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LAMINAR HEAT TRANSFER IN A CIRCULAR TUBE UNDER SOLAR RADIATION IN SPACE

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Abstract—The problem of laminar heat transfer in a circular tube under radiant heat flux boundary conditions has been analyzed. Fully developed velocity profile is assumed and the tube is considered stationary. A steady radiant energy flux is being incident on one half of the tube circumference while the fluid emanates heat through the wall on all sides by radiation to a zero degree temperature environment. A solution by finite-difference procedure has been obtained. The temperature distribution and the Nusselt number variation are presented for a wide range of the governing physical parameters.

NOMENCLATURE

- a, radius of tube [ft];
- G, incident radiation flux [Btu/hft^2];
- k, thermal conductivity of fluid [Btu/hft °R];
- l, tube length [ft];
- L, l/a, dimensionless tube length;
- p, fluid pressure [lb/ft^2];
- q, heat transfer rate [Btu/hft^2];
- r, radial coordinate [ft];
- R, r/a, dimensionless radial coordinate;
- Pe, $Pr \cdot Re$, Péclét number, dimensionless;
- Pr, $\mu C_p/k$, Prandtl number, dimensionless;
- R, r/a, dimensionless radius;
- $\hat{R}$, $R\sqrt{Pe}$, dimensionless radius in transformed coordinates;
- Re, $2Ua/\nu$, Reynold's number, dimensionless;
- T, temperature at any point [°R];
- U, average fluid velocity [ft/h];
- v_x, axial fluid velocity [ft/h];
- v_r, radial fluid velocity [ft/h];
- x, axial coordinate [ft];
- X, x/a, dimensionless axial coordinate.

Greek symbols

- α, coefficient of absorptivity of tube wall, dimensionless;
- γ, $\varepsilon\sigma T_0^3\, a/k$, a dimensionless parameter;
- ε, coefficient of emissivity of tube wall, dimensionless;
- λ, $T/(k/a\sigma)^{\frac{1}{3}}$, dimensionless temperature;
- ν, kinematic viscosity of fluid, [ft^2/h];
- ξ, X/Pe, dimensionless axial distance;
- ρ, density of fluid [lb$_m$/f^3];
- σ, Stefan–Boltzmann constant [0.1714×10^{-8} Btu/hft^2°R^4];
- ψ, $\alpha(G^3a^4\sigma/k^4)^{\frac{1}{4}}$, radiation–conduction parameter, dimensionless;
- θ, angular coordinate.

Subscripts

- 0, at entrance ($x = 0$);
- b, fluid bulk;
- cr, critical;
- r, radial;
- w, at wall ($r = a$);
- x, axial.

INTRODUCTION

HEAT transfer problems relating to laminar flow in tubes have been the subject of investigation for many years. Various investigators have dealt

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with various types of boundary conditions. Solutions involving prescribed (although variable) temperature boundary conditions include the classical work of Sellars *et al.* [1]. Other solutions in this field are well reviewed by Singh [2] who also included the effects of axial heat conduction, viscous dissipation and constant heat generation. Kuga [3] considered a sinusoidal wall temperature distribution. Solutions involving prescribed (although variable) heat-flux boundary conditions include the work of Siegel *et al.* [4]. Hsu [5] considered a sinusoidal wall heat flux distribution and Kuga [6] solved the problem for sinusoidal and exponential wall heat fluxes.

obtained an approximate solution in terms of the Liouville–Neumann series and also obtained an iterative numerical solution. Dussan and Irvine [10] also presented an approximate solution for the same problem and verified the results experimentally.

