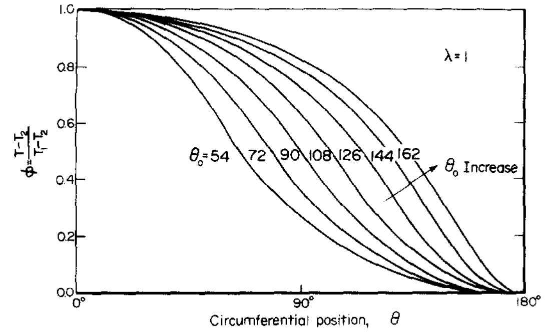

FIG. 6. Parameter θ_0 effects on dimensionless wall temperature distribution.

fixed λ. Increasing θ_0 (which means increasing the cross-section area occupied by warm fluid at the top portion of the tube) tends to increase ϕ and decrease the temperature gradient at the top portion of the tube. It can also be seen that θ_0 has no significant effect on the magnitude of maximum temperature gradient. The point of inflection as illustrated in Fig. 6 moves toward the bottom of the tube as θ_0 increases. For $\theta_0 = 90^\circ$ the temperature profile is symmetric with respect to $\theta = 90^\circ$. This result agrees with the previous numerical solution.

The effect of parameter λ on temperature profiles for fixed θ_0 are illustrated in Fig. 7. It is clear that the maximum temperature gradient increases as λ increased. For a strong circulation, one expects that the heat-transfer coefficient will be increased and thus λ will be increased. With a large value of λ the temperature gradient around the tube wall will be increased in order to carry more heat through the tube wall. The location of the maximum temperature gradient is only slightly affected by λ.

A comparison of the present analytical results with

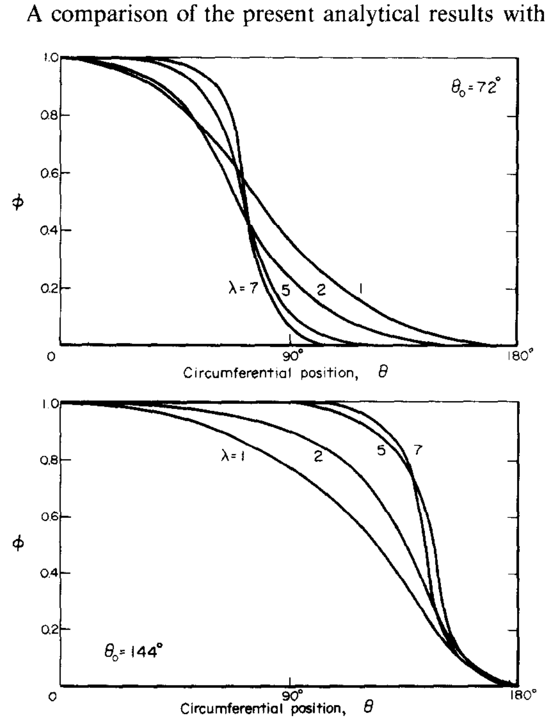

FIG. 7. Parameter λ effects on dimensionless wall temperature profiles with $\theta = 72^\circ$ and 144°.

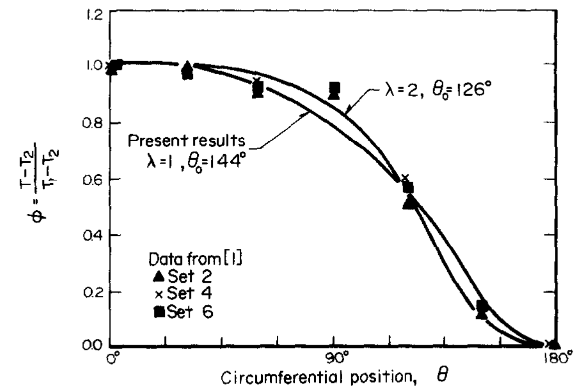

FIG. 8. Comparison of the present analytical results with experimental data from a plastic model test.

experimental data from a plastic model test [1] is shown in Fig. 8. Two sets of λ and θ_0 were chosen to predict the data. It is seen that the numerical value for λ in the plastic model test is in the neighborhood of 1 and 2, and θ_0 is about 125°. In the applications of this analysis to the case of a bypass line around a valve in the recirculation line of a nuclear reactor the numerical value of λ may be as high as 7.

CONCLUSIONS AND REMARKS

1. Thermal analysis of the temperature variation around a closed-end, horizontal pipe can be achieved by solving the coupled fluid energy equation and the pipe wall conduction equation, provided that the fluid momentum equation can be solved separately from the energy equation. This two-dimensional flow analysis seems to be sufficient to describe the problem. Further simplifying assumptions reduce the problem to one whose solution is the same as that for the one-dimensional fin analysis for prediction of the wall temperature profiles. The solution to this problem has been shown to depend on five major parameters, i.e. C_1, C_3, C_4, λ and θ_0.

