Int. J. Heat Mass Transfer. Vol. 10, pp. 333–347. Pergamon Press Ltd. 1967. Printed in Great Britain

CONVECTION HEAT TRANSFER IN THE ENTRY REGION OF A TUBE WHICH REVOLVES ABOUT AN AXIS PARALLEL TO ITSELF

J. F. HUMPHREYS, W. D. MORRIS and H. BARROW

Department of Mechanical Engineering, University of Liverpool

(*Received* 25 *February* 1966 *and in revised form* 12 *September* 1966)

Abstract—The local and mean heat-transfer characteristics have been studied experimentally for air (Prandtl number = 0·7) flowing turbulently in the entrance region of a circular duct revolving about a horizontal axis parallel to the duct axis. One geometry only has been studied. Nominal Reynolds numbers of 5000 to 20000 were used and the boundary conditions were those of uniform initial temperature distribution and uniform wall heat flux. Rotational speeds of 0, 200 and 500 rev/min. were employed and the radius of rotation was 6 in.

Results showed that significant increases in heat transfer occurred due to rotation and it has been reasoned that these changes were a result of two effects, namely, the inherent inlet swirl relative to the revolving tube together with Coriolis forces, and centrifugal buoyancy forces. The former was found to be of more significance than the latter.

NOMENCLATURE

a_1, a_2, constants [dimensionless];
A, acceleration ratio ($=H\Omega^2/g$) [dimensionless];
c, specific heat at constant pressure [Btu/(lb degF)];
d, inside diameter of tube [ft];
g, acceleration due to gravity [ft/h^2];
h, heat-transfer coefficient [Btu/h ft^2 degF];
H, distance of tube axis from axis of rotation [ft];
K, constant [dimensionless];
k, thermal conductivity [Btu/h ft degF];
q, specific wall heat flux [Btu/h ft^2];
T, temperature [degF];
ΔT, temperature difference as defined in text [degF];
u, v, w, components of velocity [ft/h];
w_m, mean axial velocity [ft/h];
r, θ, z, rotating cylindrical co-ordinates [ft, radians, ft];
Ω, angular velocity [radians/h];
ρ, density [lb/ft^3];
μ, absolute viscosity [lb/(ft h)];
β, coefficient of volume expansion [per degF Abs.];
ν, kinematic viscosity ($=\mu/\rho$) [ft^2/h];
Pr, Prandtl number ($=c\mu/k$) [dimensionless];
$\overline{Re}$, mean Reynolds number ($=w_m d/\nu$) [dimensionless];
Ro, Rossby number ($=w_m/H\Omega$) [dimensionless];
Nu, local Nusselt number ($=hd/k$) [dimensionless];
Nu, mean Nusselt number [dimensionless];
$\overline{Nu}^*$, excess Nusselt number as defined in text;
$\overline{Gr}_g$, mean gravitational Grashof number ($=gd^3\beta\Delta T/\nu^2$) [dimensionless];
$\overline{Gr}_r$, mean rotational Grashof number ($=H\Omega^2 d^3\beta\Delta T/\nu^2$) [dimensionless];
$\overline{Ra}_g$, mean gravitational Rayleigh

number ($=\overline{Gr} \times Pr$) [dimensionless];

$\overline{Ra}_r$, mean rotational Rayleigh number ($=\overline{Gr}_r \times Pr$) [dimensionless];

ϕ_1, ϕ_2, ϕ_3, functions.

Subscripts

a, ambient;

b, bulk;

g, non-rotational case;

m, mean;

r, rotational case;

w, wall.

1. INTRODUCTION

DURING the past five or so decades, the most common investigations into convective heat transfer for flow in pipes and tubes have been devoted to cases where unidirectional and axisymmetric fluid and heat flows prevail. To enable the numerous theoretical predictions for such conditions of flow to be appraised and also to furnish reliable empirical correlations for the design of heat-transfer plant, careful experimentation which ensures these unidirectional and axisymmetric characteristics has been necessary.

In recent years it has become increasingly apparent, however, that large order influences on heat-transfer rates may occur because of secondary flow components in a plane perpendicular to the main flow direction. We may generalize by saying that these secondary flow components may be an inherent characteristic of the flow itself, may arise as the result of an asymmetry in the thermal boundary condition, or may occur under the influence of a body force field. In any real situation all three of the aforementioned reasons for secondary flow may occur simultaneously to some extent. However, it is often the case that one of the effects is dominant and much research activity has been directed towards obtaining a clear understanding of these individual secondary flow effects.

The importance of secondary flow effects on both heat transfer and pressure drop for duct flow may be exemplified by reference to the following practical situations, each of which has been selected to illustrate the various means by which the secondary flows occur. The flow in the inlet eye of a centrifugal fan may have a swirl component relative to the rotating inlet owing to the fact that the flow tends to remain irrotational. The action of viscous shear eventually may cause the fluid to rotate solidly with the entry, provided the entry itself is sufficiently long. Again, the nature of the flow through a pipe bend is complicated by the fact that secondary flow is superimposed onto the mean flow through the pipe, as described by Dean [1]. The resulting distortion of the usual pipe flow velocity field may survive for a considerable distance after the bend has been negotiated so that, if the bend leads into a heated straight portion, significant changes in the local heat-transfer rates may be expected, as shown by Ede [2]. This flow configuration is frequently encountered in practice. With both these examples, the secondary flow is an inherent characteristic.

Interesting cases where secondary flow is produced because of asymmetrical thermal conditions have been reported by Whittley and Barrow [3] and Papapanayiotou [4] where distortion of an otherwise unidirectional flow occurs because of a transverse gradient of internal heat generation. This phenomenon is particularly related to the field of nuclear power generation

For heated duct flows where the Earth's gravitational field is not aligned in the direction of flow, free convection produces secondary flow. A typical case which may be used for illustration is that of the uniformly heated horizontal pipe. This configuration has been studied theoretically by Morton [5] and experimentally by McComas and Eckert [6]. Once more, significant alterations to the heat-transfer rates are discernible. Body forces associated with duct rotation can induce secondary flows without the free convection motion achieved when fluid density gradients are present. For example, the flow in a tube which rotates about an axis perpendicular to its axis of symmetry spirals in a manner similar to that which occurs in curved pipes, see Barua [7]. A case where centrifugal buoyancy in a heated rotating pipe causes secondary flow has been studied by Morris [8]. Here fluid flows in a uniformly heated pipe which rotates about an axis parallel to its axis of symmetry. Improvements in the heat transfer are found to occur.

