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FORCED CONVECTIVE HEAT TRANSFER IN UNIFORMLY HEATED HORIZONTAL TUBES

1ST REPORT—EXPERIMENTAL STUDY ON THE EFFECT OF BUOYANCY

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Abstract—In forced convective heat transfer in a horizontal straight tube, buoyancy is likely to have a serious effect on the velocity and temperature fields. This report is devoted to clarifying the effect of buoyancy by means of accurate experiments on the fully developed flow of air in a uniformly heated horizontal tube. Velocity and temperature distributions are measured for large *Re Ra* and in laminar flow these distributions are shown to be essentially different from those known so far, for example, the velocity profile of Poiseuille flow.

Nusselt numbers are calculated from the measured velocity and temperature distributions and shown to be about twice as large as those calculated by neglecting the effect of the secondary flow caused by buoyancy at $Re\,Ra = 4 \times 10^5$. In turbulent flow, buoyancy has little effect on the velocity and temperature fields, but the critical Reynolds number is shown experimentally to be affected by the secondary flow.

NOMENCLATURE

- a, inside diameter of a tube;
- r, distance from the central axis in a cross-section; downward is positive;
- t, fluid temperature at a distance r from the centre;
- t_c, fluid temperature on the central axis;
- t_w, wall temperature;
- w, fluid velocity in an axial direction;
- w_c, fluid velocity on the central axis;
- z, distance from the heating point downstream along the tube axis;
- β, coefficient of thermal expansion;
- κ, thermal diffusivity;
- ν, kinematic viscosity;
- λ, thermal conductivity;
- τ, dt/dz;
- Ra, Rayleigh number, $g\,\beta\,\tau_a^4/\kappa\,\nu$;
- Re, Reynolds number, $2U_m\,a/\nu$;
- Nu, Nusselt number, $2\alpha\,a/\lambda$.

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1. INTRODUCTION

THE THEORETICAL study of convective heat transfer with laminar flow in a tube was begun by Graetz and Nusselt. Levêque, Drew, Yamagata, Abramovitz, etc., solved the same problem for the condition that the physical properties of fluid are constant and the flow is axisymmetrical. These theoretical analyses are limited to a fully developed laminar flow, i.e. Poiseuille flow, and the temperature distribution is obtained by putting this velocity distribution into the energy equation. These theoretical analyses have nothing to do with whether the tube axis is horizontal or vertical. But it is observed by experiments on laminar flow that the results are considerably affected by the inclination of the tube, when the heat flux at the wall is large. More particularly, the results obtained by experiments on convective heat transfer with laminar flow in horizontal tubes show higher heat-transfer coefficients than those obtained by the theoretical studies mentioned above. Drew [1] introduced empirical formulae by correlating experimental results, and pointed out that they were 20–50 per cent higher than the values resulting from the theoretical analysis of Levêque and others. Later Colburn [2] correlated the experimental results for air and water flowing in comparatively large tubes and proposed an empirical formula. McAdams [3] gave the following formula of heat-transfer coefficient for constant wall temperature, referring to the results of experiments with oil which were made later, and also considering the formula which Martinelli *et al.* found by theoretical analysis for upward flow in a vertical tube:

$$\frac{\alpha_{ca} D}{\lambda_b}\left(\frac{\eta_w}{\eta_b}\right)^{0\cdot 14} = 1\cdot 75\left[\frac{W\, C_{pb}}{\lambda_b\, L} + 0\cdot 04\left(\frac{Gr_b\, Pr_b\, D}{L}\right)^{0\cdot 75}\right]^{1/3} \tag{1}$$

In this formula, the suffixes w and b stand for the values at the wall and average fluid temperatures; α_{ca} is based upon a temperature difference which is given by the relation $t_w - (t_1 + t_2)/2$, where t_1 and t_2 stand for the fluid temperature at the starting point and that at the terminal point of heating. W is a weight flow rate, η the coefficient of viscosity, λ the thermal conductivity, C_p the specific heat at constant pressure, D the inside diameter of a round tube, L the length of heat transfer, Pr the Prandtl number and Gr the Grashof number.

Recently Jackson *et al.* [4] made experiments on air flowing in constant temperature horizontal tubes, including the entry length, and introduced the following formula:

$$\frac{\alpha_m D}{L} = 2\cdot 67\left[\left(\frac{W\, C_{pm}}{\lambda_m\, L}\right)^2 + (0\cdot 0087)^2\,(Gr_m\, Pr_m)^{1\cdot 5}\right]^{1/6} \tag{2}$$

Here the suffix m shows the value based upon the average fluid temperature. It seems that the vectorial sum of the forced convective flow term and the natural convective flow term is taken into consideration in this equation. Oliver [5] derived a similar empirical formula to equation (1) by experiments with ethyl alcohol, water and an 8 % glycerol solution. In this formula physical properties for the right-hand side are taken at the average temperature and its second term is expressed as $0\cdot 0083\,(Gr_m\, Pr_m)^{0\cdot 75}$, excluding D/L.

