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AN EXPERIMENTAL STUDY OF THE SENSITIVITY TO FREESTREAM TURBULENCE OF HEAT TRANSFER IN WAKES OF CYLINDERS IN CROSSFLOW

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Abstract—An experimental study of the effect of mainstream turbulence on heat transfer from a cylinder is presented. The cylinder is electrically heated and the effect of turbulence on the pressure distribution and local heat-transfer coefficient round the cylinder is reported; the latter data are compared with other published information.

Measurements of the mean velocity and r.m.s. velocity fluctuations in the region of $\pm$ 40° from the rear of the cylinder were taken using a single wire anemometer and the reduced velocity profiles fitted to simple power relationships. The frequency spectrum at the rear of the cylinder is shown to be dominated by twice the vortex shedding frequency.

The heat-transfer data, collected at the rear of the cylinder, are correlated in terms of a turbulent energy Reynolds number and this turbulent energy is related to the mainstream turbulence level and the mainstream Reynolds number to produce an empirical correlation connecting the heat transfer at the rear of the cylinder to the mainstream Reynolds number and turbulence level. A simple one dimensional theoretical interpretation of these results is given. The results show that in the Reynolds number range 5000–35000 increases in freestream turbulence intensity of 10 per cent result in improvements of heat transfer at the rear of the cylinder of up to 100 per cent.

NOMENCLATURE

A, n, constants in hot wire equation;
B, p, constants in equation (13);
C_p, specific heat at constant pressure;
D, cylinder diameter;
E, instantaneous total voltage across wire;
E_0, voltage at zero flow;
ΔE, $E - E_0$;
e', instantaneous voltage;
$\sqrt{\overline{e'^2}}$, r.m.s. voltage;
$\bar{E}_b$, mean voltage;
h, heat-transfer coefficient;
k, $\overline{u'^2} + \overline{v'^2} + \overline{w'^2} \sim \overline{u'^2}$, here;
L, a length scale of turbulence;
m, n, indices in Figs. 11 and 12;
Nu, Nusselt number;
Nu^*, ratio of heat transfer with freestream turbulence to heat transfer without freestream turbulence;
Pr, Prandtl number;
q_w, heat flux;
Re_∞, $\frac{\bar{u}_\infty D}{\nu}$ freestream Reynolds number;
ΔT, $T_w - T_\infty$;
Tu, $\frac{\sqrt{\overline{u'^2}}}{\bar{u}}$ turbulence intensity;
$\bar{u}$, mean velocity;
u', instantaneous velocity fluctuation;
$\sqrt{\overline{u'^2}}$, r.m.s. velocity fluctuation;

u_G^*, $\dfrac{\text{r.m.s. velocity fluctuations at rear stag. pt. with freestream turbulence}}{\text{r.m.s. velocity fluctuations at rear stag. pt. with no freestream turbulence}}$

Greek

α, θ, degrees of arc;
ρ, density;
ν, kinematic viscosity;
τ, shear stress.

Subscripts

$_G$, edge of boundary layer;
$_\infty$, main stream.

1. INTRODUCTION

THIS experimental investigation is a first step at understanding the interaction between freestream turbulence and the wake motion behind a cylinder in crossflow. The relative ease with which heat-transfer measurements can be made provides a useful means of demonstrating the sensitivity of heat transfer, and thus the wake motion, to freestream turbulence. Little information is available however, concerning the actual fluid motion near the wall and a main part of this investigation is a study of this region using a hot wire anemometer. Although this analysis is restricted to using a single wire probe to examine an extremely complex flow, useful information is obtained concerning the local flow near the wall and the likely interaction between wake and freestream motion. The flow past bluff bodies is of considerable practical interest but the prediction of heat-transfer at the rear of such bodies has not yet been adequately achieved. This is largely due to the lack of understanding of the fluid motion. Even for the case of a single cylinder experimental data are conflicting for a single Reynolds number.