Three reasons why secondary flows in ducts may occur have been given and a number of examples where one of these effects has been dominant have been quoted. In the present paper, heat transfer in the entry region of a uniformly heated cylindrical pipe is experimentally studied for the particular case when two of the reasons which give rise to secondary flows exist simultaneously. Specifically, the pipe is constrained to rotate about an axis parallel to its axis of symmetry as shown in Fig. 1. Firstly, as with flow in the inlet eye of the centrifugal fan, the fluid experiences a swirling component relative to the tube. At entry to the test section the swirl may approximate to that of solid rotation relative to the tube in the core of the fluid. Viscous effects require the velocity at the wall to be zero, however, so that near the wall the swirl component will deviate from that of solid rotation.

It is conceivable that this swirl decays along the pipe until eventually some terminal velocity field is achieved. It is interesting to note that Morris [9] shows for isothermal laminar flow that the terminal velocity profile is the usual parabolic axial distribution with zero secondary flow. In this case rotational influences are made manifest as pressure gradients in the plane perpendicular to the flow.

Secondly, centrifugal and Coriolis forces together with fluid density gradients produce buoyant motion which enhances the secondary flow already present due to the inherent entry swirl. The influence of the Coriolis buoyancy is probably not as marked as that due to its centrifugal counterpart. This is also apparent from the asymptotic solutions presented by Morris [8] for a similar geometry. Initially it was envisaged that the influence of rotational buoyancy would be more marked than the influence of entry swirl.

From the foregoing qualitative description of flow conditions in the entry region of this particular rotating tube it is evident that a highly complex velocity field occurs which may have marked effects on both heat-transfer and pressure drop data.

Before a detailed discussion of the present work is given it is interesting to note some of the practical situations where this type of rotating geometry is encountered. One of the factors

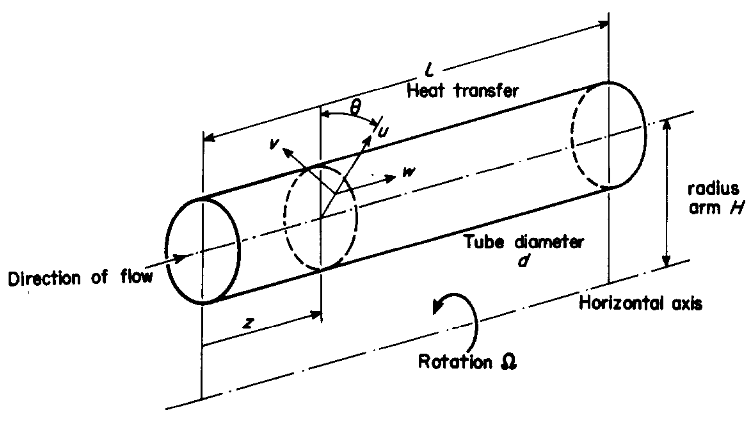

FIG. 1. Idealized model.

Y

which controls the reliable operation of large electrical machines is the maximum permissible temperature of the insulation surrounding the rotor conductors making it necessary to incorporate some cooling system into the design. This is commonly achieved by pumping a suitable coolant through hollow passages situated inside the conductors themselves. As far as the authors are aware no information concerning the influence of conductor rotation on the heat-transfer mechanism is available. Also, the actual rotor drum is sometimes cooled by air flowing through axially located holes in the drum. Davies and Morris [10] have suggested the use of closed loop rotating thermosyphons for the solution of the rotor conductor cooling problem and present results [11] for one such geometric configuration. These authors make the point, however, that lack of precise information concerning the general problem of flow inside rotating ducts makes the prediction of thermosyphon performance difficult which, incidentally, resulted in the present investigation. Again for reliable gas turbine performance at high gas inlet temperatures some method of rotor blade cooling is essential with present day materials of construction. For forced internal cooling of these blades, flow geometries similar to that being studied in this paper are often used. This type of flow geometry may also be used for the matrix of rotary regenerators suitable for the improvement of automobile gas turbine plants.

As well as its academic interest the flow and heat-transfer characteristics of the present rotating geometry have significant practical importance and this investigation is an attempt to determine the salient parameters and to obtain a qualitative assessment of their relative importance. Details of the investigation now follow.

2. FORMULATION OF PROBLEM

An investigation designed to illustrate the salient effects of tube rotation on the mechanism of heat transfer for any rotating geometry brings to light a number of fundamental difficulties not encountered in conventional tube-flow heat-transfer experiments. Traditional non-rotating tube studies are performed with tubes sufficiently long to permit the establishment of terminal velocity and temperature profiles from which so called asymptotic heat-transfer coefficients are determined. For rotating tube experiments such lengths are prohibitive making it necessary to confine the investigation to the entry region. It should be noted, however, that this is not a serious limitation for in the majority of practical applications, which involve this rotating geometry, the attainment of a so called established flow is unlikely; making the entry region of prime importance. In the light of these difficulties it was decided to conduct a basic investigation using a test section nominally 12-in long by 1-in diameter with an eccentricity of 6 in.

A further difficulty associated with this type of problem concerns the velocity profile at entry to the test section. When the thermal entry problem for non-rotating tubes is investigated the entry velocity profile is, to some extent, under the control of the experimenter. This is not possible with a rotating tube system because the extent of the inherent entry swirl may alter with changes in rotational speed. The extent of the variation of entry velocity profile with rotational speed may also be expected to depend on the flow geometry upstream of the test section. Thus data obtained from rotating duct experiments must be related to the flow geometry prior to the test section as well as to the rotational speed itself. In order to completely specify the influence of intake geometry on the data it is obvious that numerous intake geometries must be studied requiring extensive experimentation. For the test-section dimensions given above, this paper presents the results obtained for one type of intake geometry for a number of rotational speeds. Actual details of the intake geometry and the entire apparatus are given in Section 4.

3. THEORETICAL CONSIDERATIONS

A theoretical analysis of the hydrodynamic and thermal fields in the entry region of this rotating tube is extremely complex. However, by the simple expedient of dimensional analysis it can be shown that the functional relationship given by equation (1) may be expected.