However, neither of the investigators [9, 10] considered the effect of incident radiation flux on the heat-transfer rate of the fluid. This particular problem has applications in nuclear reactors and in spacecraft. In spacecraft applications the problem may arise either in heat rejection systems or in coupling of two satellites in space. In the present investigation a finite-difference procedure has been employed to solve

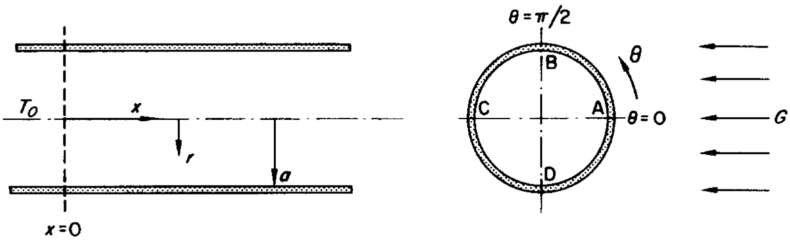

FIG. 1. Tube nomenclature and geometry.

There is another class of problems in which neither the wall temperature nor the wall heat flux is prescribed. Instead, the wall heat flux is specified as a function of the wall temperature. This type of problem has only recently received some attention. This is a more difficult problem since the heat transfer equation now involves the unknown variable (either temperature or heat flux) in an implicit rather than explicit form. References 7–10 have treated such type of problems. Sideman *et al.* [7] extended Graetz solution to include surface resistance to heat transfer in laminar flow in circular tubes and flat conduits. Stein [8] solved the Graetz problem pertaining to the concurrent flow double pipe heat exchangers and introduced an effectiveness coefficient for heat exchangers. Chen [9] solved the problem of radiant cooling of a fluid in laminar flow through a tube. He

the heat-transfer problem for fully developed laminar flow of fluid in a tube being heated by a uniform incident flux and also undergoing radiation cooling from the surface.

FORMULATION OF THE PROBLEM

Consider a constant property fluid in laminar flow through a circular tube of radius a (Fig. 1). A steady radiant energy flux of G Btu/hft^2 of projected area is being incident on one half of the tube circumference while the fluid emanates heat though the wall on all sides by radiation to a 0°R environment. At $x = 0$, the fluid is considered to have a fully developed velocity profile and a uniform temperature T_0. Heat transfer at the wall starts at $x = 0$. It is assumed that the tube is not rotating about any axis so that secondary flow effects arising from centrifugal force may be neglected.

For the physical situation stated above the continuity equation is identically satisfied and the solution of the momentum equation is of the well known form given below.

$$v_x = 2U\left[1 - \left(\frac{r}{a}\right)^2\right] \tag{1}$$

where U is the average fluid velocity.

Energy equation

Using the solution of the velocity profile from equation (1) the energy equation for the system in dimensionless form is,

$$(1 - R^2)\frac{\partial y}{\partial x} = \frac{1}{Pe}\left(\frac{\partial^2\lambda}{\partial R^2} + \frac{1}{R}\cdot\frac{\partial\lambda}{\partial R} + \frac{1}{R^2}\cdot\frac{\partial^2\lambda}{\partial\theta^2}\right). \tag{2}$$

In the above energy equation the axial heat conduction in the fluid is neglected, since it is known that the effect of this term in the energy equation is negligible for $Pe > 100$. The boundary conditions for the system in dimensionless form are as follows:

B.C. (i) At $x = 0$, $\quad \lambda = \lambda_0$. (3)

B.C. (ii) At the point $R = 0$ and $\theta = \pi/2$,

$$\frac{\partial\lambda}{\partial R} = 0. \tag{4}$$

B.C. (iii) At the wall we have the following two boundary conditions for the two regions of the circumference.

$$\left.\frac{\partial\lambda}{\partial R}\right|_{R=1} = \psi\cos\theta - \varepsilon(\lambda^4)_{R=1}, \qquad 0 \leqslant \theta\frac{\pi}{2} \tag{5}$$

and

$$\left.\frac{\partial\lambda}{\partial R}\right|_{R=1} = -\varepsilon(\lambda^4)_{R=1}, \qquad \frac{\pi}{2} \leqslant \theta \leqslant \pi, \tag{6}$$

where $\psi = \alpha(G^3a^4\sigma/k^4)^{\frac{1}{4}}$ is a dimensionless radiation–conduction parameter. Convection losses from the surface have been ignored, although their inclusion will introduce no additional mathematical difficulty.

