Int. J. Heat Mass Transfer. Vol. 20, pp. 51–56. Pergamon Press 1977. Printed in Great Britain

LOCAL HEAT TRANSFER FROM A HORIZONTAL CYLINDER TO AIR IN CROSS FLOW: INFLUENCE OF FREE CONVECTION AND FREE STREAM TURBULENCE

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(*Received* 28 *October* 1975 *and in revised form* 26 *April* 1976)

Abstract—Local measurements have been made of the heat transfer from a horizontal circular cylinder to air in cross flow in forced convection and mixed convection. The studies have been conducted with the constant heat flux boundary condition at low Reynolds numbers ranging from 500 to 4700 and at modified Grashof numbers ranging from 0.8×10^7 to 3.3×10^7. The effect of varying the free stream turbulence intensity from about 0.5 to 20% has also been studied.

NOMENCLATURE

D, diameter of the test cylinder;
Gr, Grashof number, $g\beta(T_{s_{av}} - T_a)D^3/\nu^2$;
$\overline{Gr}$, modified Grashof number, $g\beta(\Sigma Q_c)D^4/2k\nu^2\pi DL$;
h, local heat-transfer coefficient;
h_{av}, average heat-transfer coefficient;
k, thermal conductivity of air;
L, length of the test cylinder;
Nu, local Nusselt number, hD/k;
Nu_{av}, average Nusselt number, $h_{av}D/k$;
q_w, average convective heat flux, $(\Sigma Q_c/\pi DL)$;
Q_c, amount of heat lost by convection from each local element;
Re, Reynolds number, $\rho UD/\mu$;
T_a, ambient temperature;
T_s, local surface temperature;
$T_{s_{av}}$, average surface temperature;
Tu, turbulence intensity, u_{RMS}/U;
U, average velocity of flow;
u_{RMS}, RMS value of longitudinal velocity fluctuations.

Greek symbols

β, coefficient of volumetric expansion;
μ, dynamic viscosity of air;
ν, kinematic viscosity;
ρ, density of air.

List of abbreviations

fsp, forward stagnation point;
rsp, rear stagnation point.

INTRODUCTION

LOCAL heat transfer from a circular cylinder in cross flow in the range of Reynolds numbers from 500 to 5000 has not been adequately studied. It depends among other factors on the boundary condition at the surface, the amount of free convection present and the turbulence level in the external stream. Only limited data exist for the local heat-transfer coefficients in pure forced convection under the constant heat flux condition in the low Reynolds number range [1]. The influence of body forces has been studied to some extent analytically and experimentally. However these studies have been confined to the parallel and counter flow situations [2, 3] and in the case of a cross flow situation, only measurements on the average heat-transfer coefficient are available [4, 5]. Further, although there exists a vast amount of literature on the influence of free stream turbulence in forced convection, these studies generally pertain to the high Reynolds number range [6, 7]. There is comparatively little data available on the influence of free stream turbulence on heat transfer in the low Reynolds number range [8].

The purpose of this investigation, which has been conducted in the low Reynolds number range, is: (i) to provide local measurements of the heat-transfer coefficient in forced convection for the uniform heat flux boundary condition; (ii) to study the influence of free convection on forced convection heat transfer when the forced flow is perpendicular to that due to buoyancy, and (iii) to examine the effects of free stream turbulence on the local and average heat-transfer coefficient in forced convection and mixed convection.

EXPERIMENTAL TECHNIQUE

A test cylinder with which it is possible to measure the local heat-transfer coefficient over the entire circumference under the constant heat flux condition has been fabricated. The cylinder (Fig. 1) consists of thirty identical brass pieces (each with its own thermocouple and heating arrangement) placed in grooves made in a "fiber" rod and turned to form a rigid, smooth and almost continuous brass surface 7.62 cm in diameter and 15.1 cm long. A central hole in the "fiber" rod provides a path for bringing the heater leads and thermocouple wires out. The brass surface is polished, thereby minimizing the heat loss due to radiation. The "fiber" material separating any two brass pieces minimizes the circumferential heat conduction from one