When the difference between fluid temperature and wall temperature is small, and when the effect of natural convection and the change in the physical properties with temperature are small, these formulae become $1\cdot 75\,(W\, C_p/\lambda)^{1/3}$, tending to the expressions obtained theoretically. It is therefore considered that the results of the above-mentioned theoretical analyses can be applied only when the heat flux is small. When a horizontal tube is used as a part of a heat-transfer device, and a large heat flux is to be treated, the difference between the fluid temperature and that of the wall is large. In this case the effect of the change of viscosity with temperature is approximately covered by the term $(\eta_w/\eta_b)^{0\cdot 14}$ in the left-hand side of equation (1). However, as for the effect of natural convection, the results of the experiments so far made differ considerably. Almost all of these experiments have been conducted under the conditions of constant wall temperature and large L/D. In the case of a constant temperature tube, though natural convection has a remarkable influence in the temperature entry length, its effect decreases with the increase of the tube length as the fluid temperature approaches to the wall temperature. However, it is considered that natural convection has a remarkable effect when the heat flux at the wall or the axial temperature gradient of the tube is constant, or the difference between fluid temperature and wall temperature is large just as in the temperature entry length even for a constant wall temperature.

When the flow is fully developed and has a Poiseuille distribution, and the heat flux at the wall or the axial temperature gradient of the tube is constant, it is known that the Nusselt number is calculated as follows from the energy equation:

$$Nu_0 = \tfrac{48}{11} \tag{3}$$

Morton [6] made a theoretical analysis by setting up the equation of motion, including the buoyancy term and the equation of energy simultaneously under the condition mentioned above, and obtained the following formula by a perturbation method for low Rayleigh number Ra:

$$Nu = \frac{48}{11}\left\{1 + (0{\cdot}0505 - 0{\cdot}1054\, Pr + 0{\cdot}3334\, Pr^2)\left(\frac{Re\, Ra}{4608}\right)^2 + \ldots\right\} \quad (4)$$

This formula is applicable in a range of $Re\, Ra < 3000$, but within this range the effect of natural convection does not exceed several per cent. As the Rayleigh number is proportional to the fourth power of the diameter, natural convection has a small effect on convective heat transfer in a horizontal tube of small diameter. But in comparatively large tubes, the effect seems to be quite considerable. The present research was intended to investigate experimentally and theoretically the effect of buoyancy on forced convection in horizontal tubes of larger diameter with larger heat fluxes. In order to examine the effect of buoyancy, this first report will consider experimentally the case of constant heat flux.

2. APPARATUS

2.1 *Measurement of velocity and temperature*

The outline of the experimental apparatus is shown in Fig. 1. Air from a centrifugal blower enters a settling chamber and is passed to a heating section by way of a velocity entry length of 7 m. All straight tubes are made of brass with 35·6 mm inside diameter, and have a wall thickness of 1·2 mm. Thin tubes are used in order to make it possible to produce a sufficient temperature gradient in the axial direction. The tube is 14 m long overall. The settling chamber, of sufficient volume, is provided with honeycombs so that flow is laminar over a wide Reynolds number range, and the velocity distribution in fully developed unheated flow is the Poiseuille distribution. The flow rate of the air is adjusted roughly by a butterfly-shaped valve attached just after the blower, and more precisely by controlling the rate of rotation of the blower. As a consequence, experiments can be conducted over a Reynolds number range from 10^2 to $1{\cdot}3 \times 10^4$. The straight tube is not heat-insulated over the initial 7 m down the settling chamber, but heating is applied downstream, by means of 0·5 mm nicrome wires wound around the tube at constant pitch. As shown in Fig. 1, thermocouples are attached at 13 points of the tube wall, and by measuring the temperatures at these points it is arranged that the temperature gradient of the tube in the axial direction is constant after the temperature entry length.

Velocity distributions are measured by calibrated, cylindrical, yaw probes. These probes are

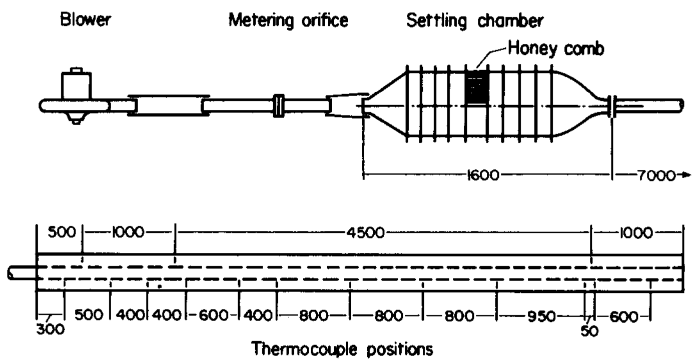

FIG. 1. Schematic diagram of the experimental apparatus (units in mm).

of stainless steel, 0·8 mm in diameter, and have one pressure hole of 0·2 mm diameter. A probe is inserted throughout from one side of the wall to the other side as shown in Fig. 2.

