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Laminar heat transfer in an asymmetrically heated rectangular duct

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Abstract—This paper presents a numerical study concerning the effects of non-uniform heating on the heat transfer of a thermally undeveloped gas flow in a horizontal rectangular duct; a vertical side wall is uniformly heated, and the other walls are insulated. As an initial step of the study, the duct flow is assumed to be laminar, and buoyancy effects are considered. The heat transfer rate and drag increase with the secondary flow due to buoyancy; the effects of the buoyancy force on the heat transfer and friction coefficient of the thermally undeveloped region are found to depend only upon modified Grashof numbers of the duct entrance.

INTRODUCTION

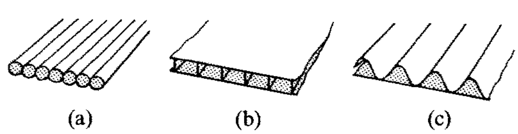
HEAT transfer in a duct flow with a circumferentially uniform heat flux has been the subject of many theoretical and experimental investigations; that is, the thermal boundary conditions usually encountered in both engineering applications and research have been considered to be uniform around the circumference of a cross-section of the duct.

On the other hand, such circumferentially uniform thermal conditions are not always achieved in practice. For instance, the first wall of the blanket in a magnetically confined nuclear fusion reactor is subjected to a large amount of thermal radiation from the plasma. The construction of the first wall proposed in a conceptual design are illustrated in Fig. 1: tubes, ribbed panels with channels having a rectangular cross-section, and embossed panels consisting of one flat sheet welded to a second sheet having embossed half-channels. Each coolant duct in these first walls is highly heated from the plasma side; circumferential nonuniformities of heating occur in these ducts.

The information and understanding of the effects of circumferential thermal nonuniformities on heat transfer are needed for the newer technologies, such as space-vehicle radiators and nuclear fusion reactor coolant passages. However, so far, there have been only a very limited number of studies done concerning heat transfer under circumferentially non-uniform thermal conditions.

Knowles and Sparrow [1] conducted a turbulent heat transfer experiment in a tube flow with a half-circumferentially heated wall. They reported that the length of the thermal entrance region on the heated side became longer than that on the unheated side. Black and Sparrow [2] made experimental studies on turbulent heat transfer in a tube with circumferentially varying thermal boundary conditions. The thermal nonuniformity in their experiment was not as strong as the one in the first wall of the fusion reactor. Sparrow *et al.* [3] obtained an experimental result concerning the heat transfer in a rectangular duct, the horizontal walls of which were heated independently. The results showed that the Nusselt numbers on a strongly heated wall were lower than the ones on a weakly heated one. Reynolds [4], Gärtner *et al.* [5] and Sparrow and Lin [6] made theoretical analyses of heat transfer in a duct flow with non-uniform heat flux by using the Fourier expansion for temperature distribution. Each of these studies employed different assumptions for turbulent eddy diffusivities. The flow fields were given, and the contributions of non-uniform heating to the flow were not discussed in those results. The effects of the secondary flow due to the buoyancy force and thermal radiation were not taken into account. In a uniformly heated tube, the effect of natural convection on heat transfer was studied by Mori and co-workers [7, 8], and the effect of thermal radiation with high heat flux was examined by Yamada and Akaike [9].

Hitherto, there have been few analytical studies about the influences of circumferential highly non-uniform heating on heat transfer in a duct flow including interaction between flow and thermal fields. The thermal and hydraulic designs of the first wall/blanket of nuclear fusion reactors require more information about heat transfer with highly non-uniform heat flux. Especially, the information about the heat transfer of a thermally undeveloped region is required for designing these coolant passages because the heat transfer coefficient of this region is inherently higher than that of the fully developed region; the usable length of the coolant passage which is subject to the high heat flux may be restricted by the heat transfer coefficient of the thermally undeveloped region.



(a) (b) (c)

FIG. 1. Typical structure of the first wall of a fusion reactor: (a) tubes; (b) ribbed panel; (c) embossed panel.