There is an abundance of experimental data available in the literature [16] and only research directly relevant to this work will be reviewed. A comprehensive study on the effects of freestream turbulence, both intensity and scale, on overall heat-transfer from cylinders in crossflow has been completed by Zijnen [1]. Data are available for a range of cylinder diameters in the Reynolds number range up to 25000. Two functions Φ and Ψ were used to relate the overall heat transfer to the freestream motion by the empirical expression

$$Nu^* = 1 + \Phi(Re\,Tu) \times \Psi\left(\frac{L}{D}\right) \tag{1}$$

where Tu is the freestream turbulence intensity, L/D the ratio of the length scale of turbulence to cylinder diameter and Nu^* the ratio of heat-transfer with turbulence to that with no freestream turbulence. The Ψ function exhibits a maximum which is dependent on the length of turbulence. This was attributed to an interaction between the vortex shedding motion and the freestream but no measurements were made to determine either where the increase in heat transfer occurred on the cylinder or to study the fluid motion in the wake. Some experimental data are available for heat transfer in the separated flow region and are reported by Richardson [2] and Sogin [3]. It appears that the overall heat transfer is dependent upon $Re^{\frac{2}{3}}$ but no account has been taken of freestream turbulence effects. Dyban and Epick [4] have measured local heat transfer at different levels of freestream turbulence, and Hanson and Richardson [5] have investigated the near wake region using a single wire probe but the data are not in a form which can be easily interpreted.

In recent years several studies have been completed to measure the effects of sound and vibration on heat transfer. Although large increases are most likely to occur in natural convection it has been demonstrated that for forced convection across a cylinder there appears to be an interaction between the wake motion and the induced oscillations caused either by vibration of the cylinder or by the introduction of a sound field in the vicinity of the wake. Fand

and Cheng [6] introduced a sound field normal to the airstream and obtained improvements in heat transfer of order of 25 per cent, but sound pressure levels greater than 135 dB, which is beyond the threshold of pain, where required. They observed that for one applied sound frequency a maximum in heat transfer occurred over a range of Reynolds numbers. For example, with an applied frequency of 1100 Hz, a maximum occurs at a Reynolds number of 9000 which is claimed to be caused by some kind of resonant interaction. Clearly this is not one between the vortex shedding frequency and the freestream as at this Reynolds number the vortex shedding frequency is approximately one tenth of that of the applied sound field and there is therefore no apparent connection. Bloor [7] however, who has completed a detailed study of wakes has detected other frequencies which are a function of the vortex shedding frequency. For this Reynolds number, the "transition frequency" as it is called, coincides with that of the applied sound field. Kezios [8] has shown that by vibrating a cylinder normal to an airflow, increases in overall heat transfer of 20 per cent can be obtained. It was observed that a maximum in heat transfer occurred when the frequency of vibration coincided with the vortex shedding frequency.

It therefore appears then that results are available for the effect of turbulence, noise and vibration on the overall heat transfer from cylinders, but the detailed study of the effect of turbulence on the rear of the cylinder, the region of most interest, is limited. A lack of knowledge about the fluid motion, particularly the interaction between freestream and wake oscillations, is a barrier to accurate prediction of heat transfer in the rear of cylinders. This is the main purpose of this paper.

2. EXPERIMENTAL APPARATUS AND INSTRUMENTATION

2.1 *Heat transfer measurements*

The experimental determination of local heat transfer coefficients requires that measurements be made of the local heat flux and surface temperature. Many devices which have been used to effect these measurements on the outer surface of a cylinder have employed electrical heating on account of the ease with which heat flux may be measured and controlled. For the present problem a thin metallic film located at the outer surface of a cylinder was heated electrically thus producing conditions of constant heat flux; the variations in surface heat transfer thus appear as variations in surface temperature. Basically the instrument comprised of a sheet of stainless steel foil, of mean thickness 0·05 mm, rolled into a cylinder 19·1 mm dia., two cylindrical copper conductors and thirteen surface mounted thermocouples, the interior of the foil cylinder being filled with an epoxy resin. The construction of the cylinder is illustrated in Fig. 1.

Under steady state conditions the directions of heat flux is essentially outwards from the probe; temperature gradients in the region of the inside surface can be expected to be weak. Errors due to non-uniform heat generation resulting from variations in thickness will also be of this order. The temperature drop across the foil was calculated from the equation given by McAdam [9] and errors due to circumferential temperature gradients were estimated using the method outlined by Giedt [10].

