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ON TRANSIENT NATURAL CONVECTION HEAT TRANSFER IN THE ANNULUS BETWEEN CONCENTRIC, HORIZONTAL CYLINDERS WITH ISOTHERMAL SURFACES

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Abstract—The transient natural convection heat transfer problem between two horizontal isothermal cylinders is formed within the Boussinesq approximation and solved numerically through the vorticity-stream function approach. Specifically, we discretize both the vorticity and energy equations via the alternating direction implicit (ADI) method and the stream function equation by the successive over relaxation (SOR) method. Within our range of interest, the transient time from the transient state to the steady state is found to be very small in comparison with a typical operation time. Under such circumstances, the steady-state Nusselt number can be used in lieu of the transient Nusselt number. Our numerical results are summarized by three Nusselt number vs Grashof number curves with the diameter ratio as a parameter, which serve as a guide to natural convective heat transfer calculations for an annulus.

NOMENCLATURE

a	ratio of inner radius to gap, r'_i/L'
b	ratio of outer radius to gap, r'_o/L'
c'_p	specific heat at constant pressure
L'	annulus gap
g'	gravitational acceleration
Gr	Grashof number, $L'^3 g' \beta' (T'_h - T'_c)/v'^2$
h_r	mesh interval in r-direction
h_θ	mesh interval in θ-direction
k'	thermal conductivity
Nu	Nusselt number
Pr	Prandtl number, $c'_p \mu'/k' = v'/\kappa'$
Ra	Rayleigh number, $Ra = Gr\,Pr$
R	diameter ratio, radius ratio
q'	heat flux
r	dimensionless radial coordinate, r'/L'
t	dimensionless time, $t'v'/L'^2$
T	dimensionless temperature, $(T' - T'_0)/(T'_h - T'_c)$
u_r	dimensionless radial velocity, $u'_r L'/v'$
u_θ	dimensionless tangential velocity, $u'_\theta L'/v'$.

Greek symbols

β'	thermal expansion coefficient of fluid
ζ	dimensionless vorticity function, $(L'^2/v')\zeta'$
θ	polar coordinate
κ'	thermal diffusivity, $k'/\rho' c'_p$
μ'	viscosity
v'	kinematic viscosity
ρ'	fluid density
ψ	dimensionless stream function, ψ'/v'.

Subscripts

h, c, 0	hot, cold and ambient, respectively
i, o	inner and outer, respectively
cond	conduction
conv	convection
eq	equivalent.

Superscripts

—	mean.

INTRODUCTION

THE PROBLEM of natural convective heat transfer in an annulus bounded by two horizontal cylinders has been a subject of intensive research in recent years due to its wide technological applications, which range from nuclear reactors, thermal storage systems, cooling of electronic components, aircraft fuselage insulation, to underground electrical transmission lines. A comprehensive literature survey on the subject matter can be found elsewhere [8]. In this paper, we present a brief review of selected papers which are more or less limited to theoretical–numerical studies.

Crawford and Lemlich [3] achieve the first numerical solution for a Prandtl number of 0.7 and for diameter ratios of 2, 8 and 57 via a Gauss–Seidel iterative approach. Abbott [4] obtains solutions for diameter ratios close to unity by means of matrix inversion techniques. For gases at moderate Rayleigh numbers where heat conduction is dominating, Mack and Bishop [5] employed a power series expansion valid in the range of diameter ratios from 1.15 to 4.15. However, as pointed out by Hodnett [6], if the diameter ratio becomes too large, there is a region in the annulus where convection effects are as important as conduction effects. Such a problem has been attacked by Hodnett [6] using a perturbation method. Powe *et al.* [7] examined the transition from steady to unsteady flow for air with a Prandtl number around 0.7 by determining the critical Rayleigh number at which an eddy forms and turns in the opposite direction of the main cells. Their theoretical–numerical findings are substantiated by their previous experimental ones. Kuehn and Goldstein [8] performed both experimental and theoretical–numerical studies for air and water at Rayleigh numbers from 2.1×10^4 to 9.8×10^5 at a diameter ratio of 2.6. Charrier-Mojtabi *et al.* [9] presented numerical solutions at a Prandtl number of 0.7 and 0.02 with various diameter ratios and Rayleigh numbers. All references cited except ref. [9] are confined to the steady-state analyses. Even ref. [9] gives the steady-state results only. The purpose of this paper is to present the transient-state results which are new to the authors' knowledge.