In his work on planetary waves, Rossby [12] established the importance of a dimensionless group indicating the relative importance of inertia and Coriolis forces. This parameter, now called the Rossby number (Ro) is also applicable in the present problem. It was pointed out earlier that Morris [8] showed for a similar geometry

$$\overline{Nu} = \phi_1\left(\frac{L}{d}, \frac{d}{H}, \overline{Re}, \overline{Gr}_r, Ro, Pr\right) \tag{1}$$

where

$\overline{Nu}(=qd/k\Delta T)$	= mean Nusselt number
L/d	= aspect ratio of test section
d/H	= eccentricity parameter
$\overline{Re}(= w_m d/\nu)$	= mean Reynolds number
$\overline{Gr}_r(=H\Omega^2\beta d^3\Delta T/\nu^2)$	= mean rotational Grashof number
$Ro(=w_m/H\Omega)$	= Rossby number
$Pr(=c\mu/k)$	= Prandtl number.

and established laminar flow, that the influence of Coriolis forces on the heat transfer is small for the low rotational speed solutions presented. For the purposes of the present paper it is perhaps more convenient to think of the Rossby number as a parameter which characterizes the magnitude of the inherent entry swirl.

Although the Nusselt, Reynolds and Prandtl numbers are well known and require no further discussion it is necessary, however, to comment on the remaining terms in equation (1). It will be noted that, whereas in non-rotating tube experiments only one dimensionless parameter is usually required to specify the geometry, it is necessary to specify two dimensionless groups for the rotating case. These are respectively the test section aspect ratio (L/d) and the eccentricity parameter of the tube in relation to the axis of rotation (d/H). Thus when an experimental programme is envisaged, changes in test-section diameter, for example, alter two of the pertinent groups.

Further, equation (1) shows that rotational speed must be taken into account by the formation of *two* dimensionless groups and it is interesting to note the physical meaning attached to both these groups. Firstly, the rotational Grashof number ($\overline{Gr}_r$) measures the relative importance of centrifugal buoyancy, based on tube centre line centripetal acceleration, to viscous forces.

Dimensional analysis does not yield the actual functional relationship of equation (1), it being usually necessary to furnish a considerable amount of experimental data to empirically determine suitable correlating equations. Even so, it is interesting to speculate here on the expected form of equation (1).

For a given geometric configuration equation (1) becomes

$$\overline{Nu} = \phi_2(\overline{Re}, \overline{Gr}_r, Ro, Pr) \tag{2}$$

which, for zero rotational speed, frequently reduces to the form

$$\overline{Nu}_g = K(\overline{Re}^{a_1})(Pr^{a_2}) \tag{3}$$

where K, a_1 and a_2 are constants which depend on the nature of the flow. A consistent reference temperature for property evaluation will be required of course. It is possible, consequently, that data obtained under rotary conditions may expressed as

$$\overline{Nu} = K\,\overline{Re}^{a_1}Pr^{a_2}\,[1 + \phi_3(\overline{Re}, \overline{Gr}_r, Ro, Pr)] \quad (4)$$

or

$$\overline{Nu}^* = K\overline{Re}^{a_1}\,Pr^{a_2}\,\phi_3\,(\overline{Re}, \overline{Gr}_r, Ro, Pr) \quad (5)$$

where $\overline{Nu}^*$ is an excess Nusselt number given by

$$\overline{Nu}^* = \overline{Nu} - \overline{Nu}_g. \quad (6)$$

Experimental data obtained for free convection influenced by the Earth's gravitational field has shown that the Grashof and Prandtl numbers may be grouped together to form a composite term called the Rayleigh number, Ra_g. That is $Ra_g = Gr_g \times Pr$. If it is assumed, and experimental work [11] appears to substantiate this, that a similar simplification may be used in the present work, then

$$\overline{Nu}^* = K\,\overline{Re}^{a_1}\,.\,Pr^{a_2}\,.\,\phi_3(\overline{Re}, \overline{Ra}_r, Ro) \quad (7)$$

where $\overline{Ra}_r = \overline{Gr}_r \times Pr$ rotational Rayleigh number.

The empirical determination of ϕ_3 in equation (7) is not easy even though a single geometric configuration is considered. A simple multiplication of $\overline{Re}\,.\,\overline{Ra}$, and Ro raised to suitable constant exponents need not be the form of ϕ_3 because of the mutual coupling which exists in the equations expressing conservation of momentum and energy for a combined forced and free convection regime. Further $\overline{Re}$, $\overline{Ra}_r$ and Ro cannot be independently varied during an experiment because, for example, a change in $\overline{Re}$ automatically changes Ro and changes in rotational speed alter both $\overline{Ra}_r$ and Ro.

If the mean Nusselt number is plotted against the rotational Grashof number or, in view of the remarks above, against the rotational Rayleigh number then zero speed data cannot be represented. To overcome this the rotational Rayleigh number may be standardized as regards the applied body force by noting that

$$\overline{Ra}_r = A \times \overline{Ra}_g \quad (8)$$

where

$A = H\Omega^2/g$, centre line acceleration ratio

$\overline{Ra}_g = \dfrac{g\beta\rho^2 cd^3\Delta T}{\mu k}$, gravitational Rayleigh number.

Note that this artifice is not meant to imply that the earth's gravitational acceleration is an important parameter to be included in the original dimensional analysis. It is solely a means to present non-rotating and rotating data on the same basis.

4. DESCRIPTION OF APPARATUS

The apparatus is shown schematically in Fig. 2 and consisted essentially of a built up rotor shaft mounted between self-aligning bearings. Two steel support arms permitted the test section together with its entry and exit chambers to be mounted parallel to the shaft axis with an eccentricity of 6·0 in. Weights fitted diametrically opposite the test section facilitated balance of the rotor which was driven by means of a variable speed electric motor and pulley system. Air was circulated through the test section via internal passages in the rotor shaft and radial connecting tubes by a blower. The air flow rate was measured by means of a bell mouth flowmeter. A photograph of the assembled apparatus is shown in Fig. 3.