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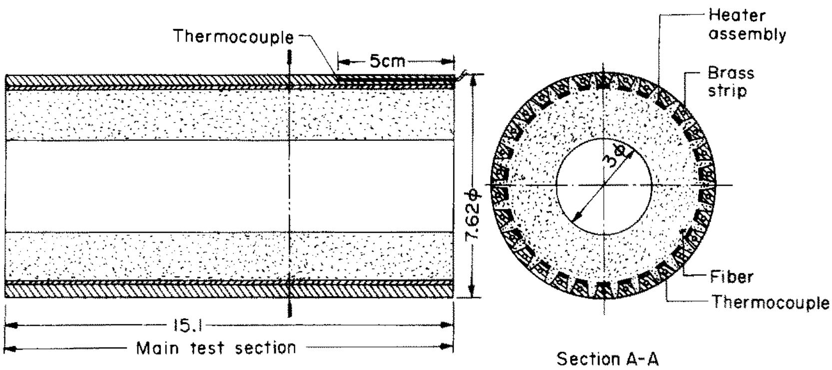

FIG. 1. Details of the test cylinder (all dimensions in cm).

piece to another which occurs as a result of the circumferential temperature variation.

Two guard heater assemblies (whose heat inputs can be adjusted circumferentially) are placed on either side of the main test portion to minimize the axial heat conduction. These guard heater assemblies are similar in construction to the main test portion but each contains only six elements. For further details reference may be made to Sarma [9].

The cylinder is mounted in a low speed wind tunnel with a test section 60 × 75 cm (giving a blockage of 10%), in which uniform and constant velocities and reasonably low values of turbulence intensity (<0.5%) can be obtained. Different grids placed upstream of the test cylinder are used for obtaining higher values of turbulence intensity up to about 20%. The mean velocity and turbulence intensity measurements are carried out with a DISA hot wire anemometer system. Following Kestin and Wood [10], the turbulence intensity is measured two diameters upstream of the test cylinder. The length scale of turbulence is not measured. It is, however, estimated to be about the size of the cylinder and is essentially constant throughout the investigation.

In each test, a steady state temperature distribution over the entire circumference of the cylinder is obtained for a given value of heat flux, average velocity and turbulence intensity. The same heat flux is supplied to each of the thirty elements. In the steady state, it is ensured that the heat lost in the axial direction is negligible by individually adjusting the heat inputs to the six elements in each guard heater. The heat lost by convection is computed by subtracting the components due to radiation and the net circumferential heat conduction from the total heat input to each element. The calculation of the radiation and heat conduction components requires a knowledge of the emissivity of the test surface and the thermal conductivity of the "fiber" material. These values are determined by conducting some preliminary calibration tests. Knowing the heat lost by convection (Q_c) and the local surface temperature excess over the ambient temperature ($T_s - T_a$), the local heat-transfer coefficient (h) is determined. Because of the above mentioned corrections, the constant convective heat flux boundary condition is only realized to within about ±10%. The average heat-transfer coefficient (h_{av}) is obtained by taking the average of the thirty values of the local heat-transfer coefficient. Finally the non-dimensional numbers Re, $\overline{Gr}$ and Nu_{av} are determined by evaluating all properties at the mean film temperature, which is the arithmetic mean of the average surface temperature and the ambient temperature.

The accuracies of determination of the various quantities are as follows:

Power input to a local element ±1%
Velocity of flow ±5%
Temperature ±0.2°C
Turbulence intensity ±10%
Reynolds number ±5%
Local Nusselt number ±8%.

It may be noted that the uncertainty of ±8% in the local Nusselt number determination is mainly due to the uncertainties in the determination of the radiation and circumferential heat-conduction correction. The radiation correction usually amounts to about 25% of the total heat input and is estimated to within ±7%. It could contribute an error of ±3% in the determination of the local Nusselt number. The corresponding numbers for the circumferential heat conduction are 15%, ±15% and ±4%.