The probe was held rigidly in the horizontal plane on the centre line of the duct and normal to the airstream. The electrical heating was supplied from an a.c. source and the power dissipated was obtained by measuring the voltage drop across the foil and the current passing through it.

2.2 *Velocity and pressure measurements*

All measurements were made in an open section wind tunnel which had a freestream turbulence intensity of less than 1 per cent for Reynolds Numbers between 5000 and 35000. The turbulence intensity was increased by placing grids manufactured from horizontal bars at a fixed distance upstream of the test

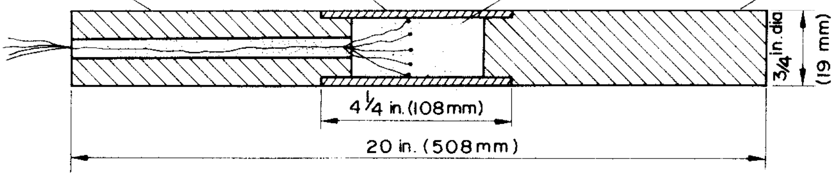

FIG. 1. The constant heat flux cylinder.

cylinder. Using bar diameters between 3·18 mm and 19·1 mm, the intensity could be varied up to 10 per cent. A general layout of the test section in the wind tunnel is shown in Fig. 2. In addition two square pitched grids were used, but no difference on the heat transfer was detected between these and the corresponding horizontal bar system.

The mean velocity in the freestream was measured by a Pitot-Static tube which was used in conjunction with a hot wire anemometer to obtain the freestream turbulence intensity. The wall static pressure was also measured by having a pressure tapping on the surface of a test cylinder which was rotated to record the wall pressure at 15° intervals around the cylinder.

The primary task in this investigation was to relate the heat transfer to the main stream turbulence through the wake motion. Since most of the heat transfer occurs in the region very close to the wall measurements were restricted to this area. It was assumed that the axis of vortices being shed from the cylinder were parallel to the cylinder. While this may not be absolutely correct (see Hanson and Richardson [5]) it is an adequate simplification. A commercially available constant temperature anemometer was used for this work in conjunction with a single wire sensing element. It is assumed here that the direction of the fluctuating component is parallel to the mean flow. This must introduce some error, the magnitude of which will remain unresolved until hot wire techniques in this type of flow are perfected. However, it should be noted that the detailed wake measurements obtained here are largely used to interpret the heat transfer data and it is considered that the single wire anemometer is adequate for this purpose.

No explicit corrections were made for aspect ratio, end losses and other non-linear effects of the probe; instead these effects were accounted for by means of a calibration for each wire used. This was done for two ranges of mean velocity; 0·305–3·65 m/s, for the region near the cylinder wall and 3·05–27·4 m/s for the freestream

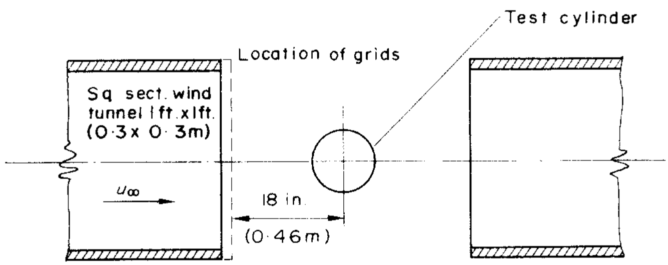

FIG. 2. Test section.

measurements. The calibration equation was written in the form

$$\bar{E}_b^2 = E_0^2 + A\bar{u}^n \tag{2}$$

where the index n was approximately 0·7 for the former case and 0·5 for the latter. This is in agreement with the work done by Turner [11]. When the mean velocity is large and the ratio $(\sqrt{\overline{u'^2}})/\bar{u} \ll 1$ the turbulence intensity may be calculated from

$$Tu = \frac{\sqrt{\overline{u'^2}}}{\bar{u}} = \frac{2\bar{E}_b^2}{n(\bar{E}_b^2 - E_0^2)} \frac{\sqrt{\overline{e'^2}}}{\bar{E}_b}. \tag{3}$$