Temperature distributions are measured by means of a T-shaped thermocouple probe. A traversing device and a detailed diagram of the probes are shown in Fig. 2.

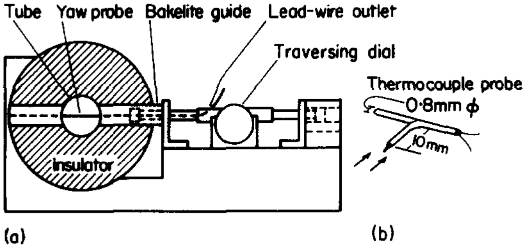

FIG. 2. The traversing device and the thermocouple probe.

2.2 *Measurement of critical Reynolds number*

There are many experimental studies of the critical Reynolds number of tube flow. When the initial turbulence intensity of the flow is very high, it is well known that the critical Reynolds number is about 2000, but in a flow of low turbulence level, Rotta [7] reported a critical Reynolds number as high as $1{\cdot}3 \times 10^4$. Judging from these experimental results, the critical Reynolds number is considered to depend on the intensity of turbulence. Therefore, when there is a secondary flow, as treated in this paper, the critical Reynolds number may depend on the intensity of the secondary flow. In these experiments the critical Reynolds number is measured by a hot-wire anemometer fitted to the measuring section. The initial turbulence level is varied by inserting a disk with many small holes as a turbulence generator just after the settling chamber, and the intensity of the secondary flow by adjusting the electrical power in the heaters around the tube. The critical Reynolds number is taken as soon as there appears an intermittency in the signal from the hot-wire anemometer when increasing the Reynolds number gradually, under the condition of constant heat flux.

3. THE EXPERIMENTAL RESULTS AND DISCUSSION

3.1 *Preliminary experiments*

Preliminary experiments were carried out to study the velocity distribution in a round tube. Before installing heat insulators around the tube, the velocity distributions were measured both at the finishing point of the velocity entry length, 7 m down the tube from the entrance of the tube, and at the measuring section, 13 m downstream. As has been mentioned above, special attention was paid to obtaining an axisymmetrical velocity distribution in the laminar regime, and it was hoped to obtain a Poiseuille distribution. It was, however, difficult to achieve an axisymmetrical distribution. As the result of repeated experiments, it was found that this difficulty was due to the slight curvature of the tube axis, and to the existence of a small axial temperature variation. The latter was about 1 degC and resulted in a slight difference between the temperature of the tube wall and that of the fluid. The supporters of the tube were therefore so adjusted, by micrometer devices, as to keep the tube axis in a completely straight line, and great attention was paid to keeping the wall temperature constant within 1 degC. Thus the velocity distributions shown in Fig. 3 were obtained in the vertical and horizontal planes at the measuring section. The solid line in this figure shows the Poseuille

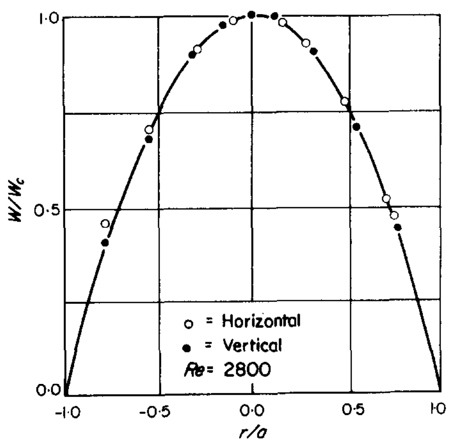

FIG. 3. Velocity distributions when not heated.

distribution, and the experimental values agree with it very well, demonstrating the reliability of the apparatus and experimental method.

3.2 *Velocity and temperature distributions in the heating experiments*

As heating was begun at 7 m downstream of the entrance of the straight tube, z was measured along the tube axis from this point. As the present report is concerned with fully developed velocity and temperature fields under the constant heat flux condition at the wall, we first confirm that the fully developed condition is being satisfied. In this report, air is used as the working substance, and its specific heats are almost constant since the fluid temperature in the axial direction does not change very greatly, so that the condition of constant temperature gradient of the tube in the axial direction agrees with the condition of constant heat flux at the wall.

The velocity and temperature distributions in the vertical and horizontal planes were measured at two points: $z = 6$ and $z = 6{\cdot}5$ m where the axial temperature gradient was kept completely constant by adjusting the heating power. As an example, the temperature distribution of the tube in the axial direction at $Re = 4250$, temperature gradient $\tau = 11{\cdot}3$ degC/m and $Ra = 39{\cdot}3$, is shown in Fig. 4. Figure 5 shows the temperature distribution and the velocity distribution in the vertical plane including the centre of the tube; it will be seen that, near the point $z = 6$ m, both the flow field and the temperature field are fully developed, so that the expected condition of the present study was satisfied. In these figures, t_w stands for wall temperature, t_c for temperature on the central axis, w_c for velocity on the centre. We made a similar experiment at $Re = 1890$ and $Ra = 45{\cdot}4$ and confirmed that both velocity and temperature fields were completely developed. Therefore the following measurements were made only at $z = 6$ m, and this point will be called the measuring point. We measured wall temperature distributions around the circumference in the cross-section normal to