The above boundary conditions assume that the tube wall is very thin and of low thermal conductivity so that there is no temperature drop through the tube wall and that the axial and circumferential heat conduction in the tube wall are negligible compared to the heat transfer normal to the tube wall.

Since there is symmetry about the lines $\theta = 0$ and $\theta = \pi$, only the upper part of the circle, i.e. arc ABC (Fig. 1) is considered for boundary condition (iii). This symmetry also leads to the following two boundary conditions.

$$\text{B.C. (iv) At } \theta = 0, \quad \frac{\partial\lambda}{\partial\theta} = 0, \quad 0 \leqslant R \leqslant 1. \tag{7}$$

$$\text{B.C. (v) At } \theta = \pi, \quad \frac{\partial\lambda}{\partial\theta} = 0, \quad 0 \leqslant R \leqslant 1. \tag{8}$$

METHOD OF SOLUTION

An exact analytical solution of the system of equations (2)–(8) appears to be impossible. No exact series solution for the energy equation (2) is known either. A finite-difference solution was therefore attempted.

Transformation of coordinates

If the energy equation (2) were to be solved in its present form, for high values of Péclét numbers there is a danger of losing all high-order derivatives and thus getting inaccurate results. To avoid this we introduce the following transformation:

$$\hat{R} = R\cdot\sqrt{Pe}. \tag{9}$$

Then the energy equation (2) takes the following form

$$\left(1 - \frac{\hat{R}^2}{Pe}\right)\frac{\partial\lambda}{\partial x} = \frac{\partial^2\lambda}{\partial\hat{R}^2} + \frac{1}{\hat{R}}\cdot\frac{\partial\lambda}{\partial\hat{R}} + \frac{1}{\hat{R}^2}\cdot\frac{\partial^2\lambda}{\partial\theta^2}. \tag{10}$$

Boundary condition equations (5) and (6) change to

$$\left.\frac{\partial\lambda}{\partial\hat{R}}\right|_{\hat{R}=\sqrt{Pe}} = \frac{1}{\sqrt{Pe}}[\psi\cos\theta - \varepsilon(\lambda^4)_{\hat{R}=\sqrt{Pe}}], \qquad 0 \leqslant \theta \leqslant \pi/2 \tag{11}$$

$$\left.\frac{\partial \lambda}{\partial \hat{R}}\right|_{\hat{R}=\sqrt{Pe}} = -\frac{\varepsilon}{\sqrt{Pe}}\cdot(\lambda^4)_{\hat{R}=\sqrt{Pe}}, \qquad \frac{\pi}{2} \leqslant \theta \leqslant \pi. \tag{12}$$

The finite-difference procedure

The partial differential equation (10) is parabolic in X and R and also in X and θ. Using the forward difference approximation for the derivative $\partial\lambda/\partial x$ we obtain an explicit method of solution. The central difference approximation is used for the remaining derivatives appearing in the equation (10). The finite-difference approximation to the energy equation (10) is thus as follows:

$$\left(1-\frac{\hat{R}^2}{Pe}\right)\left[\frac{\lambda_{i,j,k+1}-\lambda_{i,j,k}}{c}\right] = \frac{1}{Pe}\left\{\frac{1}{a_1^2}(\lambda_{i+1,j,k}-2\lambda_{i,j,k}+\lambda_{i-1,j,k}) + \frac{1}{\hat{R}}\cdot\frac{1}{2a_1}(\lambda_{i+1,j,k}-\lambda_{i-1,j,k}) + \frac{1}{\hat{R}^2}\cdot\frac{1}{b^2}(\lambda_{i,j+1,k}-2\lambda_{i,j,k}+\lambda_{i,j-1,k})\right\}$$

where i, j, k and a_1, b, c are the subscripts and step sizes for the $\hat{R}$, θ and x directions respectively.