RESULTS AND DISCUSSION

During the tests, the heat flux has been varied from 140 to 500 kcal/h m^2, the average velocity from 10 to 100 cm/s, and the turbulence intensity from 0.5 to 20%. As a result, the value of ($T_s - T_a$) has varied from about 10 to 50°C and the independent dimensionless parameters have varied as follows:

Reynolds number (Re): 500–4700
Modified Grashof number ($\overline{Gr}$): 0.8 × 10^7–3.3 × 10^7
Mixed convection parameter ($\overline{Gr}/Re^{2.5}$): 0.02–3.2.

It will be noted that a modified Grashof number ($\overline{Gr} = g\beta q_w D^4/2kv^2$) is being used in preference to the usual Grashof number. This is because a boundary-layer analysis for the case of free convection under the constant heat flux condition [11] indicates that the modified Grashof number is the governing parameter. Further, a boundary-layer analysis for the case of mixed convection with the constant heat flux condition [2] indicates that $\overline{Gr}/Re^{2.5}$ is the governing parameter instead of the usual parameter Gr/Re^2 and that buoyancy effects start to become important when the value of this parameter is around unity. Unfortunately because of experimental limitations on the maximum permissible surface temperature ($\sim 90°C$), it has not been possible to work with sufficiently large values of $\overline{Gr}$ or to vary the value of $\overline{Gr}$ much. It has varied only by a factor of four from 0.8×10^7 to 3.3×10^7.* Hence only a limited amount of data has been obtained with values of $\overline{Gr}/Re^{2.5}$ greater than unity. Also the large variation obtained in the value of $\overline{Gr}/Re^{2.5}$ is essentially due to the variation of the Reynolds number.

The experimental data are considered in two parts. First the data in forced convection and mixed convection at very low values of turbulence intensity are discussed. This is called the basic data. The influence of free stream turbulence on this basic data is then presented.

Basic data

Figure 2 shows typical distributions of the local Nusselt number in (i) forced convection ($Re = 3480$, $\overline{Gr} = 2.62 \times 10^7$, $\overline{Gr}/Re^{2.5} = 0.037$), and (ii) mixed convection ($Re = 506$, $\overline{Gr} = 1.82 \times 10^7$, $\overline{Gr}/Re^{2.5} = 3.17$). In forced convection, the agreement between the

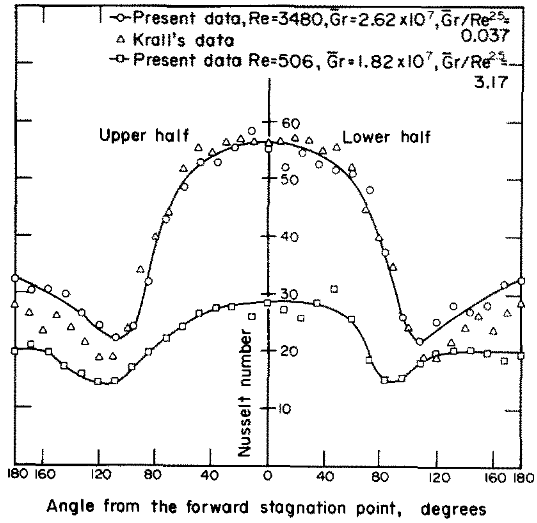

FIG. 2. Typical distribution of local Nusselt number in (i) forced convection, $Re = 3480$ and (ii) mixed convection $Re = 506$.

function of the Reynolds number. On the same figure, Krall's data (for the same range) along with the theoretical prediction due to Squire [12] for the forward stagnation point is also reproduced. It can be seen