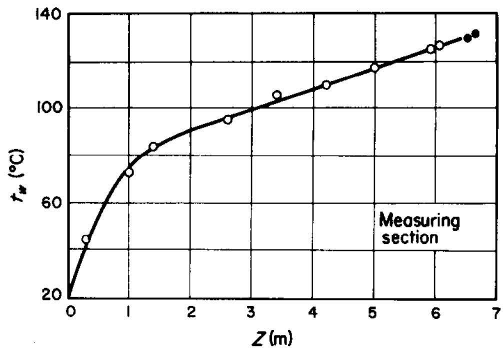

FIG. 4. Temperature distribution in the axial direction.

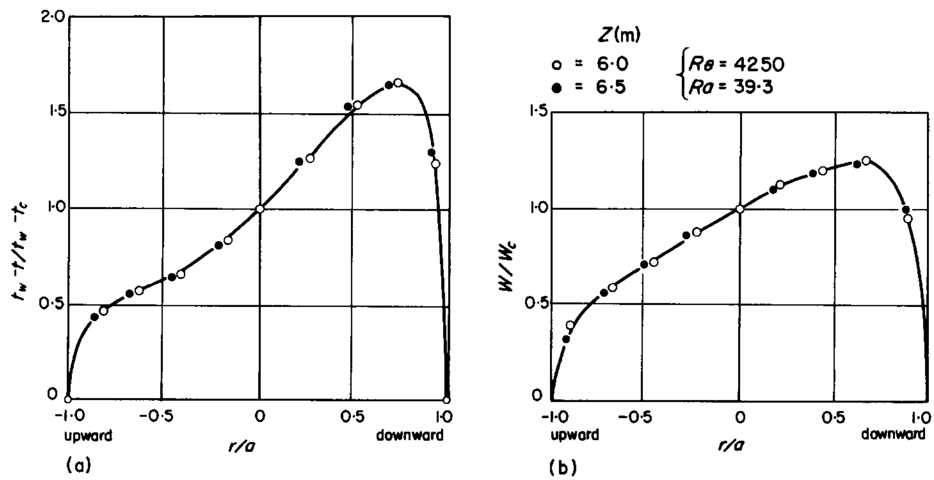

FIG. 5. Temperature and velocity distributions of fully developed flow.

the tube axis, and found that the temperature variation was very small, within 0·3 degC.

In these experiments, the electrical input to each heater was almost constant, and it was confirmed that the condition of constant heat flux at the wall agreed with the condition of constant temperature gradient in the present experiment. As can be seen in Fig. 4, the part from $z = 0$ to $z = 3$ m is the temperature entry length, and after that the temperature gradient becomes constant. Though it is not shown in Fig. 4, far downstream from $z = 6{\cdot}5$ m the axial temperature gradient decreases and at the outlet the temperature gradient is considered to tend to zero. In traditional experiments of convective heat transfer in tubes with a temperature gradient, mixed mean temperatures are measured, with mixing chambers placed upstream and downstream of the tube. However, according to our confirmation, as stated above, the axial temperature gradient changes even though the tube is completely insulated near the mixing chambers, and the intended condition of constant temperature gradient can hardly be fulfilled in all the length of the tube. This may be all right if the tube is long enough compared with the length of the region affected by this influence, but in this case as the physical properties change substantially, the results need to be examined.

3.3 *Nusselt numbers*

3.3.1 *Laminar flow.* As previously mentioned, velocity and temperature fields at $z = 6$ m were completely developed, therefore measurements were made at $z = 6$ m.

Considering first the horizontal direction, the temperature distribution closely resembles the velocity distribution, and both fall on the same curve if expressed as ratios to the values on the central axis. Examples are shown in Fig. 6. Velocity and temperature distributions, Nusselt numbers, etc., in laminar flow are known to be functions of the product of Reynolds number and Rayleigh number if the Prandtl number is given, according to Morton's analysis for small *Ra* [6] and our analysis for large *Ra* (to be reported soon). Figure 7 shows the measured velocity and temperature distributions de-dimensionalized by the values on the central axis when *Re* is different but the product of *Re Ra* is equal. To calculate the Reynolds and Rayleigh numbers, physical properties are taken at the measuring point. In these figures, r of the axis of abscissa is the distance from the centre with the positive direction downwards and a is the inside diameter of the tube. As the wall temperature is higher than the fluid temperature in a section in this experiment, the fluid near the wall is heated, and the upward flow along the tube wall is caused by buoyancy. The two upward flows along both sides of the tube wall coincide with each other at the upper part of the tube, change direction and come down in the central part. Therefore, it is considered that these flows form a pair of vortices with axes in the horizontal direction, symmetrical against the vertical plane at the axis. Due to this secondary flow, the velocity in the axial direction shows a distribution which is concave downwards.