On the boundaries, however, we do not try to satisfy the differential equation but satisfy the boundary conditions only.

Using the forward-difference approximation for the derivative of a function at a point, one obtains, in general, for a function ϕ at any boundary

$$\phi_n = \frac{2}{3}\left(2\phi_{n-1} - \tfrac{1}{2}\phi_{n-2} + h\frac{\partial \phi_n}{\partial n}\right). \tag{13}$$

For example, for the boundary $\hat{R} = \sqrt{Pe}$, over the range $0 < \theta < \pi/2$ the finite-difference relation used to determine function values at the boundary is

$$\lambda_{NA+1,j,k} = \tfrac{2}{3}\{2\lambda_{NA,j,k} - \tfrac{1}{2}\lambda_{NA-1,j,k} + a_1\psi\cos\theta - a_1\varepsilon(\lambda_{NA+1,j,k})^4\} \tag{14}$$

where NA is the number of intervals chosen in the $\hat{R}$-direction. The above relation is nonlinear and is solved for $\lambda_{NA+1,j,k}$ by the Newton–Raphson method.

A coarse grid was first used and then the grid size made finer until the results remained appreciably the same with any further increase in grid fineness.

Determination of Nusselt number

Once the temperature solution is obtained the Nusselt number can be evaluated as follows.

A semi-local Nusselt number Nu_x is defined as the Nusselt number at a certain distance x, averaged over the circumference

$$Nu_x = 2ha/k = \frac{2a}{k}\cdot\frac{\bar{q}_w}{(\bar{T}_w - T_b)}, \tag{15}$$

where bar over a quantity denotes its average value over a cross-section at a certain value of x. T_b is the fluid bulk temperature for any cross-section. Thus, in dimensionless form,

$$Nu_x = \frac{2\int_0^\pi \partial\lambda/\partial R|_{R=1}\,d\theta}{\int_0^\pi \lambda|_{R=1}\,d\theta - 4\int_0^\pi\int_0^1 \lambda[1-R^2]\,R\,dR\,d\theta}$$

Evaluating $\partial\lambda/\partial R|_{R=1}$ by using the boundary conditions (5) and (6) and changing to transformed coordinates the above relation takes the following form:

$$Nu_x = \frac{2\{\psi - \varepsilon\int_0^\pi \lambda^4|_{\hat{R}=\sqrt{Pe}}\,d\theta\}}{\int_0^\pi \lambda|_{\hat{R}=\sqrt{Pe}}\,d\theta - 4/Pe\int_0^\pi\int_0^{\sqrt{Pe}} \lambda[1-\hat{R}^2/Pe]\,\hat{R}\,d\hat{R}\,d\theta}. \tag{16}$$

Overall energy balance

Although consistency of results with finer grid size is normally used as a criterion of convergence and accuracy in finite difference solutions, a heat balance check was also made in this study.

Equating the difference in enthalpies at two sections over a certain length of the tube to the net heat transfer from the tube wall, the heat balance equation in dimensionless transformed coordinates can be written as,

$$\int_0^{\sqrt{Pe}}\int_0^{\pi}\lambda\Big|_{X=L}\hat{R}\left(1-\frac{\hat{R}^2}{Pe}\right)\mathrm{d}\theta\,\mathrm{d}\hat{R} = \psi L + \frac{\pi\lambda_0 Pe}{4} - \varepsilon\int_0^L\int_0^{\pi}\lambda^4\Big|_{\hat{R}=\sqrt{Pe}}\mathrm{d}\theta\,\mathrm{d}x. \tag{17}$$

The percentage error in thermal energy balance can be evaluated from equation (17).

DISCUSSION OF RESULTS

The results are presented in the form of variation of average wall temperature and Nusselt number with axial distance. The angular wall temperature distribution at any section is also studied.