The test section was made from brass tubing nominally 12·0-in long by 1·0-in internal diameter with a wall thickness of $\frac{1}{32}$ in. The tube surface was electrically heated with uniformly wound "Ferry" resistance wire and electrical insulation between tube and wire achieved with glass fibre tape. The tube wall temperature distribution was measured at seventeen axial locations and also the circumferential variation at a location nine diameters downstream of the entry. Air inlet and exit temperatures were measured by thermocouples located in the entry plenum and exit mixing chambers respectively. All temperatures were measured with copper–constantan thermocouples with the signals

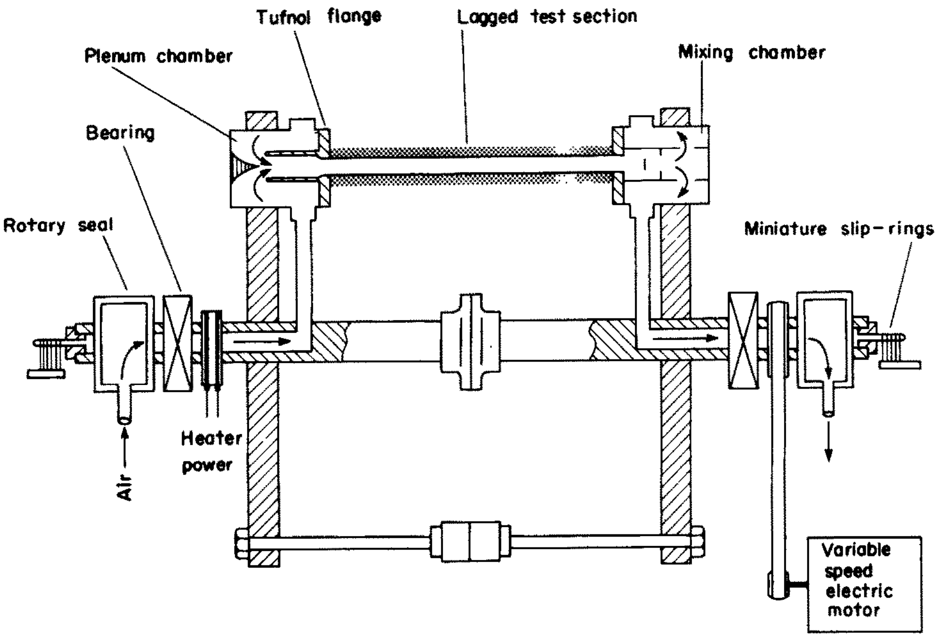

FIG. 2. Diagrammatic layout of rotating tube assembly.

being taken from the rotor to the stationary potentiometer via miniature instrumentation slip rings. Tufnol flanges were attached to both ends of the heater section.

5. EXPERIMENTAL TECHNIQUE

Prior to the commencement of the main experimental programme an investigation of the atmospheric heat loss from the test section was undertaken at various operating conditions. To achieve this, the interior of the test section was completely filled with tightly packed glass-fibre insulation; thus all heat generated in the heater wire would eventually be transferred to atmosphere when steady conditions prevailed. This usually required a period of approximately one hour. Tests were carried out at rotational speeds of 0, 200 and 500 rev/min with various heating rates. When steady conditions prevailed the wall thermocouples were read, and from the resulting temperature profile a mean value (T_{wm}) was found by integration. A family of heat loss curves in the form of ($T_{wm} - T_a$) vs. loss were then plotted. At each rotational speed the calibration was linear with the losses increasing with speed.

For the main experimental programme the following procedure was adopted. The motor drive was adjusted to give the desired test section rotation and the blower throttled to permit the required air flow. In turn, tests were conducted with four air flow rates selected to give nominal Reynolds numbers of 5000, 10000, 15000 and 20000, with rotational speeds of 0, 200 and 500 rev/min. Twelve combinations of Reynolds number and rotational speed were thus studied. For each combination, tests were performed over a tube wall heat flux range such that the local tube wall temperature did not exceed 400°F, this constraint being necessary to avoid mechanical failure of the test section. The actual range of heat flux was 0–3000 Btu/ft^{2}h. When steady conditions were achieved, again requiring about one hour, the following measurements were taken: tube wall temperatures, air inlet and exit bulk temperatures, ambient temperature and pressure, air flow rate, rotational speed, and heater power.

6. METHOD OF DATA EVALUATION

The experimental data was evaluated on a mean and local basis using the calculation procedure listed below. For the air, viscosity, specific heat and thermal conductivity variations with temperature were obtained from a preprint of reference [12] and the coefficient of volume expansion from reference [13]. Density was calculated from the perfect gas equation.

Wall heat flux

The wall heat flux was calculated from the measured air bulk temperature rise, the mass flow rate and the tube inner surface area. The heat loss calibration was used to check this calculated value using the measured heater power consumption. Generally, agreement to within 8 per cent was achieved. It was found that axial conduction along the tube wall was of the order of 1 per cent of the heat transfer to the air and was consequently ignored.

Inner tube wall temperatures

To calculate either mean or local Nusselt numbers the heater tube inside wall temperature distribution is necessary. Actual measurements were taken on the outer surface but, for the range of heat fluxes covered by the experiments, the temperature difference across the tube wall was only of the order $\frac{1}{10}$ degF. This difference was consequently ignored and inner wall temperatures taken as those recorded by the thermocouples.

Local analysis

The local variations in heat transfer for all tests were also investigated. Local variations in Nusselt number were evaluated along the tube axis with the local difference in wall and air bulk temperatures as the characteristic temperature difference, that is $(T_w - T_b)$. The air bulk temperature was taken to vary linearly along the test section and air properties evaluated at the local bulk temperature.

Mean analysis

For the evaluation of mean data the characteristic temperature difference in the Nusselt and Grashof numbers is the difference between the integrated mean wall temperature and the arithmetic mean bulk temperature of the air between inlet and exit, that is $(T_{wm} - T_{bm})$. Properties were evaluated at the arithmetic mean bulk temperature of the air (T_{bm}).

7. RESULTS

Typical graphical representation of the test data obtained is shown in Figs. 4–8 inclusive. For the minimum and maximum Reynolds numbers employed. Figure 4(a) shows the axial wall temperature distribution obtained for nominally the same gravitational Rayleigh number of $1{\cdot}5 \times 10^5$. Air bulk temperatures are also shown. Representative axial wall temperature profiles obtained at various rotational speeds and heat flux conditions are shown in Fig. 4(b) for a constant nominal Reynolds number of 10000. The curves in Figs. 4(a) and 4(b) were obtained at different heat flux levels and are intended to typify the distributions obtained. During all tests no circumferential temperature variations were discernible.

An indication of the variation in heat transfer owing to tube rotation may be seen in Fig. 5(a). Here local Nusselt numbers are plotted against the axial location measured from tube entry. Two sets of curves are presented each for a nominal Reynolds number of 10000 and 20000 respectively. Each set comprises of three individual curves at rotational speeds of 0, 200 and 500 rev/min which respectively give centre line accelerations of 0, 6·84 and 42·6 g. Further, each curve was selected from the available data so that the gravitational Rayleigh number was approximately constant at $1{\cdot}5 \times 10^5$. A further indication of the extent of the changes in heat transfer with rotational speed is given in Fig.