Figure 8 shows the temperature and velocity distributions in two such typical cases when *Re* is equal, and *Re Ra* is different because the wall temperature and the wall temperature gradient are different. Because of the difference in *Re Ra*, the temperature distributions are different, but the velocity distributions show little change. In our experiments, velocity distributions do not show any very remarkable difference owing to *Re* number, *Ra* number, heat flux at the wall, etc., as stated above.

It is seen from Figs. 5–8 that, in the layer near the wall, velocity and temperature change abruptly. In other words, the situation in this thin layer is remarkably different from that in the inner part. As mentioned before, in the layer around the circumference, the fluid heated along the wall rises because of buoyancy and descends in the central part. When *Re Ra* is large, this secondary flow is strong, and so the velocity distribution in the axial direction becomes utterly different from that of Poiseuille flow. Therefore when the value of *Re Ra* is large, the

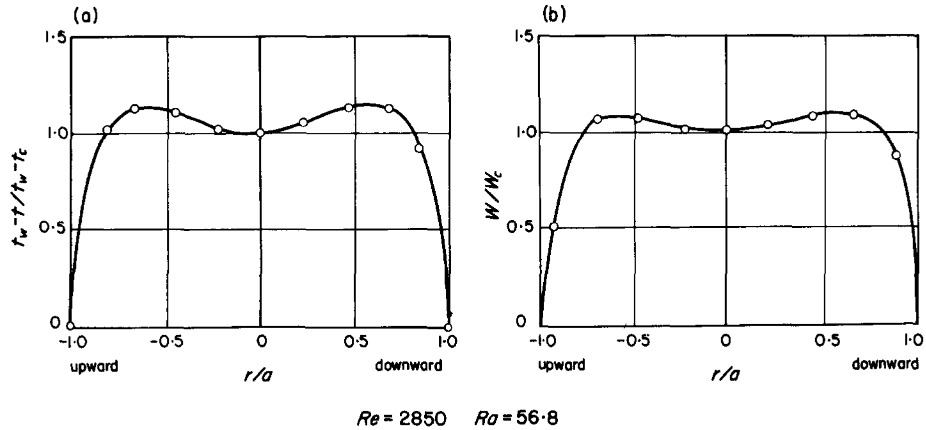

FIG. 6. Temperature and velocity distributions in the horizontal direction.

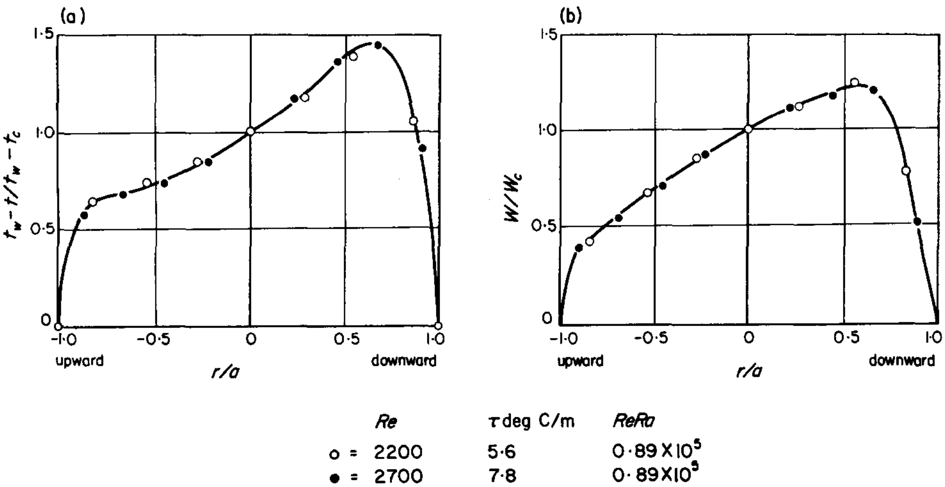

FIG. 7. Temperature and velocity distributions in the axial direction in case of equal *Re Ra*.

assumption used by Morton in his theoretical analysis that the velocity distribution is not very different from the Poiseuille distribution is not correct, so that the results of the present experiments may not be compared with Morton's analysis. In this connection we might assume from the results of our experiments that when the value of *Re Ra* is large, the influence of viscosity and thermal conductivity is dominant only in the thin layer around the circumference of the tube, and the remainder including the centre, is not influenced by them and the flow there is downward. An analysis by this model will be given in our next report.