2. These parameters are all dimensionless except C_1 which is the scale of temperature profile. The absolute value of temperature drop between top and bottom of the tube wall increases as C_1 increases. A tube with poor thermal conductivity or large tube diameter and thickness ratio (with large C_3) tends to increase the temperature gradient around the wall. Heat loss through the outside of the tube wall, a large value of C_4, is seen to decrease the temperature drop. The maximum temperature gradient around the wall is controlled mainly by the parameter λ. Increasing λ will increase the maximum temperature gradient. The location of the maximum temperature gradient moves toward the bottom of the tube as θ_0 increases. The numerical value of λ in the test to which these results were compared was around 2. In practical application, λ may be as high as 7.

3. One way to prevent high circumferential temperature gradients is to eliminate the natural circulation inside the pipe. This could be accomplished by allowing a small leakage or by installing a device to promote turbulence inside the pipe. For an existing system, a small axial temperature drop, high thermal conduc-

tivity of the pipe wall, small diameter-to-thickness ratio and a large heat-transfer coefficient on the outside of the pipe will all help to reduce the circumferential temperature gradient inside the pipe. Figure 7 can be utilized for stress analysis.

4. Although this study seems to clarify some of the questions related to temperature variation around the pipe, some problems still remain unsolved. The heat-transfer coefficient in parameter λ needs to be determined experimentally by using full-scale steel pipes. The conclusions reached from this preliminary study need further experimental verification. More extensive investigations experimentally and theoretically are required to fully understand the mechanism involved in this problem. An interesting and useful extension of this analysis would be to determine the two-dimensional temperature profiles in the pipe wall.

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CONVECTION NATURELLE DANS LES TUBES HORIZONTAUX

Résumé—La convection naturelle dans un tube horizontal dont une extrémité est réunie au tube chaud principal et l'autre extrémité fermée peut créer un gradient de température important à la fois dans le sens axial et le sens circonférenciel. La distribution de température autour d'un tel tube peut être calculée en résolvant les équations de conservation à l'intérieur du tube et l'équation d'énergie autour du tube. Ces équations ont été résolues à l'aide de la méthode des séries SOR (successive sur relaxation). Cette solution numérique a été testée en comparant ses résultats avec la solution exacte dans le cas simplifié où le coefficient de transfert de chaleur autour du tube est constant.

Les variations de température du tube calculées par cette méthode ont été caractérisées par cinq paramètres adimensionnels. Sur ces cinq paramètres, le paramètre λ, qui est lié au nombre de Biot est le paramètre déterminant. Le gradient maximum de température autour du tube augmente fortement avec λ. Des applications de cette méthode sont discutées et on présente un ensemble de recommandations permettant de réduire le gradient de température.

NATÜRLICHE KONVEKTION IN HORIZONTALEN ROHREN

Zusammenfassung—Die natürliche Konvektion in einem horizontalen Rohr, von dem ein Ende mit einer heißen Hauptrohrleitung verbunden ist, während das andere Ende verschlossen ist, kann erhebliche Temperaturgradienten sowohl in axialer Richtung wie in Umfangsrichtung hervorrufen. Die Temperaturverteilung um ein solches Rohr kann aus der Lösung der Erhaltungsgleichungen für den Innenrohrbereich und der Energiegleichung für den Bereich um das Rohr herum ermittelt werden. Zur Lösung wurde die SOR-Methode (schrittweise Überrelaxation) verwendet. Diese numerische Lösung wurde durch Vergleich mit der exakten Lösung für den vereinfachten Fall eines einheitlichen Wärmeübergangskoeffizienten an der Außenseite des Rohres bestätigt.

Die analytisch ermittelten Veränderungen der Rohrtemperatur wurden mit Hilfe von 5 dimensionslosen Kennzahlen erfaßt. Von diesen 5 Kennzahlen dominiert der Parameter λ, der mit der Biot–Zahl verknüpft ist. Der max. Temperaturgradient um das Rohr nimmt mit wachsendem λ stark zu. Es werden Anwendungen dieser Lösung diskutiert und Empfehlungen zur Verringerung des Temperaturgradienten gegeben.

ЕСТЕСТВЕННАЯ ЦИРКУЛЯЦИЯ В ГОРИЗОНТАЛЬНЫХ ТРУБАХ

Аннотация — Естественная циркуляция в горизонтальной трубе, открытым концом подсоединенной к горячему трубопроводу, может служить причиной возникновения значительных температурных градиентов по её оси и окружности. Распределение температуры вокруг такой трубы можно рассчитать, используя уравнение сохранения для внутренней области, а уравнение энергии — для внешней области трубы. Уравнения решались линейным методом последовательной верхней релаксации. Справедливость численных результатов была подтверждена сравнением с точным решением для более простого случая постоянного коэффициента теплообмена по внешнему периметру трубы. Рассчитанное с помощью данного метода температурное поле трубы характеризуется пятью безразмерными параметрами. Определяющим из них является параметр λ, связанный с числом Био. С его увеличением максимальный температурный градиент во внешней области трубы резко возрастает. Рассмотрены случаи применения предложенного метода и даны рекомендации по снижению температурного градиента.