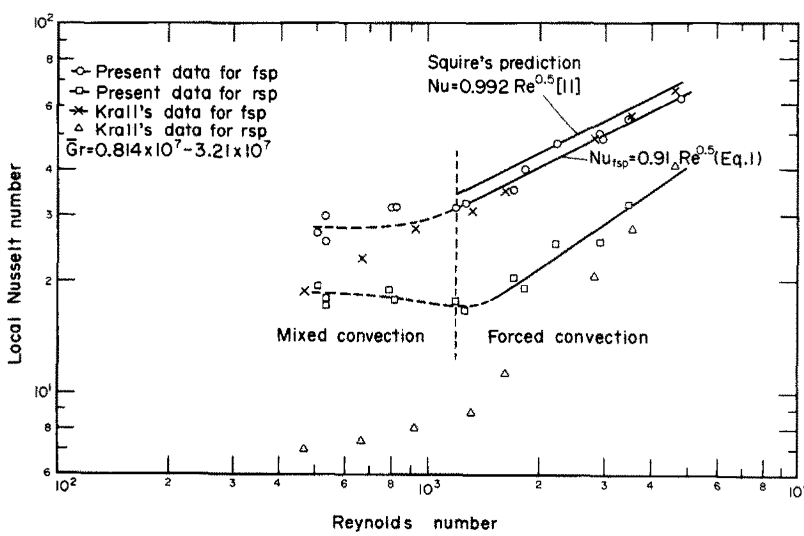

FIG. 3. Variation of forward and rear stagnation point values of Nusselt number with Reynolds number.

present data and those due to Krall [1] for $Re = 3570$ is good. The point of separation is symmetrical about the forward stagnation point and occurs at about 110°. However at $Re = 506$ where heat transfer takes place by mixed convection, the distribution is quite asymmetrical resulting in two different points of separation, 120° on the upper half and 86° on the lower half.

Figure 3 shows the variation of the local Nusselt number at the forward and rear stagnation points as a

*It is of interest to note that the values of the conventional Grashof number (calculated on the basis of the excess of the average surface temperature over the ambient temperature) are approximately one order of magnitude less and have ranged from 0.7×10^6 to 1.8×10^6.

that there is good agreement for values of $Re > 1200$ ($\overline{Gr}/Re^{2.5} \simeq 0.6$) where free convection effects are negligible. The data can be represented by the equation

$$Nu_{fsp} = 0.91(Re)^{0.5}. \qquad (1)$$

The exponent 0.5 indicates the prevalence of a laminar boundary layer. As the Reynolds number is reduced below 1200, the stagnation point value approaches a constant value, thus indicating the predominant influence of free convection. The amount of increase of the local Nusselt number at the forward stagnation point due to free convection effects is about 50% at $Re = 500$.

For the rear stagnation point, the present data show a variation of the Nusselt number similar to that at the forward stagnation point. However, in the higher range of Reynolds numbers where free convection effects are negligible the slope of the graph is more than the value of 0.5. The Nusselt number is more than doubled at $Re = 500$ due to the presence of free convection effects and it is seen that the local influence of free convection persists up to a Reynolds number of about 2000. A comparison with the data of Krall indicates lack of good agreement for which no adequate explanation can be given.

The average Nusselt number for the entire range of Reynolds numbers studied is shown in Fig. 4. The average values agree well with those obtained by Krall for Reynolds numbers greater than 1200, where free convection effects are negligible. At lower values of the Reynolds number, the present data start deviating systematically from the line representing forced convection and become independent of the Reynolds number. On the same figure, similar data from reference [4] obtained for two Grashof numbers (based on the average surface temperature) of the order of 10^3 and 10^5 are also shown. The average Nusselt number in the forced convection regime ($1200 < Re < 4700$) can be correlated by the following equation using the method of least squares.

$$Nu_{av} = 0.62 Re^{0.505}. \tag{2}$$

The lower line in Fig. 4 represents the well known correlation due to Hilpert [13], which differs from the present data for the constant heat flux case by about 35%. This is to be expected since Hilpert's data is for the constant temperature case.

Influence of turbulence intensity on basic data

Figure 5 shows typical distributions of the local Nusselt number over the circumference with turbulence intensity as parameter in forced convection at $Re \simeq 4700$. In general, the heat-transfer coefficient over the entire circumference is quite sensitive to change in turbulence intensity, the maximum changes occurring in the forward region up to about 60° from the forward stagnation point and in the wake region. There is a definite indication that the value of the heat-transfer coefficient goes through a maximum at an angle of about 50° from the fsp. The points of separation are also affected but occur symmetrically about the forward stagnation point.