Next we calculate the Nusselt number by using the temperature and velocity distributions obtained from the experiments mentioned above. The centre of the tube cross-section will be taken as origin of x and y axes; and on the x axis the downward direction will be positive and the y axis will be horizontal. Dividing the section into many small parts, the temperature and the

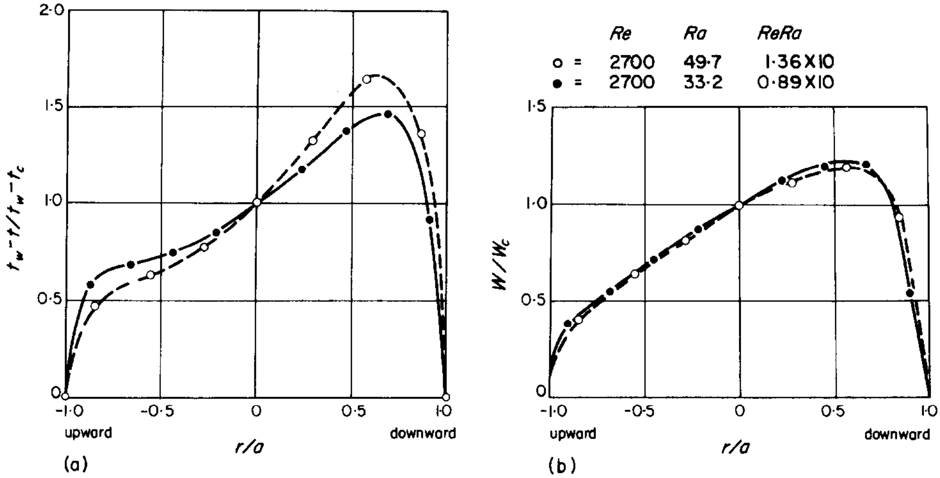

FIG. 8. Temperature and velocity distributions in the axial direction in case of equal *Re* and different *Re Ra*.

velocity in an arbitrary part of the co-ordinate x_i, y_i will be denoted by t_{ij} and w_{ij}; and its area will be denoted by $\Delta x_i \, \Delta y_i$. As the distribution of velocity or temperature in the horizontal direction at an arbitrary point is considered to be similar to the horizontal distribution at the centre, t_{ij} and w_{ij} are calculated from the measured distributions in the horizontal and vertical planes including the centre. Then, the mixed mean temperature is:

$$t_m = \frac{\sum w_{ij}\, t_{ij}\, \Delta x_i\, \Delta y_i}{\sum w_{ij}\, \Delta x_i\, \Delta y_i} \tag{5}$$

If the temperature gradient in the axial direction is denoted by τ, the heat transferred at the wall per unit axial length q is calculated by means of the average velocity u_m as follows:

$$q = a\, \gamma\, C_p\, \tau\, u_m/2 \tag{6}$$

where γ stands for specific weight. Thus the Nusselt number Nu is as follows:

$$Nu = 2aq/\lambda(t_m - t_w). \tag{7}$$

Figure 9 shows the ratio Nu/Nu_0 in relation to *Re Ra*, where Nu is calculated from equation (7) and Nu_0 from equation (3) for constant heat flux at the wall without any secondary flow. As

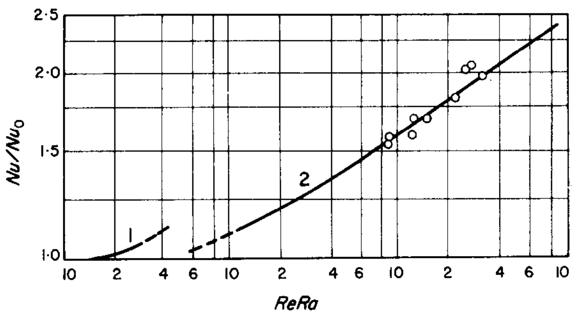

FIG. 9. (Nu/Nu_0)–*Re Ra* diagram (laminar flow).

stated before, the Nusselt number is considered to be a function of *Re Ra* and *Pr* in the laminar regime, and in these experiments the Nusselt number is considered to be a function only of *Re Ra* because the experiments were made with air, for which $Pr = 0{\cdot}72$.

In a flow with a larger Prandtl number than 1, the layer near the wall where the temperature changes abruptly becomes thinner, and the Nusselt number increases at the same *Re Ra*. It is worth noticing that even when air is used as a working fluid, Nu becomes substantially larger than Nu_0 and is nearly double its laminar flow value when *Re Ra* exceeds 10^4.

In Fig. 9, curve 1, starting at about $Re\,Ra = 10^3$, shows the results of Morton's theoretical analysis for small $Re\,Ra$, and only the region of the solid line is applicable. Our experiments were made at larger values of $Re\,Ra$ and the solid line 2 is so determined as to correlate the experimental results in laminar flow and is expressed by the following formula for the local Nusselt number Nu:

$$Nu = 0{\cdot}61(Re\,Ra)^{1/5}\left(1 + \frac{1{\cdot}8}{(Re\,Ra)^{1/5}}\right) \quad (8)$$

3.3.2 *Turbulent region.* As will be explained later, in the present experimental apparatus, when the tube is not heated, the critical Reynolds number is 7700 as long as there is no turbulence generator at the entrance of the tube, and far above this critical Reynolds number the flow is completely turbulent. In order to examine the effect of buoyancy on the Nusselt number in the turbulent regime, we made the same experiments for fully developed turbulent flow with constant heat flux at the wall as for laminar flow. As examples of the results in the turbulent region, Fig. 10 shows the temperature and velocity distributions in the vertical direction for $Re = 10^4$ and $1{\cdot}2 \times 10^4$.