Average wall temperature

The average wall temperature variation with axial distance is plotted in Fig. 2 in terms of the dimensionless parameters ξ, $\lambda(=\varepsilon\sigma T_0^3 a/k)$ and $\psi[=\alpha(G^3a^4\sigma/k^4)^{\frac{1}{4}}]$. In Fig. 2 $\overline{T}_w/T_0$ has been plotted for $\gamma = 0{\cdot}5$ and different values of ψ. As expected, the average wall temperature is higher for a higher value of ψ at any ξ. For $\psi = 0$ the problem reduces to one with no incident radiation and the temperature distribution becomes independent of θ. For this case the results are compared with those obtained by Chen [9]. The results agree fairly well and the maximum deviation in results for the range of values over which the comparison is made is less than 2 per cent. It is difficult to say whether Chen's results or the present ones are more accurate since both are iterative numerical solutions. The results also show that at $\gamma = 0{\cdot}5$, for $\psi < 1{\cdot}8$ the average wall temperature decreases with ξ while for $\psi > 1{\cdot}8$ it increases with ξ. For $\psi = 1{\cdot}8$ the

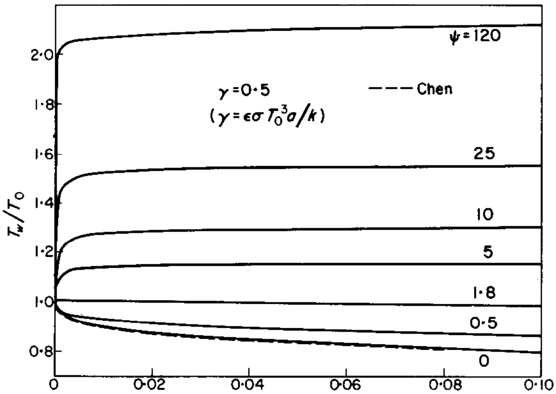

FIG. 2. Variation of average wall temperature with $\xi(\gamma = 0{\cdot}5)$.

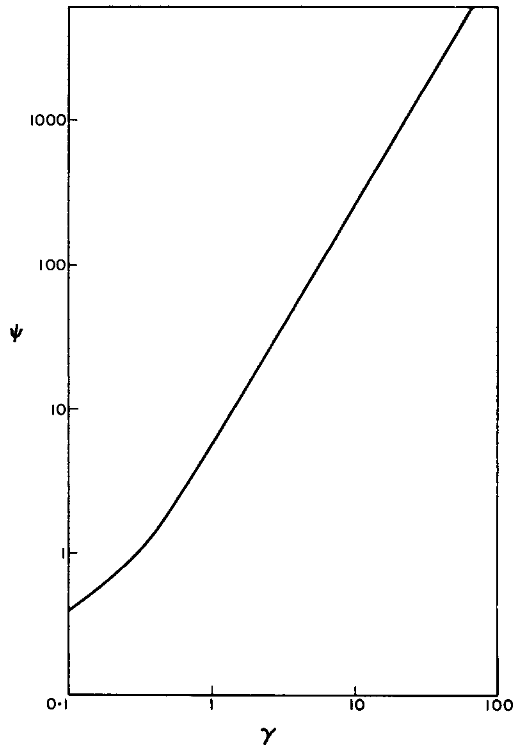

FIG. 3. Critical relationship between γ and ψ.

average wall temperature has a very small variation with ξ. This indicates that for a particular value of γ there is a critical value of ψ for which $\overline{T}_w$ does not vary appreciably from the initial value T_0. Physically this means that the amount of energy received and energy emitted are adjusted in such a way that the average wall temperature has a very small deviation.

The axial temperature analysis was also carried out for other values of γ and ψ and the critical relationship between them is plotted in Fig. 3. From this figure for a certain value of γ, if ψ is greater than the corresponding critical value, it may be concluded that the average wall temperature of the fluid at tube exit will exceed the inlet temperature. For values of ψ below the critical value the average wall temperature will continuously decrease with axial distance. It may be stressed, however, that this is only an approximate relationship since it is not possible to keep $\overline{T}_w$ constant with ξ because of the non-linearity of the problem, especially for high values of ψ. In choosing the critical values of ψ an attempt has been made to keep the net variation of $\overline{T}_w/T_0$ from (1) a minimum over the tube length. It has to be emphasized that to some extent the critical values are also dependent upon the tube length L.