FORMULATION OF THE PROBLEM

As shown in Fig. 1, the physical system consists of a Newtonian fluid, air, in an annulus bounded by two isothermal surfaces. To formulate the problem it is assumed that: (a) the fluid motion and temperature distribution are two dimensional (2-D), (b) the fluid is viscous and incompressible, (c) frictional heating is negligible, (d) the difference in temperature between the two isothermal boundaries is small compared with $1/\beta'$, and (e) fluid properties are constant except for the density variation with temperature. Thus, within the Boussinesq approximation, four governing equations (two momentum, one energy and one continuity) are as follows [1, 2]:

$$\frac{\partial u'}{\partial t'}+u'\frac{\partial u'}{\partial x'}+v'\frac{\partial u'}{\partial y'} = -\frac{1}{\rho'}\frac{\partial p'}{\partial x'}+v'\nabla'^2 u', \quad (1)$$

$$\frac{\partial v'}{\partial t'}+u'\frac{\partial v'}{\partial x'}+v'\frac{\partial v'}{\partial y'} = -\frac{1}{\rho'}\frac{\partial p'}{\partial y'}+g'\beta'(T'-T'_0)+v'\nabla'^2 v', \quad (2)$$

$$\frac{\partial T'}{\partial t'}+u'\frac{\partial T'}{\partial x'}+v'\frac{\partial T'}{\partial y'} = \kappa'\nabla'^2 T', \quad (3)$$

$$\frac{\partial u'}{\partial x'}+\frac{\partial v'}{\partial y'} = 0, \quad (4)$$

where all primed constants, variables and operators are dimensional.

Although it is possible to obtain numerical solutions from these primitive equations, we follow the current practice of numerical analysts to utilize the vorticity-stream function approach [10]. First, we eliminate the pressure from equations (1) and (2) by cross-differentiating equations (1) with respect to y' and equation (2) with respect to x', and subsequently obtain the vorticity equation. Secondly, we change the Cartesian coordinate system to the polar coordinate system. Thirdly, we render all variables dimensionless through the following transformations:

$$r=\frac{r'}{L'},\quad u_r=\frac{u'_r L'}{v'},\quad u_\theta=\frac{u'_\theta L'}{v'},$$
$$t=\frac{t'v'}{L'^2},\quad T=\frac{T'-T'_0}{T'_h-T'_c},$$
$$\psi=\frac{\psi'}{v'},\quad \zeta=\frac{L'^2}{v'}\zeta',$$
$$\nabla^2=L'^2\nabla'^2. \quad (5)$$

Finally, we introduce the stream function and obtain:

$$\frac{\partial \zeta}{\partial t}+u_r\frac{\partial \zeta}{\partial r}+u_\theta\frac{\partial \zeta}{r\partial \theta} = Gr\left(\cos\theta\frac{\partial T}{\partial r}-\sin\theta\frac{\partial T}{r\partial\theta}\right)+\nabla^2\zeta, \quad (6)$$

$$\frac{\partial T}{\partial t}+u_r\frac{\partial T}{\partial r}+u_\theta\frac{\partial T}{r\partial\theta}=\frac{1}{Pr}\nabla^2 T, \quad (7)$$

$$u_r=\frac{1}{r}\frac{\partial\psi}{\partial\theta},\quad u_\theta=-\frac{\partial\psi}{\partial r}, \quad (8)$$

$$\zeta=-\nabla^2\psi, \quad (9)$$

where all unprimed constants, variables and operators are dimensionless. It is to be noted that both the vorticity equation (6) and energy equation (7) are of the parabolic type and the stream function equation (9) is of the elliptic type. Equation (9) is coupled with equations (6) and (7) through equation (8) which relates the stream function to the velocities. Equations (6) and (7) are coupled through the buoyancy force. Furthermore, both the vorticity equation (6) and energy equation (7) are non-linear due to the convective terms. Our problem is to seek $\zeta(r,\theta,t)$, $T(r,\theta,t)$ and $\psi(r,\theta,t)$ which satisfy three partial differential equations (6), (7) and (9) as well as the following initial and boundary conditions. To begin with, the fluid in the annulus is stationary with a uniform temperature:

$$\zeta=\psi=T=0 \quad \text{everywhere at} \quad t=0. \quad (10)$$

The boundary conditions are:

$$\psi=\frac{\partial\psi}{r\partial\theta}=\frac{\partial\psi}{\partial r}=0 \quad \text{on both walls, i.e.}\quad r=a \quad\text{and}\quad r=b, \quad (11)$$

$$T=1 \quad \text{at} \quad r=a, \quad (12a)$$

$$T=0 \quad \text{at} \quad r=b. \quad (12b)$$

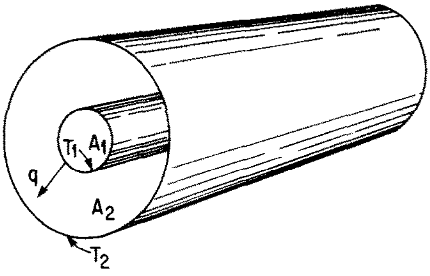

FIG. 1. Natural convection in an air filled annulus bounded by two isothermal walls.

NUMERICAL SOLUTION

The finite-difference method is employed to solve a system of equations (6)–(9) in conjunction with the initial and boundary conditions (10)–(12). Specifically, both the vorticity and energy equations (6) and (7) are discretized by the alternating direction implicit (ADI) method [10, 11], while the stream function equation (9) is discretized by the successive over relaxation (SOR) method [10, 12, 13].