When the mean velocities are low and the fluctuating component is of the same order as the mean velocity these equations are invalid. The method adopted by Sadeh [12] allows for this and leads to the equations

$$\bar{u} = \frac{2E_0^{1/n}}{A} (\Delta E)^{1/n} \tag{4}$$

where

$$\Delta E = \bar{E}_b - E_0 .$$

3. EXPERIMENTAL RESULTS

3.1 *Freestream turbulence*

The freestream turbulence was measured in the plane passing through the axis of the cylinder and normal to the flow with the cylinder removed from the test section. The variation of turbulence intensity across the duct was constant for each grid used for the tests. The data are plotted in a non-dimensional form in Fig. 3. The range of freestream turbulence intensity did not

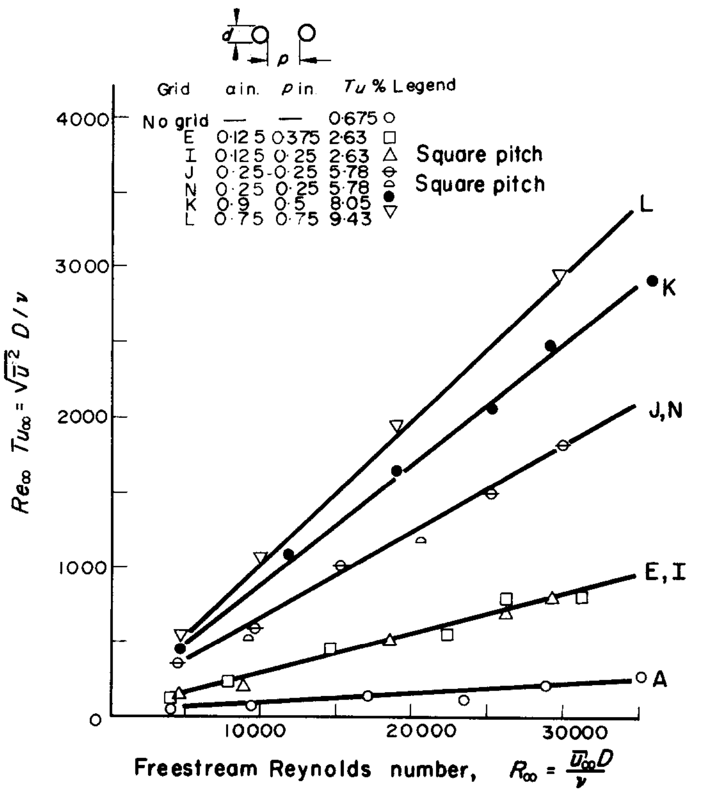

FIG. 3. Variation of freestream turbulence with grid size.

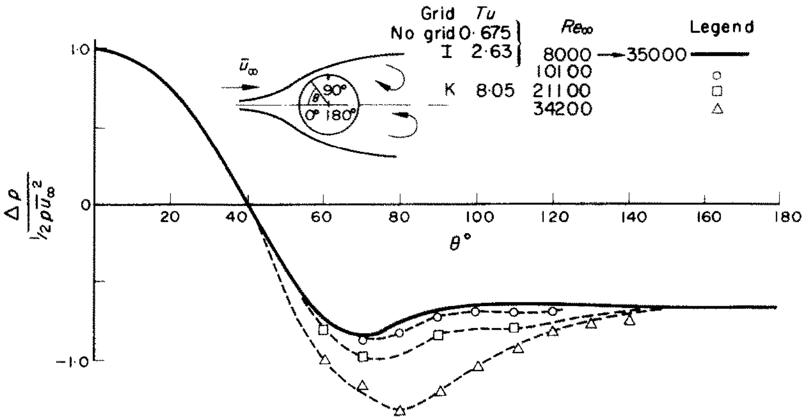

FIG. 4. Variation of static pressure coefficient around cylinder.

exceed 10 per cent and therefore the assumptions implied in derivation of equation (3) are valid. No attempt was made to measure the scale of turbulence. It was observed, however, that the turbulence spectrum did not change significantly with the grid used, implying that at the cylinder position, although the intensity changed markedly with grid size, the scale remained sensibly constant.