NOMENCLATURE

A_s — aspect ratio of the cross-section of a duct
c_p — specific heat
f — friction factor, $\tau/\frac{1}{2}\rho W_m^2$
g — gravity acceleration
Gr^* — modified Grashof number, $g(1/T_b)(q\cdot h/k)(h^3/\nu^2)$
Gz — Greatz number, $(Re\,Pr\,H_h)/Z$
h — height of heated wall
H_h — hydraulic diameter, $A_s h/2(1+A_s)$
k — thermal conductivity
Nu — Nusselt number, $q/(T_w-T_b)(H_h/k)$
P — pressure
p — dimensionless pressure, $(P-P_0)/\frac{1}{2}\rho W_m^2$
Pr — Prandtl number
q — heat flux intensity
Re — Reynolds number, $(W_m\cdot h)/\nu$ or $(W_m\cdot H_h)/\nu$
T — temperature
U, V, W — velocity components in X, Y, Z directions
u, v, w — dimensionless velocity $u = U/W_m$, $v = V/W_m$, $w = W/W_m$
X, Y, Z — coordinates
x, y, z — dimensionless coordinates, $x = X/h$, $y = Y/h$, $z = Z/h$.

Greek symbols
ν — kinetic viscosity
ρ — density
θ — dimensionless temperature, $(T-T_0)/((q\cdot h)/k)$
τ — wall shear stress.

Subscripts
m — mean value
w — wall
0 — inlet.

The final objectives of our study are to obtain the performance of the gas cooling first wall models as shown in Fig. 1, considering the effects of (1) the secondary flow due to buyancy force, (2) the thermal radiation transfer among mutual walls and (3) the heat conduction within duct walls. The heat transfer of the non-uniformly heated duct is considered to be affected evidently by these phenomena, and the understanding of these effects improves the accuracy and the feasibility of the thermal and hydraulic designs of the first wall/blanket of nuclear fusion reactors.

As the initial step of our study, this paper dealt with the numerical analysis of a laminar heat transfer for a gas flow in a horizontal rectangular duct; a vertical wall was uniformly heated, and the other three walls of the duct were insulated. To obtain the basic information about the heat transfer with the non-uniform heat flux, the influence of the aspect ratios of the duct cross-section and the secondary flow due to the buoyancy force upon the heat transfer in the thermally undeveloped region were examined; radiation heat transfer and heat conduction within the wall were neglected. One of the purposes of this paper is to develop the numerical procedure for heat transfer in a three-dimensional duct flow with a non-uniform heat flux, which will be available for further study.

BASIC EQUATIONS

The physical model used in this paper is illustrated in Fig. 2. The duct is placed horizontally, and the cross-section of the duct is rectangular (h in height and $h\cdot A_s$ in width). The flow is assumed to be steady, incompressible and fully developed at the entrance of the duct, and viscous dissipation is neglected. A vertical wall of this duct is uniformly heated, and the other walls are adiabatic. The Boussinesque approximation is applied to the buoyancy force. In these assumptions, the basic equations in non-dimensional form are expressed as follows:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad (1)$$

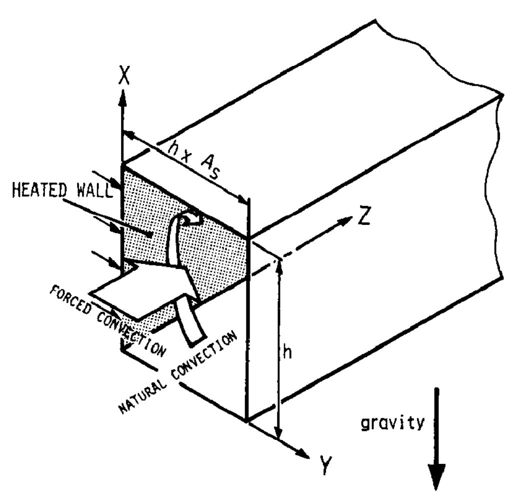


FIG. 2. Physical model and coordinates.

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{2}\frac{\partial p}{\partial x}+\frac{Gr^*}{Re^2}(\theta-\theta_b)+\frac{1}{Re}\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \quad (2)$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{1}{2}\frac{\partial p}{\partial y}+\frac{1}{Re}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \quad (3)$$

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=-\frac{1}{2}\frac{\partial p}{\partial z}+\frac{1}{Re}\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}\right) \quad (4)$$

$$u\frac{\partial \theta}{\partial x}+v\frac{\partial \theta}{\partial y}+w\frac{\partial \theta}{\partial z}=\frac{1}{Re\,Pr}\left(\frac{\partial^2 \theta}{\partial x^2}+\frac{\partial^2 \theta}{\partial y^2}\right) \quad (5)$$

where

$$Re=\frac{W_m\cdot h}{\nu},\quad Gr^*=g\frac{1}{T_b}\frac{q\cdot h}{k}\frac{h^3}{\nu^2}.$$