Figure 6 shows the effect of turbulence intensity on the distribution of the local Nusselt number at $Re \simeq 500$, $\overline{Gr} \simeq 10^7$ and $\overline{Gr}/Re^{2.5} \simeq 3$, which is in the combined convection range. It is seen that the turbulence intensity does not influence the values of the local

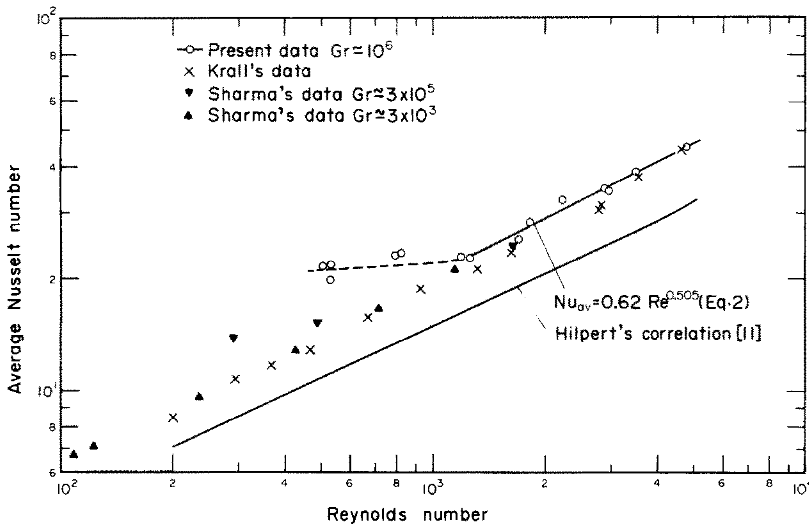

FIG. 4. Variation of average Nusselt number with Reynolds number.

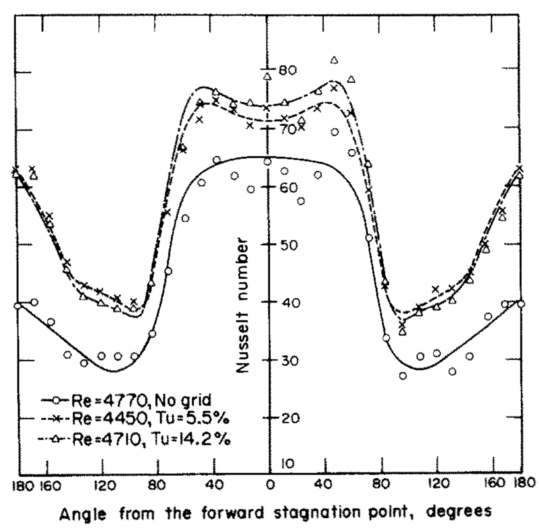

FIG. 5. Influence of turbulence intensity on local Nusselt number in forced convection ($Re \simeq 4700$).

Nusselt number much excepting for the lower part of the forward region. In this region, the local Nusselt number in fact decreases with increasing intensity. It is to be noted however that since the velocity is very low in this case, the Reynolds number based on the grid element size as the characteristic dimension is also low. Hence the turbulence spectrum is dominated by the shedding frequencies. This is especially true for the

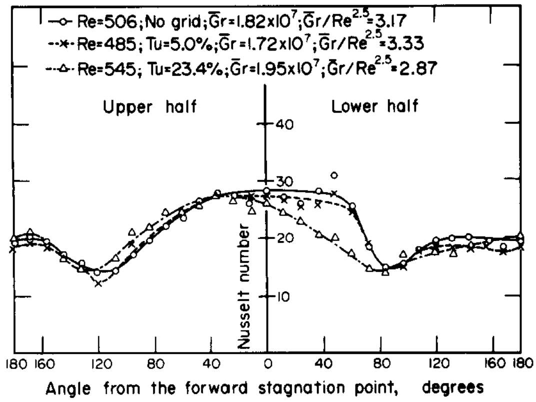

FIG. 6. Influence of turbulence intensity on local Nusselt number in mixed convection ($Re \simeq 500$).

lower values of turbulence intensity up to 5% where the ratio of the grid mesh size to the size of the grid element is rather large.