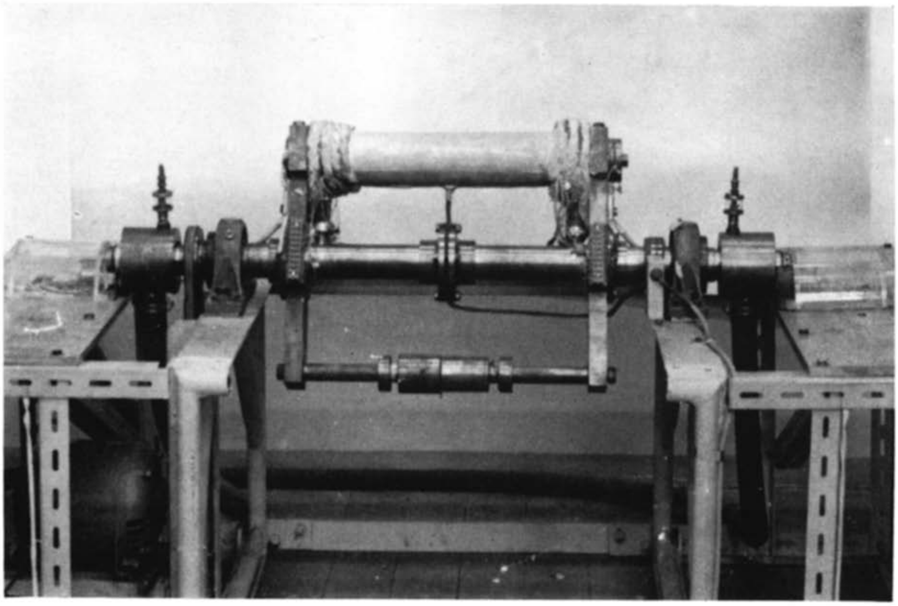

FIG. 3. Apparatus.

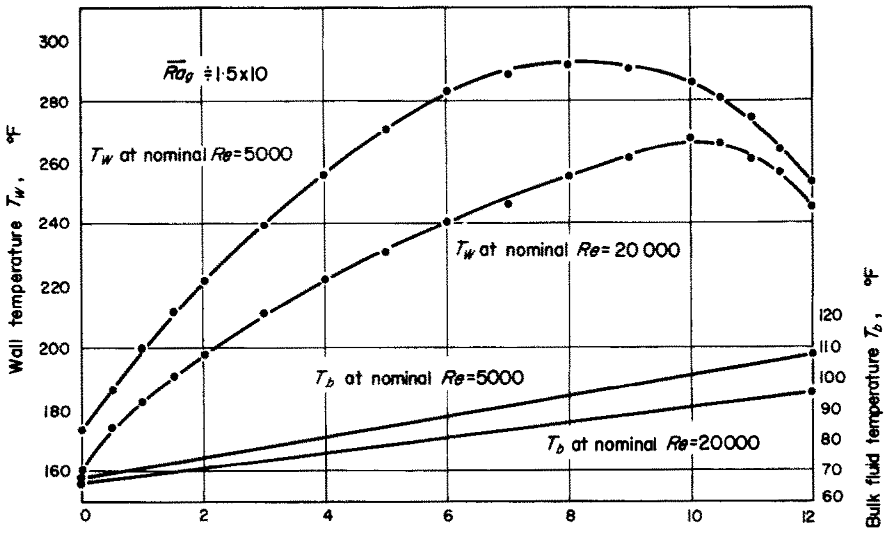

FIG. 4(a). Axial wall temperature profiles—zero rotational speed.

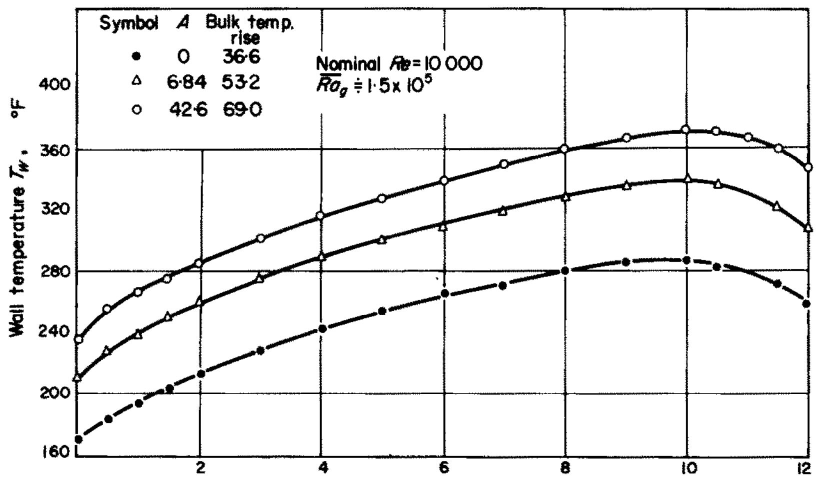

FIG. 4(b). Axial wall temperature profiles—rotating tube.

5(b) where the percentage increase in heat transfer based on zero speed conditions is shown.

The influence of tube rotation on mean heat transfer is shown in Figs. 6, 7 and 8. In Fig. 6 the mean Nusselt number is shown plotted against the mean rotational Rayleigh number giving families of nominally constant Reynolds number curves. It was mentioned earlier that, since zero speed data cannot be represented in the form of Fig. 6, it is more convenient to standardize the Rayleigh numbers using the Earth's gravitational acceleration as the characteristic body force for free convection. This has been done in Fig. 7, where zero speed data is now included.

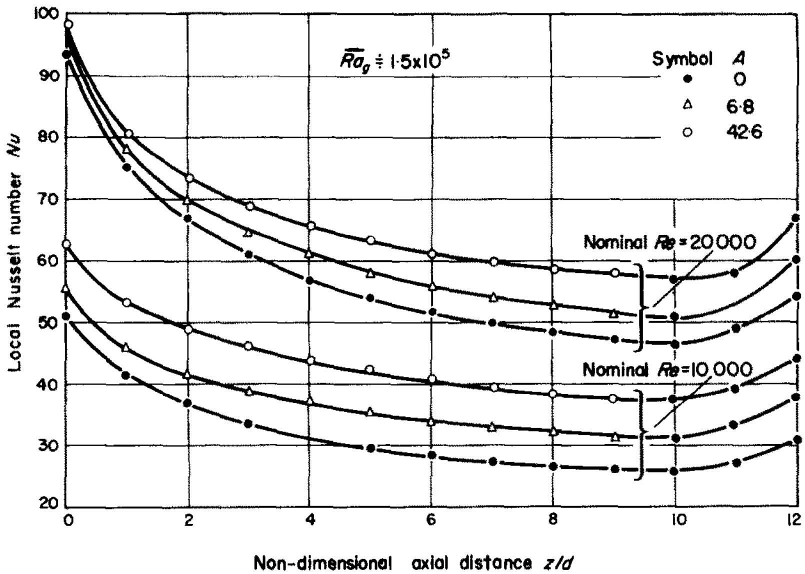

FIG. 5(a). Axial variation of local Nusselt number.