The boundary conditions for the above equations are given as follows:

on the walls

$$u=v=w=0,$$

$$\frac{\partial\theta}{\partial y}=-1\ (y=0),$$

$$\frac{\partial\theta}{\partial x}\ \text{or}\ \frac{\partial\theta}{\partial y}=0\ \text{(other walls)}; \quad (6)$$

at the entrance of the duct

$u=v=0,$

w: fully developed distribution,

$\theta=0,\quad p=0.$

NUMERICAL PROCEDURE

It is impossible to obtain the analytical solutions of equations (1)–(5) under boundary conditions (6); numerical procedures must be employed to obtain three-dimensional velocity and temperature profiles. The method adopted in this paper is basically identical to that developed by Patankar and Spalding [10]. The detail of the method is not described here.

RESULTS

To begin with, the feasibility of the numerical method proposed in this paper was confirmed by applying it to the conventional heat transfer problem: heat transfer in a rectangular duct flow with four uniformly heated walls in the absence of secondary flow. The Nusselt numbers, pressure drop and velocity profiles were obtained and were compared with the earlier results.

These results agree well with each other. Only the relationship between the Nusselt and Greatz numbers is shown in Fig. 3; the results of Chandrupatla and Sastri [11] are also shown in the same figure. The former is in good agreement with the latter. The numerical procedure described here is therefore capable of handling heat transfer in a duct flow with circumferentially non-uniform heat flux.

The effect of non-uniform heating on heat transfer

The local Nusselt numbers Nu_z defined by the spanwise averaged temperature of a heated wall in a rectangular flow with one vertical heated and other adiabatic walls are shown in Fig. 3 as a function of the Greatz number ($3\times10^1 \leqslant Gz \leqslant 5\times10^3$); the cross-section of the duct is square ($A_s = 1.0$), and the secondary flow is neglected. The flow at the entrance is assumed to have a fully developed velocity distribution. From this figure, the circumferential average Nusselt numbers on the heated wall Nu_z are seen to take larger values than those for all wall heating in the same Greatz number; the length of the thermal entrance region in the case of a vertical heated wall is longer than in the case of all uniformly heated walls. These results coincide with those reported by Knowles. This behavior is explained by considering that a longer time is needed for single wall heating to attain the same temperature field developed by all wall heating.

The effect of the aspect ratio of cross-section on heat transfer

The average Nusselt numbers Nu_z on a heated wall with single wall heating are presented in Fig. 4(a) for aspect ratios $A_s = 1.0$, 0.5 and 0.25, respectively. As the characteristic length, the hydraulic diameter H_h was used to define the Nusselt, Reynolds and Greatz numbers ($10^1 \leqslant Gz \leqslant 5\times10^3$) in Fig. 4(a). The Nusselt numbers exhibit higher values with a decreasing aspect ratio at the same Greatz number. As shown in Fig. 4(a), the average Nusselt number on a heated wall should be calculated for each aspect ratio as long as the Greatz number defined by a hydraulic diameter H_h is employed.

The calculation has to be executed to obtain the results for each given condition. However, if the results are rewritten by the redefined parameters as described below, all of them can be expressed by a single curve.