The influence of varying the modified Grashof number on the distribution of local Nusselt number while the Reynolds number and the turbulence intensity are kept constant is shown in Fig. 7. It can be seen that increased free convection results in a decrease in the Nusselt number in the forward region on each half of the cylinder, while the wake remains more or less unaffected.

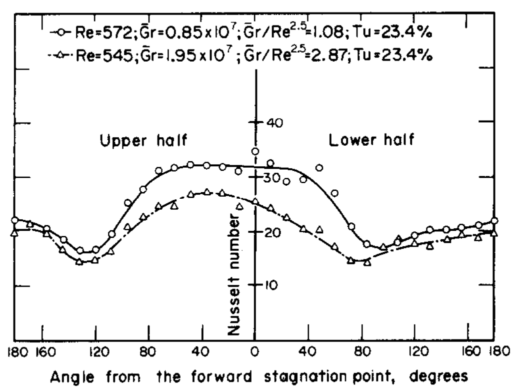

FIG. 7. Influence of increased Grashof number on local Nusselt number for the same Re and Tu ($Re \simeq 500$).

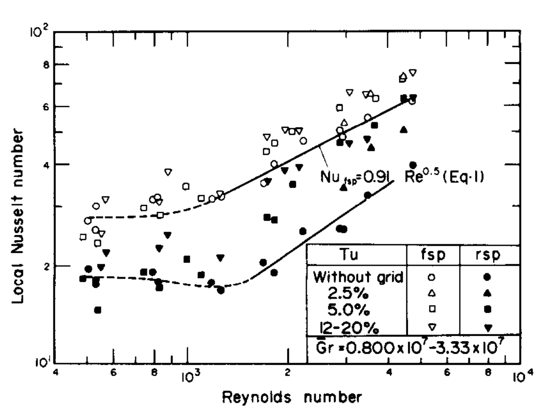

FIG. 8. Variation of local Nusselt number at fsp and rsp as a function of Reynolds number with turbulence intensity as parameter.

Figure 8 indicates the influence of turbulence intensity on the local Nusselt number at the forward and rear stagnation points for different Reynolds numbers. In the forced convection region, the forward stagnation point value increases by about 20% for an increase of 5% in the turbulence intensity. On the other hand, the rear stagnation point value increases by about 50%. However any further increase in the turbulence intensity does not have any significant effect. Below Reynolds numbers of 1200, the local Nusselt number is also influenced by buoyancy forces, resulting in an occasional decrease of the heat-transfer coefficient.

Figure 9 shows the variation of the average Nusselt number with Reynolds number at different values of the turbulence intensity. It is seen that in the forced

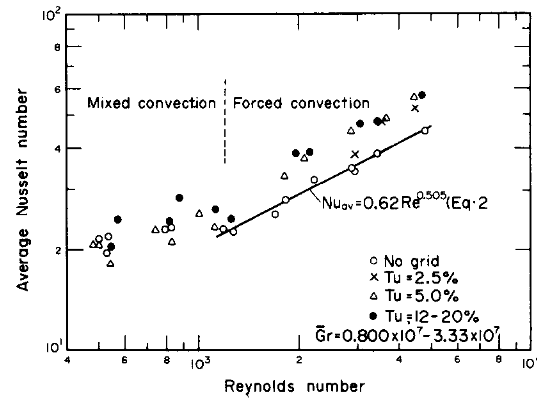

FIG. 9. Variation of average Nusselt number as a function of Reynolds number with turbulence intensity as parameter.

convection region at values of Reynolds numbers greater than 1200, a value of turbulence intensity of about 5% results in an increase in the average Nusselt number by about 25%. However, no significant improvement takes place with still higher values of turbulence intensity. Below this value of the Reynolds number, the average Nusselt number is also influenced by buoyancy effects and the behaviour is similar to that at the forward stagnation point.