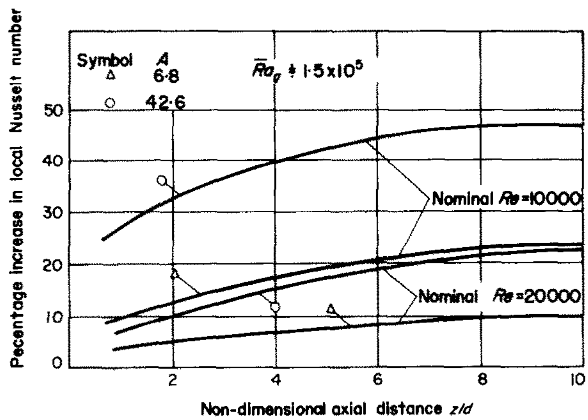

FIG. 5(b). Influence of tube rotation on local heat transfer.

Finally the effect of mean Reynolds number on the mean Nusselt number at different speeds of rotation is shown in Fig. 8.

8. DISCUSSION

Before a detailed discussion of the experimental data obtained with this rather unusual heat-transfer regime is presented, it is necessary to re-emphasize some of the subtleties concerning the internal fluid mechanics involved. Because the nature of the inherent entry swirl is governed by the duct geometry upstream of the test section as well as the rotational speed, data cannot be related to a particular form of entry velocity profile. Rather the data must be related, strictly speaking, to the entire rotating duct system. For this reason it is not intended in this paper to attempt any quantitative correlation of the test data because the final result would not be generally applicable to any tube constrained to rotate in a similar manner. As mentioned in the introductory section the intention is to highlight

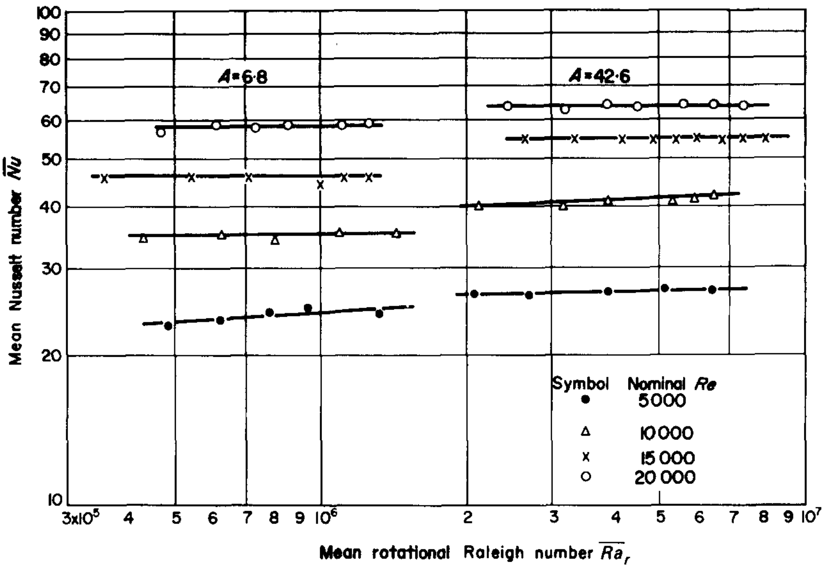

FIG. 6. Variation of mean Nusselt number with mean rotational Rayleigh number.

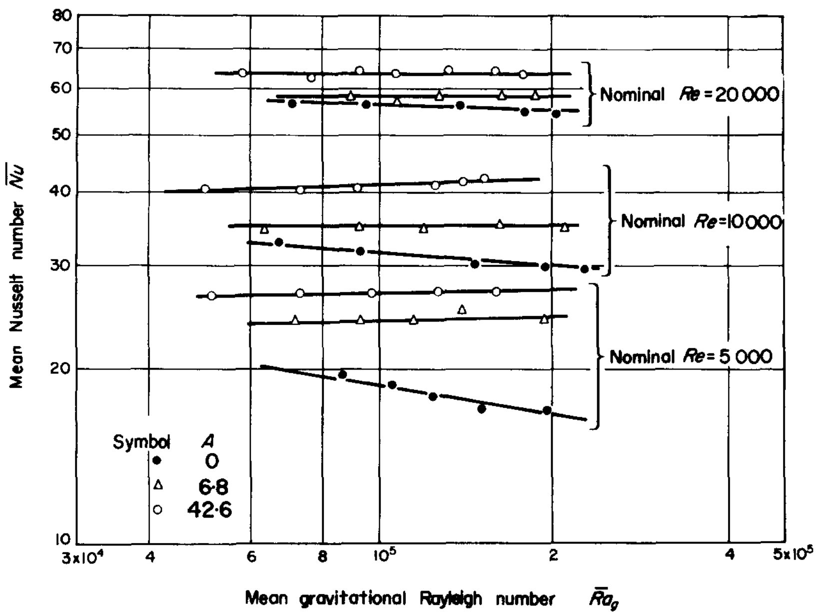

FIG. 7(a). Variation of mean Nusselt number with mean gravitational Rayleigh number and its dependence on rotation aspeed.

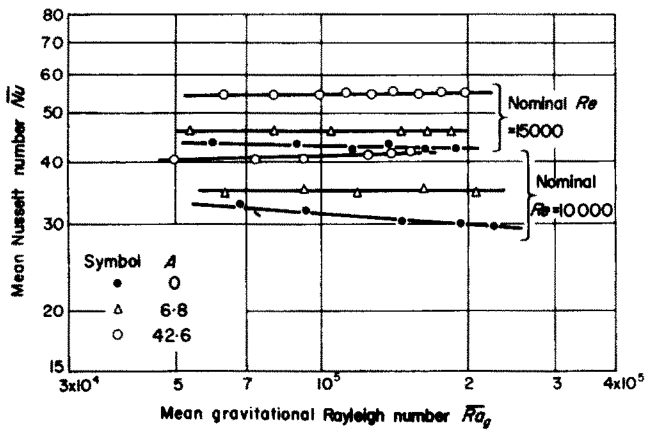

FIG. 7(b). Variation of mean Nusselt number with mean gravitational Rayleigh number and its dependence on rotational speed.