A redefined useful plotting of the average Nusselt

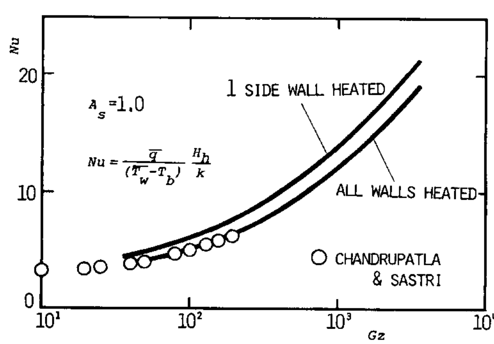
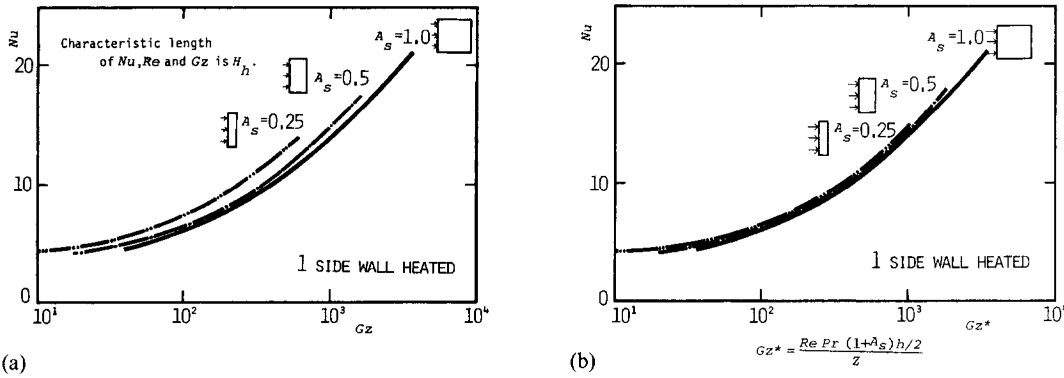


FIG. 3. Nusselt number of a non-uniform heating duct.



(a) (b)

FIG. 4. (a) Effect of aspect ratio of duct on Nusselt number. (b) Nusselt number vs modified Greatz number Gz^*.

number Nu_z vs the modified Greatz number Gz^* is proposed here; Gz^* is defined as follows:

$$Gz^* = (Re \cdot Pr \cdot (1+A_s)h/2)/Z$$

where the mean perimeter of the cross-section $(1+A_s)h/2$ is used as a characteristic length. The relationship between Nu_z and Gz^* for a few aspect ratios ($A_s = 1.0$, 0.5 and 0.25) are shown in Fig. 4(b). It is seen from this figure that the average Nusselt numbers Nu_z for various aspect ratios can be presented almost as a single curve. This result is very useful in obtaining the average Nusselt number for a heated wall with single vertical heating in a rectangular duct, whatever aspect ratio the duct itself has.

The effect of secondary flow

The heat transfer in a circumferentially non-uniform heating in a rectangular duct is affected more by the secondary flow due to the buoyancy force than the heat transfer in the case of uniform heating. The average Nusselt numbers Nu_z on a heated wall including the effect of secondary flow in the thermally undeveloped region are shown in Fig. 5 for an aspect ratio $A_s = 0.5$. The Nusselt numbers are seen to increase evidently with the decrease of the Greatz number ($Gz \leqslant 10^3$) and also with the increase of the heat flux intensity. In the fully developed region, the influence of the secondary flow was reported to depend usually on the non-dimensional parameter Gr^*/Re^2 [7, 8]; however, it was not clear in the thermally undeveloped region.

From some parameters' surveys, the modified Grashof number of the duct entrance Gr_0^*† is found to be a dominant parameter in the thermally undeveloped region. With increasing the modified Grashof number, the Nusselt numbers increase, and the onset of a secondary flow effect arises earlier in the streamwise direction of the duct. In a thermally developing region, considering the values of the parameters in this paper, the effect of secondary flow depends only on the modified Grashof number of the duct entrance as shown in Fig. 5. The Nusselt numbers for different Reynolds numbers $Re = 1395$ and 1972 with the same modified Grashof number $Gr_0^* = 0.186 \times 10^8$ are identical, whereas the Nusselt numbers for the same Reynolds number with the modified Grashof numbers $Gr_0^* = 0.373 \times 10^8$ and 0.744×10^7 exhibit different values.

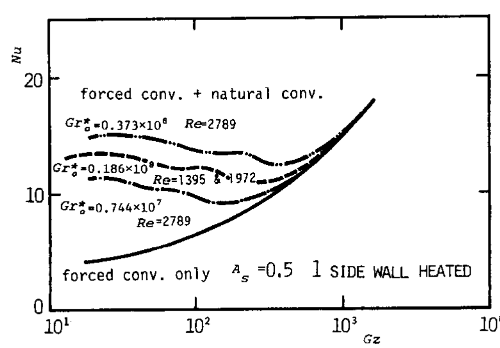


FIG. 5. Effect of secondary flow on Nusselt number.

Velocity and temperature profiles

The velocity contours and isotherms in the cross-sections both neglecting and including the effect of secondary flow due to buoyancy force are presented in Figs. 6 and 7, respectively, at three different Z positions: $Gz = 1302$, 42 and 21.