Due to flow symmetry, a grid of 16 (in the r-direction) by 21 (in the θ-direction) for half the annular area is employed. The time increment is

$$\Delta t = \frac{2}{2\left(\frac{1}{h_r^2}+\frac{1}{h_\theta^2}\right)+\frac{u_r}{h_r}+\frac{u_\theta}{h_\theta}}. \quad (13)$$

The convergence criterion for the vorticity, temperature, and stream function is

$$\frac{f^m - f^{m-1}}{f^{m-1}} \leqslant \varepsilon \quad \text{for} \quad a \leqslant r \leqslant b, -\frac{\pi}{2} < \theta < \frac{\pi}{2}. \quad (14)$$

Here f is either ζ, or T, or ψ and ε is set equal to 10^{-3}. Another way of checking the convergence is to compare the mean Nusselt numbers at the inner and outer radius. These are usually within 5% at convergence. Both convergence criteria are employed in this paper.

In order to gain confidence in our numerical results, we tried to compare ours with previously published results. Figure 2, which depicts streamlines and isotherms for $Gr = 10\,000$, $Pr = 0.71$ and $R = 2$, resembles results presented by Crawford and Lemlich [3] at $Gr = 12\,500$, $Pr = 0.714$, and $R = 2$. Figure 3, which shows streamlines and isotherms for $Gr = 38\,800$, $Pr = 0.71$, and $R = 2$, is similar to one given by Charrier-Mojtabi *et al.* [9] at $Ra = 3 \times 10^4$, $Pr = 0.7$, and $R = 2$ where R and Ra denote the radius ratio and Rayleigh number, respectively. After obtaining confidence in our results, we proceeded to compute the mean transient Nusselt numbers at the inner and outer radius of seven physically realistic cases.

The equivalent conductivity, k'_{eq}, which includes both convection and conduction was introduced. From the governing differential equations (6)–(9), it was seen that the mean heat transfer or Nusselt number is a function of the Grashof number, Prandtl number and diameter ratio, i.e.

$$\frac{q'_{conv+cond}}{q'_{cond}} = \overline{Nu} = \frac{k'_{eq}}{k'} = f(Gr, Pr, R), \quad (15)$$

where local Nusselt numbers at the inner and outer radius, Nu_i and Nu_o, and mean Nusselt number, $\overline{Nu}$, are defined as follows:

$$Nu_i = -\ln R \left[r\frac{\partial T}{\partial r} \right]_{r=a}, \quad (16a)$$

$$Nu_o = -\ln R \left[r\frac{\partial T}{\partial r} \right]_{r=b}, \quad (16b)$$

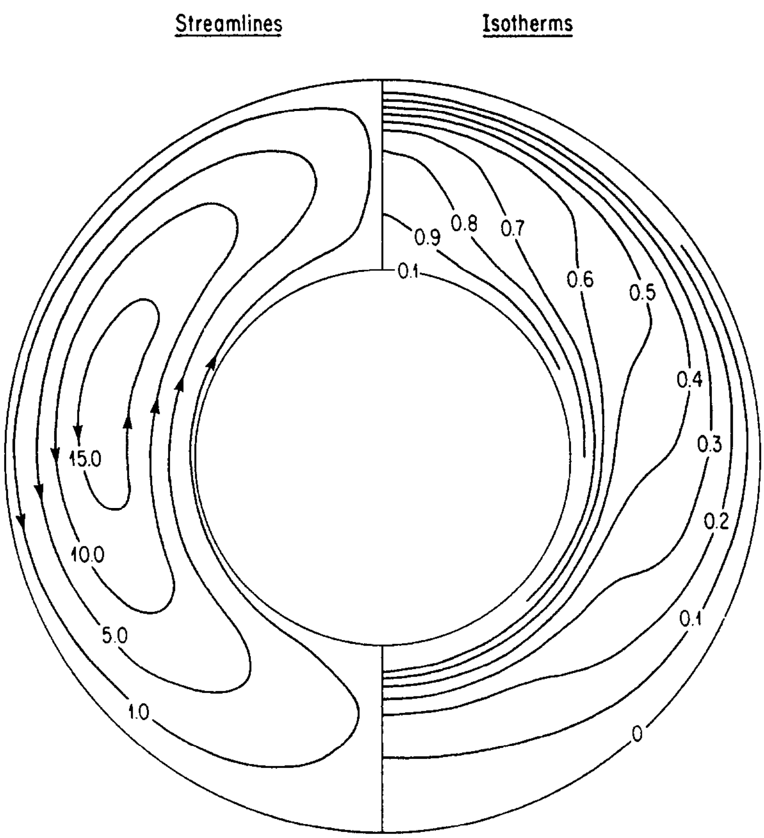

FIG. 2. Grashof number = 10 000, diameter ratio = 2.0.