In turbulent flow also, the highest point of the temperature distribution shifts slightly downward from the centre and the secondary flow caused by buoyancy is considered to exist. However, the effect of turbulence is so strong that the effect of the secondary flow is not as remarkable as in the case of laminar flow, and Nusselt numbers calculated from these distributions show a good agreement with Colburn's formula for turbulent heat transfer in tubes for $Pr = 0{\cdot}72$:

$$Nu = 0{\cdot}0204\,Re^{4/5}. \quad (9)$$

Therefore, for turbulent flow, it is not necessary to consider the influence of buoyancy on heat transfer even in case of constant heat flux at the wall.

3.4 *Critical Reynolds numbers*

In Fig. 11 critical Reynolds numbers Re_{cr} are measured and plotted against $Re\,Ra$. These empirical points are divided into two groups: the group of larger Re_{cr} is obtained when there is little turbulence at the entrance of the tube, and the group of smaller Re_{cr} is obtained when there is much turbulence caused by a turbulence generator. These closely resemble the relation (8) between the critical Reynolds number and the Dean number of a flow in a curved pipe. As

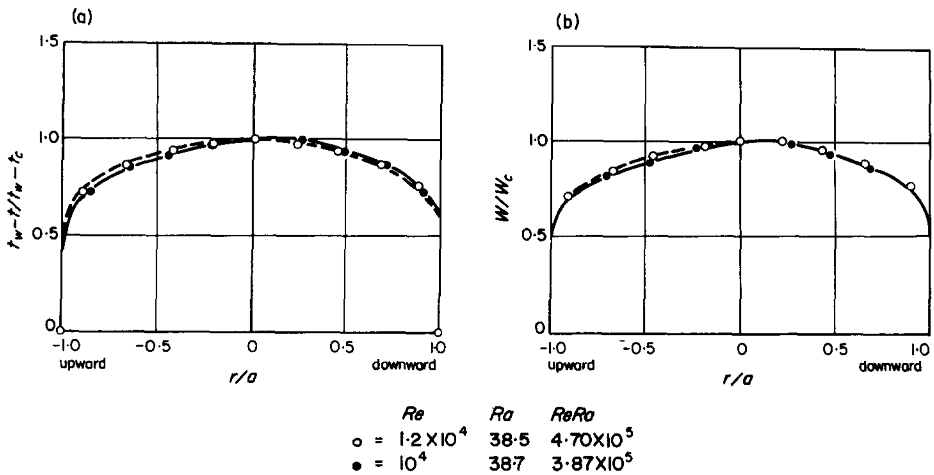

FIG. 10. Temperature and velocity distributions in the axial direction (turbulent flow).

is shown in Fig. 11, when the turbulence level at the entrance of the tube is high, the critical Reynolds number starts at about 2×10^3 and increases with Rayleigh number. This is due to the effect of the secondary flow in suppressing the turbulence level. The solid line 1 in Fig. 11 shows the following formula:

$$Re_{cr} = 128(Re\,Ra)^{1/4}. \qquad (10)$$

Contrary to this, when there is no turbulence generator, the critical Reynolds number is higher and in our experiment the critical Reynolds number without heating Re_{cro} was about 7700. When the flow is heated, the secondary flow has

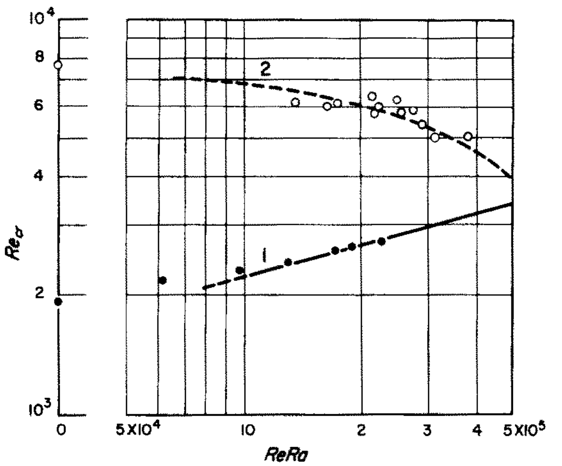

FIG. 11. Critical Reynolds number.

a similar action to turbulence, and so Re_{cr} is considered to decrease. In this case Re_{cr} is given approximately by the following equation when $Re\,Ra < 5 \times 10^5$:

$$Re_{cr} = Re_{cro}/(1 + 0{\cdot}14\,Re\,Ra \times 10^{-5}) \qquad (11)$$

In Fig. 11, the solid line 2 shows the curve expressed by equation (11). It seems that when $Re\,Ra$ is large, the secondary flow caused by buoyancy makes Re_{cr} tend to the same value whether the turbulence level at the entrance of the tube is high or low.

4. CONCLUSION

A precise experimental study of heat transfer with air flow in a horizontal round tube under the condition of constant heat flux at the wall has been carried out and the results that were obtained are summarized below.