Angular wall temperature distribution

The angular wall temperature variation was also studied. It was found that for increasing values of ψ and fixed values of γ and ξ, the wall temperature in the neighbourhood of $\theta = 0$ increases rapidly while the increase in the wall temperature in the neighbourhood of $\theta = 180°$ is very slow.

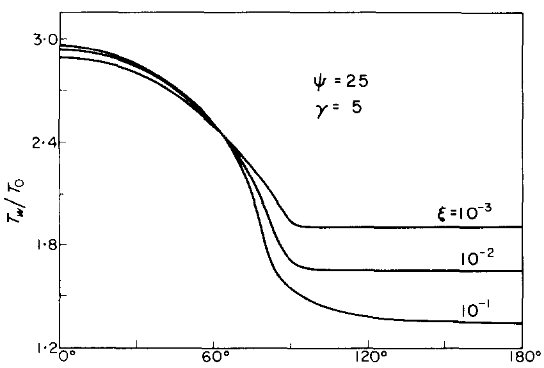

FIG. 4. Angular wall temperature distribution ($\psi = 25, \gamma = 5$).

Figure 4 gives the plots of angular variation of wall temperature for different values of ξ and a fixed value of ψ and γ. The wall temperature near $\theta = 0$ increases with ξ but near $\theta = \pi$ it decreases with ξ. Near $\theta = \pi$ the fluid does not receive any direct incident flux (although it receives heat by convection from the fluid facing the incident flux) but it loses heat according to the fourth power law of radiation. The wall temperature, therefore, decreases with ξ. Near $\theta = 0$ the effect of incident flux causes a slow increase of the wall temperature for this case where $\psi < \psi_{cr}$.

Nusselt number

The variation of Nusselt number with ξ is presented in Figs. 5–7.

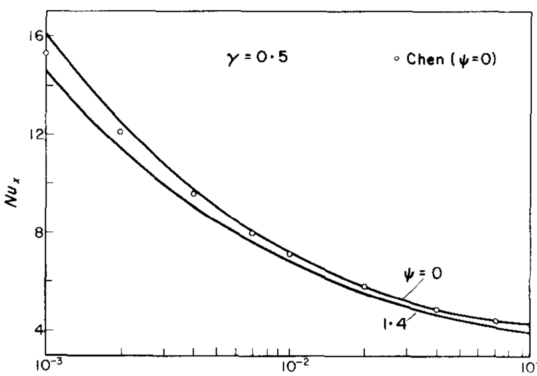

FIG. 5. Variation of Nusselt number with ξ for $\gamma = 0{\cdot}5$ ($\psi < \psi_{cr}$).

For $\psi = 0$, which corresponds to the case of no incident radiation flux, it was found that Nu decreases with increasing values of ξ and γ. A comparison was also made with Chen's [9] values and it was noted that at values of $\xi < 0{\cdot}01$ the two solutions diverge. This partial disagreement may be due to the difference in approach of the two studies, Fig. 5.

With the inclusion of incident radiation flux, however, the situation is not as simple. The Nusselt number values will depend upon whether ψ is greater or lower than the critical value and its closeness to this critical value.

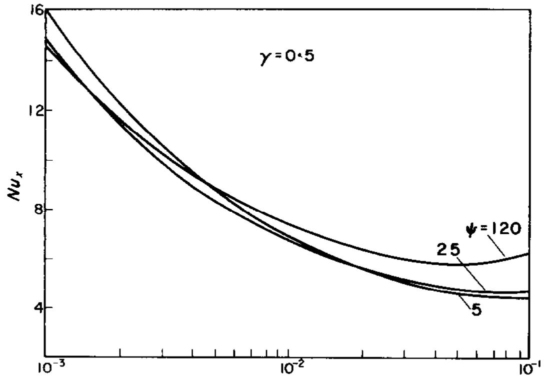

FIG. 6. Variation of Nusselt number with ξ for $\gamma = 0{\cdot}5$ ($\psi > \psi_{cr}$).