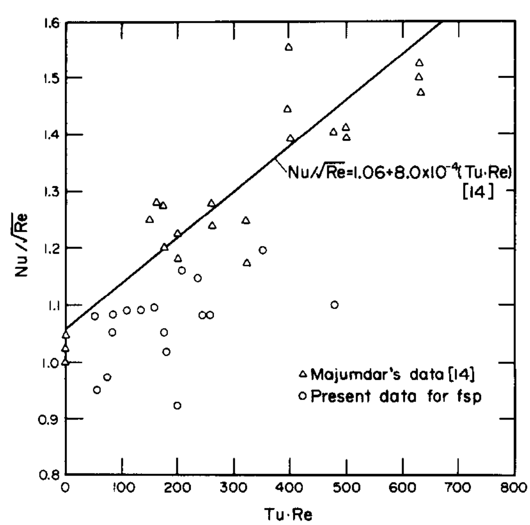

FIG. 10. Influence of turbulence intensity on the forward stagnation point value.

Following earlier investigators [6], an attempt is made to study the influence of free stream turbulence on the forward stagnation point value in forced convection ($1200 < Re < 4700$) by plotting $Nu/\sqrt{(Re)}$ against the parameter $Tu \cdot Re$ (Fig. 10). The data points and correlation obtained by Majumdar [14] in the range $5000 < Re < 11\,000$ are also shown for comparison in the same figure. This is the only investigation somewhat near to the present one. It is seen however that Majumdar's results do not agree well with those of the present investigation. This shows clearly that it is probably not adequate to use $Tu \cdot Re$ as a single similarity parameter to generalize data on the influence of free stream turbulence in this range of Reynolds numbers.

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TRANSFERT THERMIQUE LOCAL POUR UN CYLINDRE HORIZONTAL ATTAQUE TRANSVERSALEMENT PAR L'AIR: INFLUENCE DE LA CONVECTION NATURELLE ET DE LA TURBULENCE INCIDENTE

Résumé—Des mesures de transfert thermique ont été effectuées pour un cylindre circulaire horizontal et l'air en attaque transversale tant pour la convection forcée que pour la convection mixte. Les études concernent la condition de densité de flux constante à des nombres de Reynolds variant de 500 à 4700 et des nombres de Grashof modifiés allant de $0{,}8 \times 10^7$ à $3{,}3 \times 10^7$. On a considéré aussi l'effet de la variation de l'intensité de turbulence incidente entre 0,5 et 20 pour cent.

ÖRTLICHER WÄRMEÜBERGANG AN EINEM MIT LUFT QUERANGESTRÖMTEN, HORIZONTALEN KREISZYLINDER: EINFLUSS DER FREIEN KONVEKTION UND DER FREISTRAHLTURBULENZ

Zusammenfassung—Es wurden örtliche Messungen des Wärmeübergangs an einem mit Luft querangeströmten, horizontalen Kreiszylinder bei erzwungener und gemischter Konvektion ausgeführt. Die Untersuchungen wurden bei konstantem Wärmestrom, bei niedrigen Reynolds-Zahlen von 500 bis 4700 und bei modifizierten Grashof-Zahlen von $0{,}8 \cdot 10^7$ bis $3{,}3 \cdot 10^7$ durchgeführt. Der Einfluß der Freistrahlturbulenz wurde bei Intensitäten von 0,5 bis 20% ebenfalls untersucht.

ЛОКАЛЬНЫЙ ТЕПЛООБМЕН ОТ ГОРИЗОНТАЛЬНОГО ЦИЛИНДРА В ВОЗДУХ В ПОПЕРЕЧНОМ ПОТОКЕ; ВЛИЯНИЕ СВОБОДНОЙ КОНВЕКЦИИ И ТУРБУЛЕНТНОСТИ СВОБОДНОГО ПОТОКА

Аннотация — Выполнены локальные измерения теплообмена от горизонтального круглого цилиндра в воздух в поперечном потоке при вынужденной и смешанной конвекции. Исследования проведены при постоянном тепловом потоке на границе с низкими числами Рейнольдса, изменяющимися от 500 до 4700, и модифицированными числами Грасгофа в области от $0{,}8 \times 10^7$ до $3{,}3 \times 10^7$. Рассматривается также влияние изменения интенсивности турбулентности свободного потока приблизительно от 0,5 до 20%.