From Fig. 6, the flow patterns in the absence of secondary flow are seen to always be symmetrical with respect to the axis of the duct, as expected; the

† In this paper, the modified Grashof number Gr^* is defined as

$$Gr^* = g\beta \frac{q \cdot h}{k} \frac{h^3}{\nu^2}$$

where β is the volumetric expansion factor of fluid: $\beta = 1/T_b(Z)$ for gas. The bulk temperature of the fluid T_b increases in the Z direction, hence the value of the modified Grashof number decreases along the Z direction. In this paper, the modified Grashof number at the duct entrance Gr_0^* was adopted to indicate the calculation conditions.

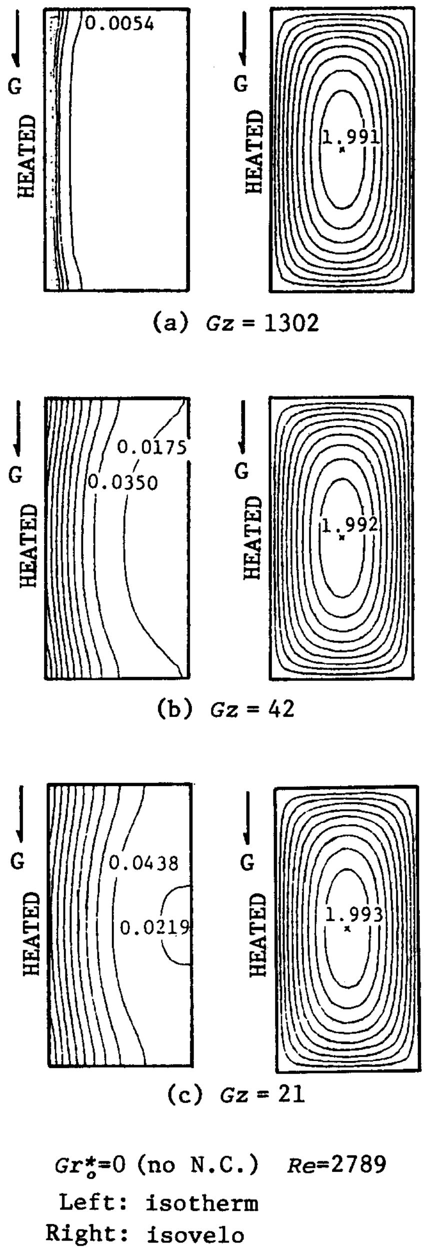


FIG. 6. Isotherms and velocity contours in the thermal developing region.

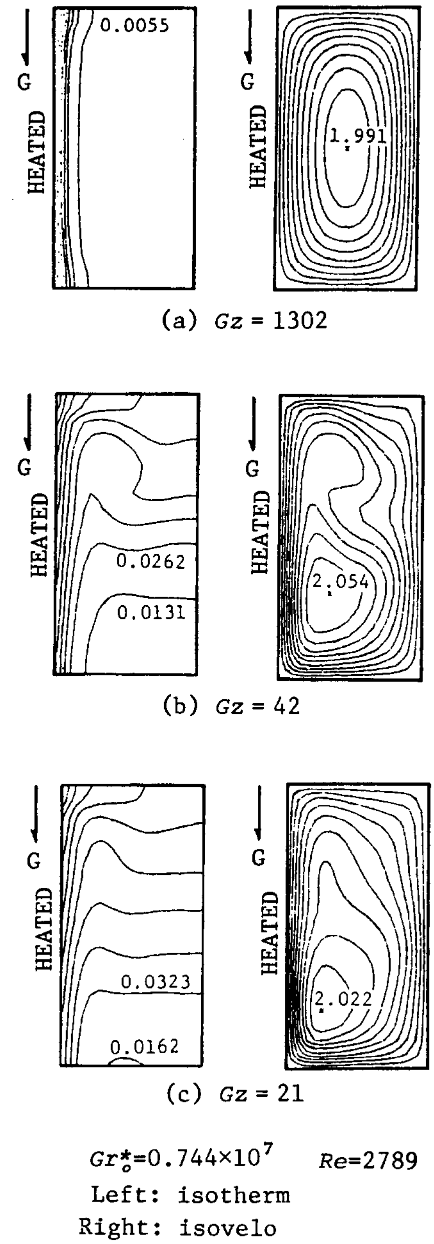


FIG. 7. Isotherms and velocity contours under buoyancy force.