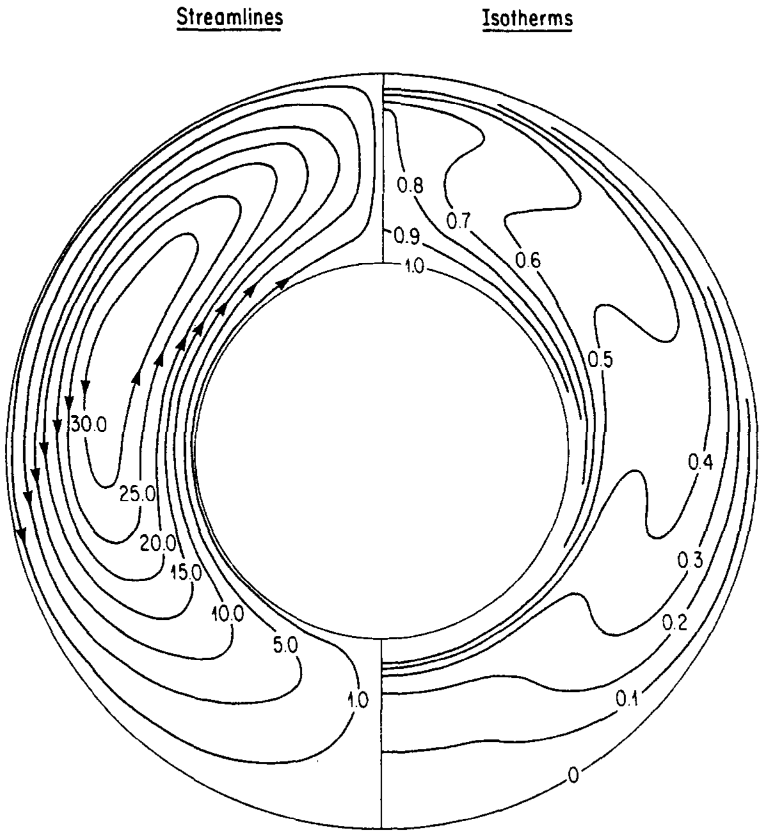

Fig. 3. Grashof number = 38 800, diameter ratio = 2.0.

$$\overline{Nu} = -\frac{\ln R}{\pi} \int_0^{\pi} \left[r \frac{\partial T}{\partial r} \right]_{r=a} \times d\theta = -\frac{\ln R}{\pi} \int_0^{\pi} \left[r \frac{\partial T}{\partial r} \right]_{r=b} d\theta, \quad (16c)$$

where it is generally assumed that the ambient air temperature, T'_0, is equal to the cold outer wall temperature, T'_c.

Both mean transient Nusselt numbers, $\overline{Nu}_i$ and $\overline{Nu}_o$, vs dimensionless time, t, are plotted in Figs. 4–10 corresponding to seven physically realistic cases [2]. As t increases, both $\overline{Nu}_i$ and $\overline{Nu}_o$ approach their steady-state values and should be equal based on a simple energy balance. In fact, due to the numerical techniques involved, the values actually obtained differ somewhat. The mean Nusselt numbers $\overline{Nu}_i$ and $\overline{Nu}_o$ are the

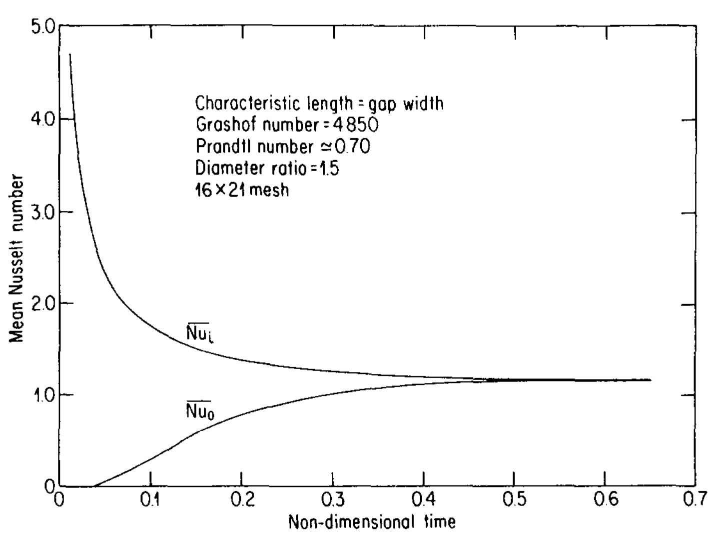

Fig. 4. Mean Nusselt number vs non-dimensional time for natural convection in an annulus.