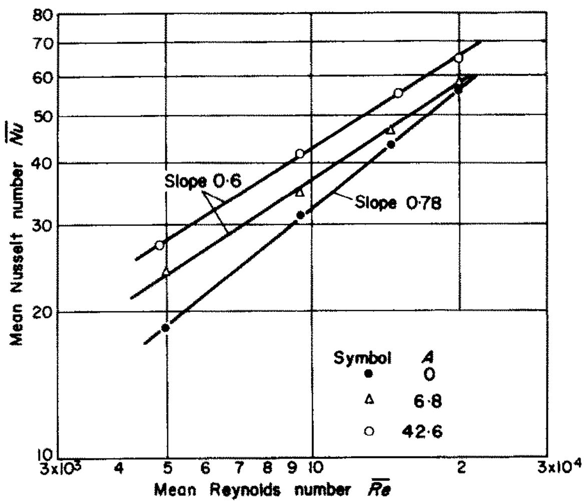

FIG. 8. Variation of mean Nusselt number with mean Reynolds number.

the salient parameters involved and to obtain a qualitative assessment of their relative importance.

Figures 4(a) and 4(b) show typical wall temperature distributions achieved and it can be seen that, for non-rotating conditions, the position of maximum temperature tends to vary with changes in the Reynolds number. The form of the axial temperature distribution is governed by the inter-related influences of thermal entry length, an exit effect due to the mixing chamber and also to axial conduction along the tube. These effects will change with Reynolds number with a consequential variation in the position of maximum tube temperature. Even though no circumferential variations in tube wall temperatures were detected this does not preclude the possibility of local circumferential variations in heat flux being present since the fluid temperature profile will not be axisymmetric.

By reference to Figs. 5(a) and 5(b) it is seen that rotation brings about substantial increases in the local heat transfer and that, for a given value of the acceleration ratio (A), the increase is more marked at the lower Reynolds numbers. The percentage improvement in heat transfer, expressed relative to non-rotating conditions, increased initially in the axial direction until, after about nine diameters, a constant value was approached. A maximum improvement of up to 45 per cent was detected over the range of variables covered in the experimental programme.

At $z/d = 0$, the local Nusselt number should theoretically tend to infinity, but in any real situation some finite value will be attained as seen in Fig. 5(a). For a fixed Reynolds number, the curves should approach the same finite value at $z/d = 0$, provided no change in the nature of the entry velocity occurs. This is clearly not the case and there must, therefore, be a change in the inlet flow conditions as the rotational speed is changed with the nature of the change such as to improve the heat transfer. An inviscid fluid would rotate solidly relative to the tube, but, for a real fluid, the action of viscosity suppresses the swirling tendency in the vicinity of the wall. Nevertheless, the extent of the swirl for a real fluid tends to increase with increased rotational speed. It is this increase in swirl which is mainly responsible for the improvement in heat transfer at the entry to the test section.

At stations downstream of the entry section, the swirl will still be present but it is now accompanied by centrifugal and Coriolis buoyancy. The sustained increases in local Nusselt number along the pipe must obviously arise as a result of the combined effects. However, the relative importance is not yet apparent. It is interesting to note that the increases in heat transfer at the higher Reynolds numbers tested are not so significant as at the lower Reynolds numbers. This is because the influence of rotation is being suppressed to some extent, for the present speed range, by the relatively high axial flow. Experiments with baffles in the inlet plenum chamber showed that the extent of the increase in heat transfer could be reduced by reducing the level of swirl. At z/d values greater than 10 the familiar "exit" effect is seen to occur.

The relative importance of entry swirl and centrifugal buoyancy is better illustrated by considering the mean heat-transfer condition for the test section, as shown in Figs. 6 and 7. In Fig. 6 variations in mean Nusselt number with mean rotational Rayleigh number are shown for a range of rotational speeds and Reynolds numbers. Because each line was obtained at both constant speed and Reynolds number it may also be taken as a line of constant Rossby number. It is seen that, for a given Reynolds number, the mean Nusselt number appears to increase slightly with rotational Rayleigh number. However, data at constant Reynolds number and various rotational speeds do not correlate on single lines when plotted in this way, suggesting a dependency on speed other than that given by the rotational Rayleigh number. As may be seen, step increases in the Nusselt-number occur as the rotational speeds are increased or, in other words, a marked Rossby number influence is present.

It appears from Fig. 6 that the influence of centrifugal buoyancy, as characterized by the rotational Rayleigh number, on the heat transfer is small. However, it is necessary to make comparisons in relation to conditions at zero rotational speed before firm conclusions are made. Since this cannot be directly done with data plotted in the manner of Fig. 6, the entire mean data at all combinations of speeds and Reynolds numbers have been replotted using the standardized gravitational Rayleigh number as shown in Figs. 7(a) and 7(b).

As mentioned previously, it is not implied that gravitational buoyancy is significant at zero rotational speeds even though Figs. 7(a) and 7(b) suggest diminished heat transfer with increases in the gravitational Rayleigh number. The reasons for this downward trend are twofold. Firstly, the nature of the experiment is such that tests are conducted at constant mass flow. Thus, at the higher heating rates, changes in air properties cause a decrease in the Reynolds numbers over the quoted nominal values. Secondly, the definition of the mean dimensionless parameters comprehends "end effects" which again modify the mean data.

In view of this, it is the authors' opinion that the slopes of these zero speed lines should be used as a basis for assessing the importance of centrifugal buoyancy. With these ideas in mind, the significance of centrifugal buoyancy is far more important than initially suggested by Fig. 6. Further, the significance is less marked at the higher Reynolds numbers where the high axial flow is over-riding the buoyancy effects. The overall increases in Nusselt number due to rotation of the tube, for the range of variables covered, was of the order of 40 per cent at $Re = 5000$ and 8 per cent at $Re = 20000$.

An interesting feature which emerges from a series of cross plots of the test data is shown in Fig. 8. Here the variation of mean Nusselt number with Reynolds number is shown for a given value of the gravitational Rayleigh number. The results shown are the same as those obtained at other values of the gravitational Rayleigh number. Generally, as the rotational speed increases, the dependency of heat transfer on Reynolds number diminishes. Thus at high rotational speed it is expected that the combined effects of entry swirl and centrifugal buoyancy will dominate the heat transfer.

9. CONCLUDING REMARKS

The influence of tube rotation on heat transfer to air flowing turbulently in the entrance region of a circular tube has been studied experimentally with the view of assessing the relative importance of the pertinent dimensionless groups. Nominal Reynolds numbers of 5000, 10000, 15000 and 20000 were used with rotational speeds of 0, 200 and 500 rev/min. With the single rotating geometry used, these speeds corresponded to a tube centre line acceleration ratio ranging between 0–42·6 g. The following is a brief list of the important findings.