temperature patterns develop symmetrically with respect to the horizontal mid-plane. On the other hand, both the velocity and temperature profiles in the presence of a buoyancy effect are distorted by the secondary flow as shown in Fig. 7. The profiles for the buoyancy-affected case clearly differ from the ones for the buoyancy-unaffected cases; especially, in the small Greatz number region. In the case including the buoyancy effect, the hot gas heated on a hot wall travels to the upper portion of the duct; the velocity and temperature profiles develop almost fully and become stable as seen in Fig. 7. Figure 8 shows the secondary flow patterns for the cases corresponding to Fig. 7; the process of the development of vortices is clearly shown.

Pressure drop

Generally speaking, the pressure drop increases when the secondary flow appears. Figure 9 shows the friction factor f of duct vs Greatz number for the cases of three different values of heat flux. The value of $f \cdot Re$ for no secondary flow is uniform throughout the duct and takes the value of 15.5, which is in good agreement with the analytical results obtained by

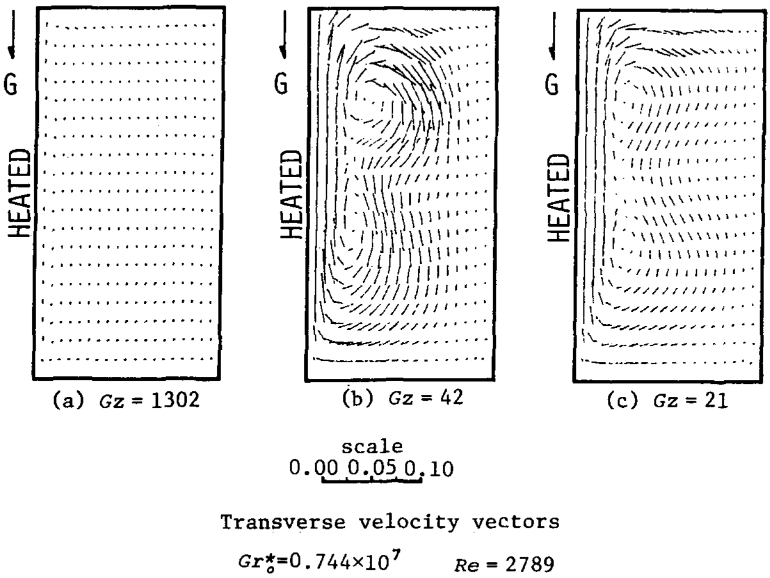


FIG. 8. Secondary flow patterns.

Shah and London [12]. The friction factor increases remarkably in the case including the secondary flow as seen in Fig. 9. The drag coefficients $f \cdot Re$ depend strongly upon the modified Grashof number, whereas they depend little on the Reynolds number similar to the aforementioned heat transfer case.

Some waviness appears in these friction factor curves in Fig. 9, where some transitions in flow pattern were observed in the calculation process; for instance, an initial vortex in the secondary flow seems to transfer to two vortices.

The circumferential distributions of the dimensionless wall temperature θ_w, the local Nusselt number Nu and the local friction factor $f \cdot Re$ are shown in Fig. 10 to clarify the relationship between the enhancement of the local heat transfer and the increase of the local friction factor due to natural convection. Those results including the effect of secondary flow due to buoyancy force at three different Z positions

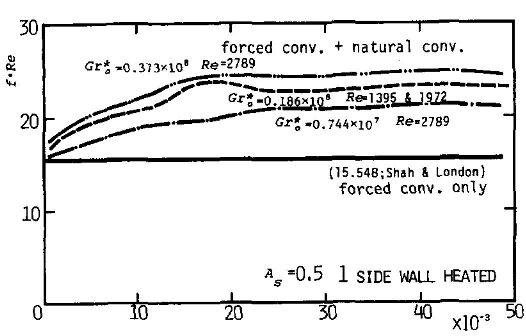


FIG. 9. Friction factor under buoyancy force.

(Gz = 1302, 42 and 21) are presented in this figure. It is noted that the heat transfer of the lower portion of the heated wall is remarkably enhanced by the secondary flow in the small Greatz number region; however, the friction factor of this portion increases concurrently. In this figure, the dimensionless wall temperatures and the local Nusselt numbers at the duct corners were not given because the grid nodes used in our calculations were not so fine as to obtain an accurate solution in these positions.