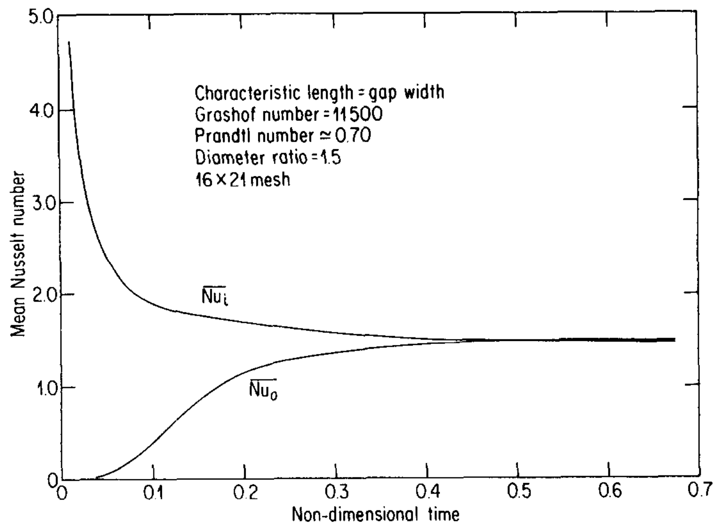

FIG. 5. Mean Nusselt number vs non-dimensional time for natural convection in an annulus.

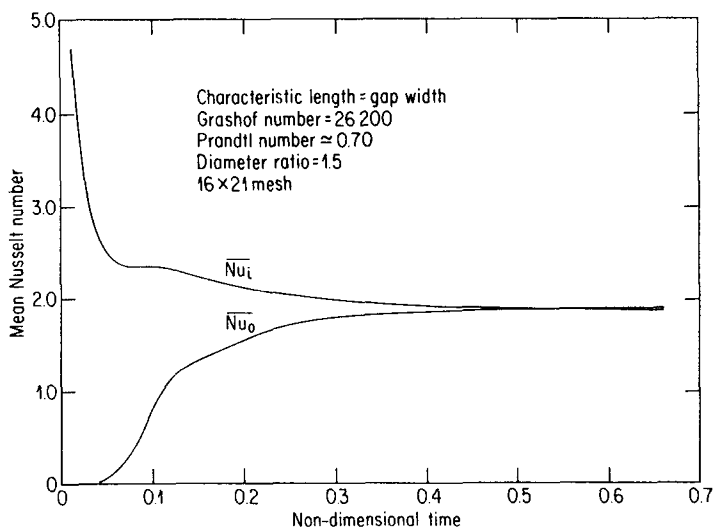

FIG. 6. Mean Nusselt number vs non-dimensional time for natural convection in an annulus.

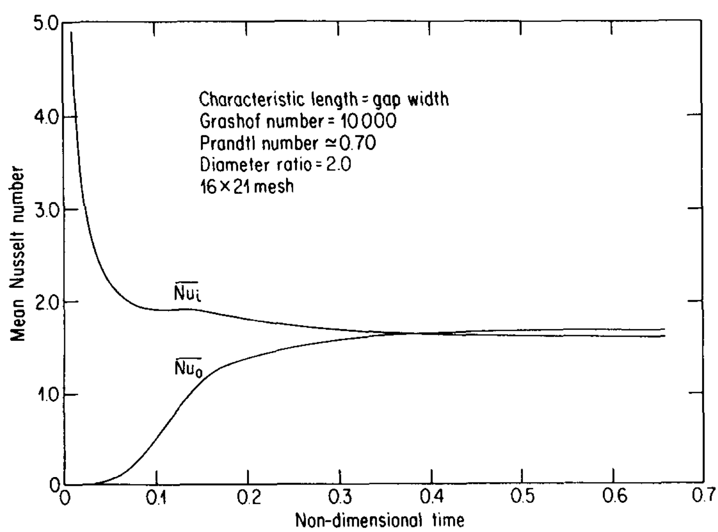

FIG. 7. Mean Nusselt number vs non-dimensional time for natural convection in an annulus.

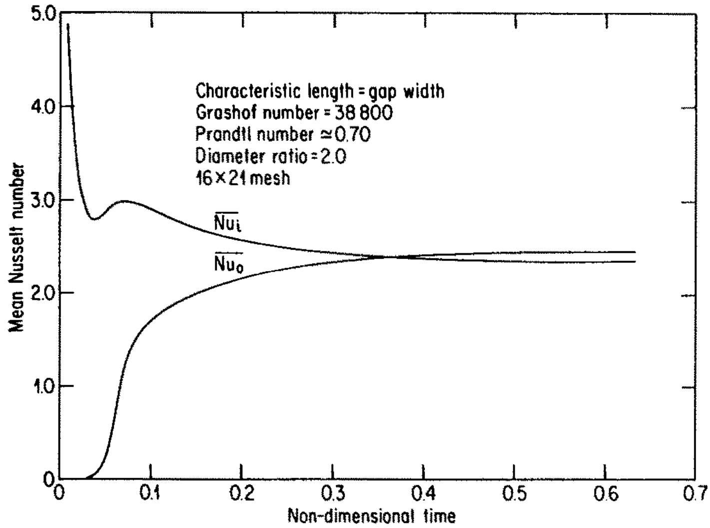

Fig. 8. Mean Nusselt number vs non-dimensional time for natural convection in an annulus.