1. Results show that significant changes in heat transfer occur over the range of Reynolds numbers and rotational speeds used in the experiments.
2. For a given speed of rotation the increase in heat transfer over that achieved under non-rotating conditions is greater at the lower Reynolds numbers.
3. Actual increases in heat transfer arise because of two effects. Firstly, the inherent entry swirl, which is characterized in this case by the Rossby number, causes a step-like change in the heat transfer. Secondly, centrifugal bouyancy causes further increases in heat transfer, the amount of change being characterized by the slope of the Nusselt number–gravitational Rayleigh number curves. In the entry region it appears that the inlet swirl is the more dominant term.
4. No attempt to fit a correlating equation to the data has been attempted because the complex interactions between upstream geometry, flow conditions and rotary geometry would not give a generally applicable result. As mentioned previously, the intention of the investigation was to assess the relative importance of the controlling parameters.

ACKNOWLEDGEMENTS

The authors wish to express their gratitude to Professor J. H. Horlock, for permission to use the equipment and facilities in the department of Mechanical Engineering, The University of Liverpool.

One of the authors (J.F.H.) is indebted to the Ford Foundation for financial support.

REFERENCES

1. W. R. Dean, Flow in curved pipes, *Phil. Mag.* **4**, 208 (1927).
2. A. J. Ede, Effect of an abrupt disturbance of the flow on the local heat transfer coefficient in a pipe, 180° hairpin bend, radius ratio 22:1. N.E.L. Report No. 104, Aug. 1963.
3. E. R. Whittley and H. Barrow, A study of the effect of space variable heat release on fluid flow in a duct,

Thermo Fluid Mech. Convention, I.Mech.E., April 1964.
4. G. Papapanayiotou, Flow in multiple parallel channels with variable heat release, M.Eng. Thesis, University of Liverpool, March 1965.
5. B. R. Morton, Laminar convection in uniformly heated horizontal pipes at low Rayleigh numbers, *Q. Jl Mech. Appl. Math.* **12** (4) (1959).
6. S. T. McComas and E. R. G. Eckert, Combined free and forced convection in a horizontal circular tube, *J. Heat Transfer* **88** (2), 147 (1966).
7. S. N. Barua, Secondary flow in a rotating pipe, *Proc. R. Soc.* A **227**, 133 (1954–55).
8. W. D. Morris, Laminar convection in a heated vertical tube rotating about a parallel axis, *J. Fluid Mech.* **21** (3), 453 (1965).
9. W. D. Morris, A theoretical investigation of the influence of rotation on flow in a tube rotating about a parallel axis with uniform angular velocity, *J. R. Aero Soc.* **69**, 201 (1965).
10. T. H. Davies and W. D. Morris, Thermosyphons: Parts I and II, *Engineers Digest* **26**, 87–91, 80–83 (1965).
11. T. H. Davies and W. D. Morris, Heat-transfer characteristics of a closed loop rotating thermosyphon, Third International Heat Transfer Conference, A.I.Chem.E., Chicago, August 1966.
12. C. G. Rossby, *Tellus* **1**, 54 (1949).
13. J. H. Keenan and J. Kaye, *Thermodynamic Properties of Air*. John Wiley, New York (1945).
14. G. W. C. Kaye and T. H. Laby, *Physical and Chemical Constants*. Longmans, London (1956).

Résumé—Les caractéristiques locales et moyennes de transport de chaleur ont été étudiées expérimentalement pour de l'air (nombre de Prandtl = 0,7) en écoulement turbulent dans la région d'entrée d'un tuyau circulaire tournant autour d'un axe horizontal parallèle à l'axe du tuyau. On n'a étudié qu'une seule géométrie. On a travaillé à des nombres de Reynolds nominaux de 5000 à 25000 et les conditions aux limites correspondent à une distribution initiale uniforme de température et un flux de chaleur pariétal uniforme. On a employé des vitesses de rotation nulle, de 200 et de 500 tours par minute et un rayon de rotation de 15 cm.

Les résultats montraient qu'il se produisait des augmentations sensibles du transport de chaleur dues à la rotation et l'on a supposé que ces changements étaient le résultat de deux effets: c'est-é-dire le tourbillon d'entrée naturel relatif au tube en rotation ainsi que les forces de Coriolis, et les forces volumiques d'origine centrifuge. On a trouvé que le premier effet est plus important que le deuxième.

Zusammenfassung—Die Charakteristik des örtlichen und mittleren Wärme bergangs wurde experimentell für Luft (Prandtl-zahl 0,7) untersucht, bei turbulenter Strömung im Einlaufgebiet eines Kanals mit Kreisquerschnitt der um eine waagerechte Achse parallel zur Kanalachse rotiert. Nur eine geometrische Anordnung wurde untersucht. Nominelle Reynolds-zahlen von 5000 bis 20000 und Grenzbedingungen mit gleichmässiger Anfangstemperaturverteilung und gleichmässiger Wärmestromdichte waren zugrundegelegt. Die Umlaufgeschwindigkeiten betrugen *O*, 200 und 500 U/min und der Umlaufradius 152 mm.

Die Ergebnisse zeigten einen beachtlichen Anstieg des Wärmeübergangs infolge der Rotation. Dieser wird auf zwei Einflüsse zurückgeführt, nämlich auf den Einlaufdrall relativ zum rotierenden Rohr zusammen mit Corioliskräften und zentrifugale Antriebskräfte. Die ersteren erwiesen sich als bedeutsamer als die letzteren.

Аннотация—Экспериментально исследованы локальные и средние характеристики теплообмена для турбулентного течения воздуха (Pr =0,7) во входном участке круглой трубы, вращающейся вокруг своей горизонтальной оси. Изучена только одна геометрия. Использовались номинальные значения критерия Рейнольдса порядка 5000–20000 и следующие граничные условия: однородное начальное распределение температуры и однородный тепловой поток на стенке. Скорости вращения составляли 0,200 и 500 об/мин, а радиус вращения равнялся 6 дюймам.

Результаты показали, что происходит значительная интенсификация теплообмена благодаря вращению трубы. Эта интенсификация имела место благодаря двум эффектам-завихрениям на входе вращающейся трубы совместно с кориолисовыми силами и центробежным силам. Завихрения и кориолисовы силы имеют большее значение, чем центробежные силы.