In Fig. 10(b) (Gz = 42), a dip of the friction factor appears on a mid-plane of the unheated vertical wall, where the secondary flow tends to leave the wall, and the velocity of the main flow is reduced by this secondary flow as shown in Figs. 7 and 8.

Wall temperature

The effects of circumferentially non-uniform heating on heat transfer have been discussed in the previous sections. However, from the point of view of designing coolant passages subject to high heat flux such as the first wall of a fusion reactor, it is a rather more important problem to know how high the wall temperatures of the passage rise than to know the heat transfer coefficients.

In a uniformly heated coolant passage, the critical heat flux can generally be evaluated from the thermal properties of the duct material and the heat transfer rate which can be obtained from the generalized results. However, it is difficult to satisfactorily estimate the critical heat flux of a circumferentially non-uniform heated coolant passage; so far, there is no ample information about heat transfer with circumferentially non-uniform heating. Therefore, the data on heat transfer in the case of uniform heating have usually been used instead of the ones for the case

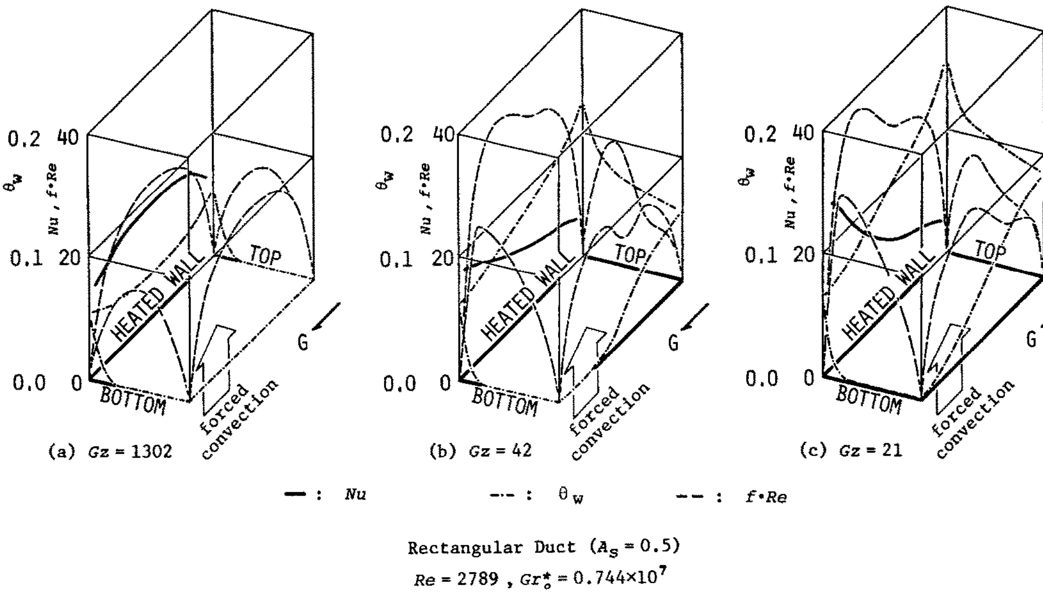


FIG. 10. Circumferential distributions of dimensionless wall temperature, local Nusselt number and local friction factor under buoyancy force.

of circumferentially non-uniform heating. Consider ducts subject to both circumferentially uniform and non-uniform heating with the same heat load, i.e. the same total heat flux; the duct with non-uniform heating is subjected to locally higher heat flux than the one with uniform heating. If a uniformly heated duct is assumed instead of a non-uniformly heated one, the maximum heat flux must be underestimated. This is a very serious problem from the point of view of safety.

The typical wall temperature variations along a duct are shown in Fig. 10 : the rectangular duct (aspect ratio A_s = 1.0) with a vertical heated and three adiabatic walls. The solid line in the figure shows the heated wall temperature for a non-uniformly heated duct; the broken line shows the one for a uniformly heated duct.