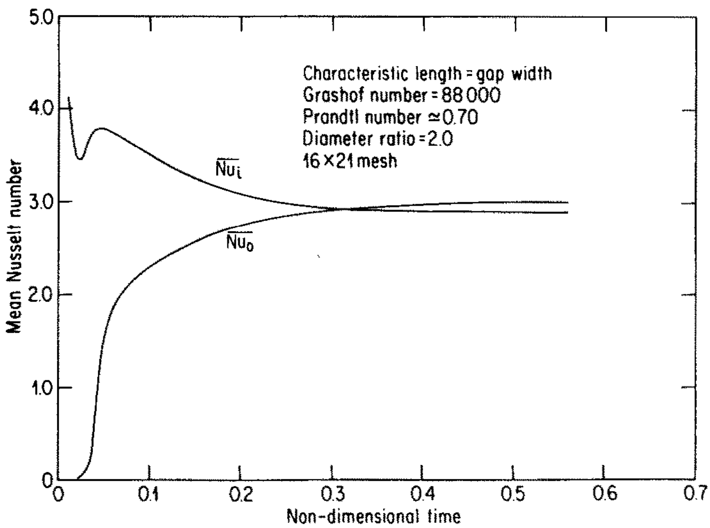

Fig. 9. Mean Nusselt number vs non-dimensional time for natural convection in an annulus.

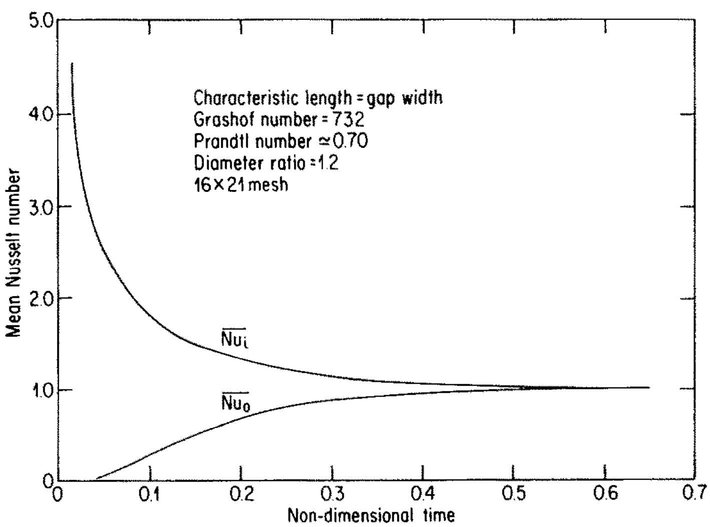

Fig. 10. Mean Nusselt number vs non-dimensional time for natural convection in an annulus.

angular average of their local ones through equation (16c).

In Fig. 10, both $\overline{Nu}_i$ and $\overline{Nu}_o$ approach unity as t increases. This means that convection is nearly nil at $R = 1.2$, and $Gr = 736$. As R increases to 1.5, the steady state $\overline{Nu} = \frac{1}{2}(\overline{Nu}_i + \overline{Nu}_o)$ approach 1.18, 1.48, and 1.88 when Gr increases from 4850, 11 500, to 26 200 as shown in Figs. 4–6, respectively. As R increases further to 2, the steady state $\overline{Nu} = \frac{1}{2}(\overline{Nu}_i + \overline{Nu}_o)$ become 1.64, 2.4, and 3.08 when Gr varies from 10 000, 38 800 to 88 000 as shown in Figs. 7–9, respectively. The variations of the steady state $\overline{Nu}$ vs Gr are plotted in Fig. 11 with the diameter ratio R as a parameter.

DISCUSSION AND CONCLUSION

In this paper, both inner and outer cylindrical surfaces are assumed to be isothermal. Mathematically, this is equivalent to assuming infinite thermal conductivity of both inner and outer cylinders. Intuitively, this isothermal approximation would be correct if the thermal conductivity of both cylinders is at least one order of magnitude higher than that of air. This is certainly the case for underground electrical transmission lines, from whose transient loading research this paper originated from. Should the isothermal approximation be not valid, this would lead to the conjugate problem which has been studied by Rotem [14]. From the computational viewpoint, only the boundary conditions have to be revised. The computer program is available from the first-named author upon request.

Our numerical calculations, as shown in Figs. 4–10, cover the Grashof number from 700 to about 90 000. At the lower end, the convection is nearly nil. At the upper end, the flow stays well within the laminar region where the governing differential equations (1)–(4) are valid. A review of Figs. 4–10 shows that the maximum non-dimensional transition time from transient state to steady state is less than unity. From the physically realizable seven cases mentioned above [2], it is shown that the maximum transition time is less than 50 s, which is very short indeed. On the other hand, in a typical overload of an underground cable, the time required for a 1°C temperature rise is about 7.5 min [2]. Therefore, at least so far as underground transmission electrical lines are concerned, the steady-state Nusselt number can be used instead of the transient-state Nusselt number for practical purposes as well as for safety sake.