(1) Under the condition of constant heat flux at the wall, the velocity field and temperature field are both completely developed at a position far enough downstream of the temperature entry length. When the product of the Reynolds number Re and the Rayleigh number Ra is more than 10^4, the secondary flow is stronger and it forms a pair of vortices with an axis in the horizontal direction symmetrical against a vertical plane.

(2) For laminar flow, when the product $Re\,Ra$ is large, measured velocity distributions are remarkably different from the velocity distribution of Poiseuille flow. The axial velocity shows a distribution concave downwards and the temperature distribution has a similar profile to the velocity distribution. It is also shown that the velocity and temperature change remarkably in a thin layer along the circumference of the tube wall.

(3) In laminar flow, the effect of buoyancy on the local Nusselt number Nu appears at about $Re\,Ra = 10^3$, and Nu increases substantially due to the secondary flow caused by buoyancy and becomes twice as large as the Nusselt number without the secondary flow at $Re\,Ra = 4 \times 10^5$. We get the following correlation formula:

$$Nu = 0{\cdot}61(Re\,Ra)^{1/5}\left\{1 + \frac{1{\cdot}8}{(Re\,Ra)^{1/5}}\right\}$$

(4) For turbulent flow, similar experiments have been carried out, but the effect of the secondary flow on the temperature and velocity distributions was very small and no effect on Nusselt number was observed.

(5) The effect of the secondary flow caused by buoyancy on the critical Reynolds number has been experimentally studied by means of a hot-wire anemometer. With a low turbulence level at the entrance of the tube, the critical Reynolds number decreases with increasing Rayleigh

number, while with a high turbulence level the critical Reynolds number increases with the Rayleigh number.

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Résumé—Il semble que la poussée d'Archimède ait un effet important sur les champs de vitesse et de température, dans le transport de chaleur par convection forcée à l'intérieur d'un tube rectiligne horizontal.

Ce rapport a pour but de rendre apparent l'effet de la poussée d'Archimède au moyen d'expériences précises sur l'écoulement d'air entièrement développé à l'intérieur d'un tube chauffé uniformément. Les distributions de vitesse et de température sont mesurées pour de grandes valeurs du produit *Re Ra* et l'on montre que dans l'écoulement laminaire ces distributions sont essentiellement différentes de celles connues auparavant, comme, par exemple, le profil de vitesse de l'écoulement de Poiseuille.

Les nombres de Nusselt sont calculés à partir des distributions de vitesse et de température et l'on montre qu'ils sont environ deux fois plus grands que ceux calculés en négligeant l'effet de l'écoulement secondaire provoque par la poussée d'Archimède pour $Re\ Ra = 4 \,.\, 10^5$. En écoulement turbulent, la poussée d'Archimede a peu d'effet sur les champs de vitesse et de température, mais l'on montre par l'expérience que le nombre de Reynolds critique est affecté par l'écoulement secondaire.

Zusammenfassung—Beim Wärmeübergang unter erzwungener Konvektion in einem waagerechten, geraden Rohr kann der Auftrieb einen spürbaren Einfluss auf die Geschwindigkeits- und Temperaturfelder ausüben. Diese Arbeit soll mit Hilfe genauer Versuche den Einfluss des Auftriebs auf eine voll ausgebildete Luftströmung in einem gleichmässig beheizten waagerechten Rohr klären. Beschwindigkeits- und Temperaturverteilungen wurden für grosse *Re Ra* gemessen; für Laminarströmung zeigen sich Verteilungen, die wesentlich von bisher bekannten abweichen, zum Beispiel vom Geschwindigkeitsprofil der Poiseuilleströmung.

Bei $Re\ Ra = 4 \times 10^5$ wurden Nusseltzahlen aus den gemessenen Geschwindigkeits- und Temperaturverteilungen berechnet und es zeigt sich, dass diese etwa doppelt so gross sind als die unter Vernachlässigung des Einflusses der vom Auftreib bewirkten Sekundärströmung errechneten Werte. Bei turbulenter Strömung hat der Auftrieb wenig Einfluss auf die Geschwindigkeits- und Temperaturfelder, doch lässt sich experimentell zeigen, dass die kritische Reynoldszahl von der Sekundärströmung beeinflusst wird.

Аннотация—Предполагается, что на поля скорости и температуры в процессе теплообмена при вынужденной конвекции в горизонтальной прямой турбе сильное влияние оказывает естественная конвекция. Эта работа посвящена выяснению влияния подъемной силы с помощью точных экспериментов в полностью развитом потоке воздуха в равномерно нагретой горизонтальной трубе. Измерены профили скорости и температуры при больших числах *ReRa*. Установлено, что при ламинарном течении эти профили сильно отличаются от известных, например, от распределения скорости для пуазейлевского течения.

На основании измеренных распределений скорости и температуры рассчитаны числа Нуссельта, которые оказались почти в два раза больше значений, вычисленных при $ReRa = 4 \times 10^8$ без учета влияния вторичных эффектов из-за естественной конвекции.

В турбелентном потоке подъемная сила меньше влияет на распределения скорости и температуры. Экспериментально установлено, что на критическое число Рейнольдса вторичные эффекты оказывают влияние.