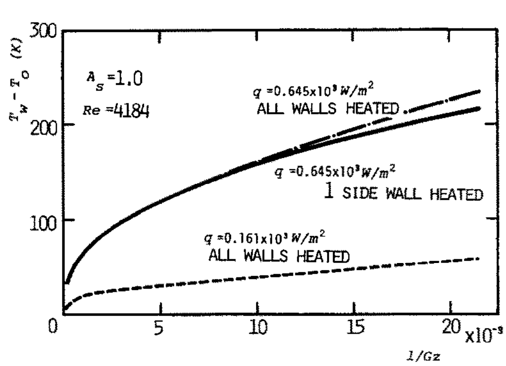


FIG. 11. Temperature distribution of heated wall.

CONCLUSIONS

As the initial step of heat transfer in a duct flow with circumferentially non-uniformly heated walls, this paper dealt with the numerical analysis of laminar convective heat transfer of a flow in a rectangular duct. A vertical wall was heated, and the other walls were adiabatic. The following conclusions have been obtained:

(1) The lengths of thermal developing regions from the entrance of a duct with a vertically heated wall are longer than the ones with all heated walls. The average Nusselt number on a heated wall for the case of a vertically heated wall is larger than the one for the case of all heated walls.

(2) The relationship of the average Nusselt number on the heated wall vs the modified Greatz number proposed in this paper was shown to be a curve for various aspect ratios of the cross-section of a duct.

(3) Heat transfer in the thermally developing region is enhanced by the secondary flow due to buoyancy force. However, the friction factor of the duct concurrently increases with the secondary flow. The extent of the effect of secondary flow on both the heat transfer and the friction factor depends only upon the modified Grashof number at the duct entrance.

(4) The critical heat flux is overestimated, as can be expected, if the heat transfer characteristics for uniformly heated walls are used to calculate it instead of non-uniformly heated ones.

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TRANSFERT THERMIQUE LAMINAIRE DANS UN CANAL RECTANGULAIRE DISYMETRIQUEMENT CHAUFFE

Résumé—On présente une étude numérique concernant les efforts de chauffage non uniforme sur le transfert de chaleur pour un écoulement gazeux thermiquement non établi dans un canal rectangulaire; une paroi latérale verticale est chauffée uniformément et les autres parois sont adiabatiques. Dans cette étude, l'écoulement est supposé laminaire et on considère les effets d'Archimède. Le transfert thermique et la trainée augmentent avec l'écoulement secondaire dû à la pesanteur; les effets de force de pesanteur sur les coefficients de convection et de frottement, pour la région thermiquement non établie, dépendent seulement des nombres de Grashof modifiés à l'entrée du canal.

WÄRMEÜBERGANG BEI LAMINARER STRÖMUNG IN EINEM ASYMMETRISCH BEHEIZTEN RECHTECKIGEN KANAL

Zusammenfassung—Die Einflüsse von ungleichförmiger Beheizung auf den Wärmeübergang in einer thermisch nicht voll ausgebildeten Gasströmung in einem horizontalen rechteckigen Kanal wurden numerisch untersucht. Eine vertikale Seitenwand wird gleichförmig beheizt, die anderen Wände sind isoliert. In einem ersten Schritt wird angenommen, daß die Kanalströmung laminar sei, Auftriebseffekte werden berücksichtigt. Wärmeübergang und Druckabfall wachsen mit der durch Auftrieb hervorgerufenen Sekundärströmung. Die Einflüsse der Auftriebskraft auf den Wärmeübergang und den Reibungskoeffizienten im Bereich der thermisch nicht voll ausgebildeten Strömung hängen nur von der modifizierten Grashof-Zahl für den Kanaleintritt ab.

ТЕПЛОПЕРЕНОС В АСИММЕТРИЧНО НАГРЕВАЕМОМ ПРЯМОУГОЛЬНОМ КАНАЛЕ ПРИ ЛАМИНАРНОМ ТЕЧЕНИИ

Аннотация—Численно исследовано влияние неоднородного нагрева на теплообмен термически неустановившегося потока газа в горизонтальном прямоугольном канале, вертикальная боковая стенка которого нагревается однородно, в то время как все остальные теплоизолированы. Предполагается, что течение в канале ламинарное; рассчитываются эффекты плавучести. За счет возникновения вторичного течения под воздействием подъемной силы интенсивность теплопереноса и сопротивление возрастают; влияние подъемной силы на теплоперенос и коэффициент трения в термически неустановившейся зоне зависит только от модифицированного числа Грасгофа на входе в канал.