As is well known, the mean Nusselt number or equivalent conductivity is a function of the Grashof number, Prandtl number and diameter ratio, i.e.

$$\overline{Nu} = \frac{k'_{eq}}{k'} = f(Gr, Pr, R) = F(Ra, R), \qquad (17)$$

where

$$Ra = Gr\,Pr.$$

In the present study, $Pr \simeq 0.71$ for air. It follows that

$$\overline{Nu} = \frac{k'_{eq}}{k'} = f(Gr, R). \qquad (18)$$

Our ranges of interest cover Grashof numbers from approximately 1×10^3 to 9×10^4 and diameter ratios from 1.2 to 2.0. Three steady-state mean Nusselt number, $\overline{Nu}$, vs Grashof number, Gr, curves are shown in Fig. 11 with diameter ratio, R, as a parameter. It is seen that, at a diameter ratio $R = 1.2$, there is no or little convective heat transfer even at $Gr \simeq 1.5 \times 10^4$, which has been substantiated by Kuehn and Goldstein's calculation (ref. [8], Table 2). Looking at the variation of $\overline{Nu}$ vs R at a fixed Gr it follows that when the diameter ratio, R, changes from 1.2 to 1.5, the convective heat transfer, i.e. $\overline{Nu}$, increases very rapidly. When the diameter ratio, R, increases further from 1.5 to 2.0, there is a substantial enhancement of the convective heat transfer. However, the rate of increase of $\overline{Nu}$ vs R slows down. After R reaches 2, the rate of increase of

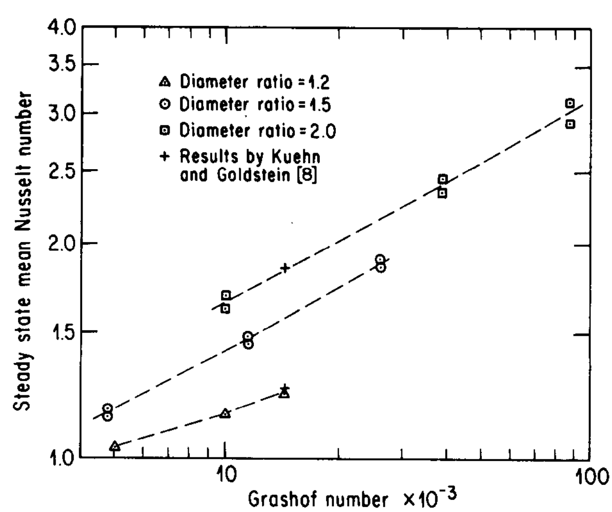

FIG. 11. Steady-state mean Nusselt number vs Grashof number for natural convection in an annulus of air.

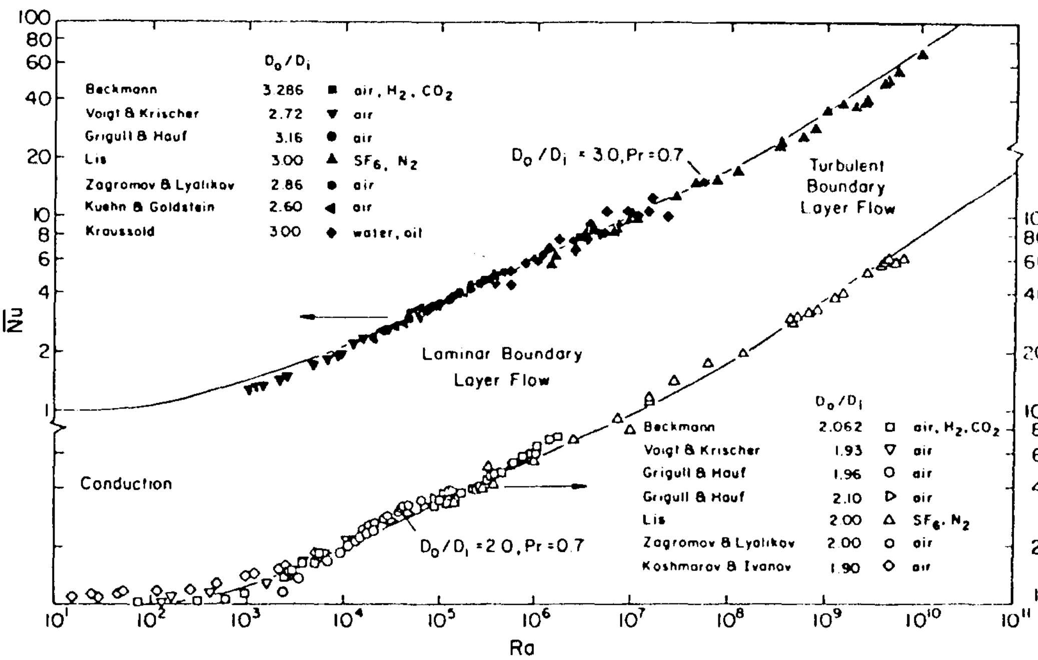

FIG. 12. Comparison of correlating equations with experimental results for natural convection between horizontal concentric cylinders [15].

convective heat transfer, $\overline{Nu}$, flattens out. This is demonstrated in Fig. 12, which depicts a collection of experimental data from various authors. From an engineering viewpoint, there is no advantage to increase the diameter ratio beyond 2 so far as natural convection is concerned. Thus, it is hoped that Fig. 11 will serve well as a guide to free convective heat transfer calculations for an annulus.