For $\gamma = 0{\cdot}5$, Fig. 5 illustrates the variation for two values of ψ. In both cases $\psi < \psi_{cr}$. Nusselt number decreases with ξ for values of ψ other than zero also, when $\psi < \psi_{cr}$. This is explained as follows: The wall temperature is greater than the inside temperature for the portion of the tube facing the incident radiation but, for the remaining portion of the tube, the wall temperature is less than the inside temperature. However, it has been noted that the integrated value of $\partial T/\partial r$ over the circumference, i.e. $\int_0^{2\pi} \partial T/\partial r|_{r=a}\,d\theta$, is negative. Also T_b is greater than T_w and, therefore, Nu remains positive. With increasing values of ξ, the net heat-transfer rate decreases in magnitude while $(T_b - T_w)$ increases so that Nu decreases.

Also, for $\psi < \psi_{cr}$, increase of ψ reduces Nu at any section because it decreases the magnitude of the net heat transfer rate from the fluid to the surroundings.

Figure 6 shows a similar plot for three other values of ψ. In this case $\psi > \psi_{cr}$ for each value of ψ. From the detailed calculations it has been noted that at any section, the wall temperature is greater than the inside temperature for the portion of the tube facing the incident radiant flux but, for the remaining portion the wall temperature is less than the inside temperature. However, the variation is in such a manner as to make $\int_0^{2\pi} \partial T/\partial r|_{r=a}\,d\theta$ positive in this case. Also $\overline{T}_w > T_b$ so that Nu remains positive. For low values of ξ, Nu is lower for a higher value of ψ. With increasing values of ξ, Nu is substantially reduced in very short distances. The decrease in Nu is less for higher values of ψ and Nu even increases towards the end for $\psi = 25$ and $\psi = 120$. (The analysis has been carried out for values of ξ up to 0·1 only.) This particular behaviour is due to the fact that although the heat transfer rate decreases with ξ, the difference $(\overline{T}_w - T_b)$ also decreases with ξ. Ultimately the Nusselt number variation depends upon the rates of decrease of these two quantities.

Figure 7 illustrates the variation of Nusselt number with ξ for a very particular situation when $\psi \simeq \psi_{cr}$ at a given value of γ. This figure illustrates that Nu decreases with ξ over most

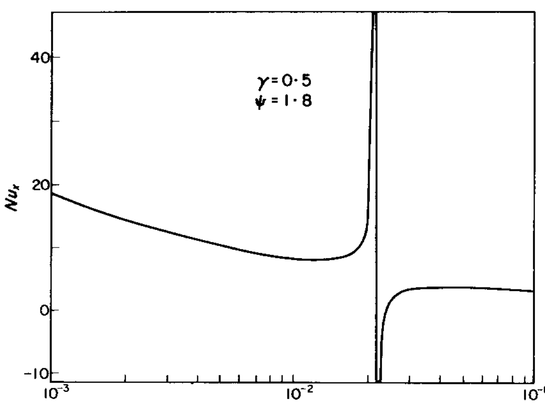

FIG. 7. Variation of Nusselt number with ξ for $\gamma = 0{\cdot}5$ ($\psi \simeq \psi_{cr}$).

of the region but fluctuates wildly in the neighbourhood of $\xi = 0{\cdot}02$. The Nusselt number even goes to negative values. To offer an explanation for this behaviour we have to analyze the variation of quantities such as net heat-transfer rate, average wall temperature and bulk temperature for this case. This is done by studying the variation of the numerator and the two terms of the denominator of equation (16). It was noted that for the indicated values of γ and ψ and for the range of ξ values covered, the heat flux decreases from $+0{\cdot}29$ to $-0{\cdot}075$. Initially, for values of ξ less than approximately 0·02, the net heat-transfer rate is positive, indicating net heat transfer into the fluid. Also, in this range of ξ, $\bar{T}_w$ is greater than T_b, so that Nu is positive. With increasing values of ξ, the difference in the values of $\bar{T}_w$ and T_b diminishes and at $\xi = 0{\cdot}021$, $\bar{T}_w = T_b$. For $\xi > 0{\cdot}021$, $\bar{T}_w$ is smaller than T_b. Therefore, in the region $\xi = 0{\cdot}020$–$0{\cdot}023$, because of the small differences in $\bar{T}_w$ and T_b, the Nusselt number fluctuates between very large positive and negative values. For $\xi > 0{\cdot}0235$ the heat transfer rate becomes negative indicating that net heat transfer now occurs from the fluid to the surroundings. Also in this range of values of ξ, $\bar{T}_w > T_b$ so that Nu is ultimately positive.

However, it may be stressed here that the wild fluctuation in Nu is only over a very small region of the tube length and if a mean Nusselt number for the whole length of the tube were to be evaluated, it would not be much affected.

Overall energy balance

The percentage error in overall energy balance as given by equation (17) was calculated and found to be less than 3 per cent for most values of γ and ψ.

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TRANSPORT DE CHALEUR LAMINAIRE DANS UN TUBE CIRCULAIRE SOUS LE RAYONNEMENT SOLAIRE DANS L'ESPACE

Résumé—Le problème du transport de chaleur laminaire dans un tube circulaire a été analysé sous des conditions aux limites de flux de chaleur par rayonnement. On a supposé que le profil de vitesse était entièrement développé et que le tube était stationnaire. Un flux d'énergie rayonnante stationnaire tombe sur une moitié de la circonférence du tube tandis que le fluide dégage de la chaleur à travers la paroi de tous les côtés par rayonnement vers un environnement à une température de zéro degré. Une solution par un processus de différences finies a été obtenue. La distribution de température et la variation du nombre de Nusselt sont présentées pour une large gamme des paramètres physiques déterminants.

LAMINARER WÄRMEÜBERGANG IN EINEM ROHR VON KREISQUERSCHNIYT BEI SONNENEINSTRAHLUNG IM WELTRAUM

Zusammenfassung—Das Problem des laminaren Wärmeübergangs in einem Rohr von Kreisquerschnitt unter dem Einfluss der Temperaturstrahlung wurde analysiert.

Voll ausgebildetes Geschwindigkeitsprofil wurde angenommen und das Rohr war als ruhend betrachtet. Ein ständiger Strahlungsstrom trifft auf eine Hälfte des Rohrumfangs, während die Flüssigkeit Wärme durch die Wände nach allen Richtungen durch Strahlung an eine Umgebung von der Temperatur Null Grad abgibt. Eine Lösung durch endliche Differenzen wurde erhalten. Die Temperaturverteilung und die Nusselt-Zahl-Änderung sind für einen grossen Bereich der beeinflussenden physikalischen Parametern angegeben.

ЛАМИНАРНЫИ ПЕРЕНОС ТЕПЛА В КРУГЛОЙ ТРУБЕ ПОД ДЕЙСТВИЕМ СОЛНЕЧНОЙ РАДИАЦИИ В КОСМИЧЕСКОМ ПРОСТРАНСТВЕ

Аннотация—Анализировалась задача ламинарного переноса тепла в круглой трубе при граничных условиях лучистого теплового потока. Предполагалось, что профиль скорости является полностью развитым, а труба стационарной. Стационарный лучистый поток энергии падал на одну половину окружности трубы, в то время как жидкость излучала тепло через стенку трубы во все стороны в пространство с нулевой температурой. Решение получено методом конечных разностей. Распределение температуры и изменение критерия Нуссельта представлены для широкого диапазона основных физических параметров.