Finally, it is interesting to note that there are overshoots for small time (<0.1) in the curves for mean inner Nusselt number in Figs. 8 and 9 where the Grashof number reaches 38 800 and 88 000, respectively. Since the overshoots occur at higher Grashof numbers, our conjecture is that either the transient convection is dominating or there is numerical instability. Perhaps, the interesting overshoots will provide food for thought in future investigations.

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CONVECTION THERMIQUE NATURELLE VARIABLE DANS UN ESPACE ANNULAIRE ENTRE DES CYLINDRES CONCENTRIQUES HORIZONTAUX AVEC DES SURFACES ISOTHERMES

Résumé—Le problème de la convection thermique naturelle variable entre deux cylindres isothermes horizontaux est posé dans l'approximation de Boussinesq et résolu numériquement par l'approche de la fonction tourbillon-courant. On discrétise les équations de tourbillon et d'énergie par la méthode implicite aux directions alternées (ADI) et l'équation de fonction de courant par la méthode de surrelaxation successive (SOR). Dans notre domaine d'intérêt, le temps entre l'état variable et l'état stable est trouvé être très petit en comparaison du temps typique d'opération. Dans ces conditions, le nombre de Nusselt de régime permanent peut être utilisé à la place du nombre de Nusselt variable. Les résultats numériques sont résumés par trois courbes des nombres de Nusselt en fonction du nombre de Grashof avec comme paramètre le rapport des diamètres, pour servir de guide aux calculs de convection naturelle pour un espace annulaire.

ZUM INSTATIONÄREN WÄRMEÜBERGANG BEI FREIER KONVEKTION IM RINGSPALT ZWISCHEN KONZENTRISCHEN HORIZONTALEN ZYLINDERN MIT ISOTHERMEN OBERFLÄCHEN

Zusammenfassung—Das Problem der instationären freien Konvektion zwischen zwei horizontalen Zylindern wird im Rahmen der Boussinesq-Approximation formuliert und numerisch mit Hilfe eines Ansatzes unter Verwendung der Wirbeltransportgleichungen gelöst. Insbesondere werden sowohl die Wirbeltransport- als auch die Energiegleichungen mittels der impliziten Methode der alternierenden Richtungen (ADI-Methode) diskretisiert, ebenso die Stromfunktion mittels der Methode sukzessiver Überrelaxation (SOR-Methode). Innerhalb des interessierenden Bereiches erweist sich die Übergangszeit vom instationären zum stationären Zustand als sehr kurz im Vergleich zu typischen Betriebszeiten. Unter diesen Umständen kann die stationäre anstelle der instationären Nusselt-Zahl verwendet werden. Die numerischen Ergebnisse sind in drei Auftragungen der Nusselt-Zahl in Abhängigkeit von der Grashof-Zahl mit dem Durchmesserverhältnis als Parameter zusammengefaßt und stellen ein Hilfsmittel für Berechnungen der freien Konvektion im Ringspalt dar.

О ТЕПЛОПЕРЕНОСЕ ПРИ ПЕРЕХОДНОМ РЕЖИМЕ ЕСТЕСТВЕННОЙ КОНВЕКЦИИ В ЗАЗОРЕ МЕЖДУ КОНЦЕНТРИЧЕСКИМИ ГОРИЗОНТАЛЬНЫМИ ЦИЛИНДРАМИ С ИЗОТЕРМИЧЕСКИМИ ПОВЕРХНОСТЯМИ

Аннотация—Задача о теплопереносе при переходном режиме естественной конвекции между двумя горизонтальными изотермическими цилиндрами основана на приближении Буссинеска и решается численно в переменных вихрь – функция тока. В частности, уравнения для вихря и энергии приводятся к конечно-разностному виду по неявному методу переменных направлений, а уравнение для функции тока – по методу последовательной сверхрелаксации. В рамках поставленной задачи найдено, что время перехода к стационарному режиму очень мало по сравнению с характерным для задачи временем. При таких условиях число Нуссельта для стационарного состояния можно использовать вместо переходного числа Нуссельта. Полученные численные результаты обобщены с помощью кривых, описывающих отношение трех чисел Нуссельта к числу Грасгофа с использованием в качестве параметра отношения диаметров, что позволяет проводить расчеты теплопереноса при естественной конвекции в кольцевых зазорах.