3.2 *Wall static pressure measurements*

The variation of pressure coefficient $\Delta p/\frac{1}{2}\rho\bar{u}_\infty^2$ with angular position θ in the cylinder is shown in Fig. 4. Clearly the freestream turbulence is only effective in the region from 40° to 140° although even in this region for turbulence levels below 2·63 per cent there is no measurable effect. The collapse of all data onto the solid line from 0° to 40° and from 140° to 180° is not surprising, since the wall static pressure is a function of mean velocity and for the region where there is known to be a steady mean flow, fluctuations are likely to remain undetected with the type of measuring equipment used. In the region from 40° to 140° where the main flow is now remote from the cylinder some kind of mean value will result from the fluctuating motion. It has been shown by Gerrard [13] that this region is sensitive to fluctuations; therefore the variation in mean pressure with Reynolds number is not surprising. In the wake region the wall pressure gradient is of order zero for about 40° either side of the rear stagnation point implying a constant mean velocity in this region which will be shown later to be in agreement with the hot wire data.

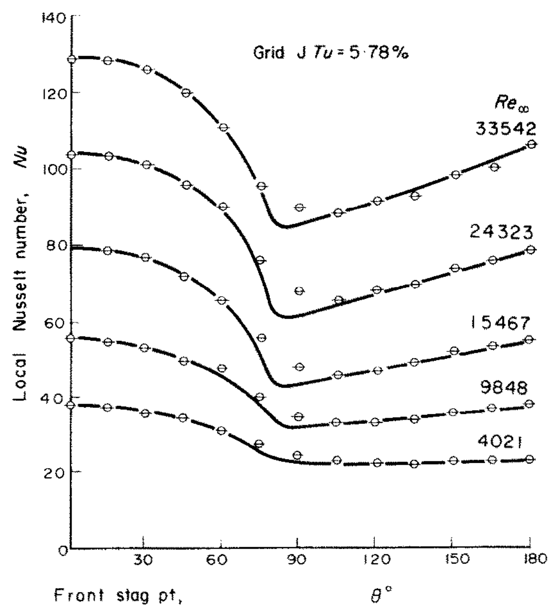

FIG. 5. Effect of free stream turbulence on local heat transfer.

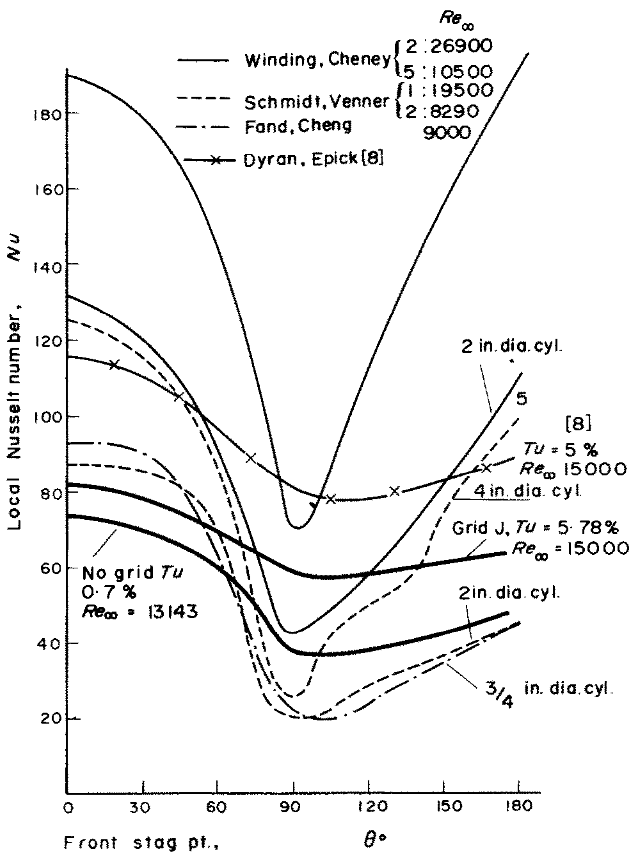

FIG. 6. Distribution of local heat transfer: experimental data of other workers compared with current data.

3.3 *Local heat transfer*

The maximum temperature of the heated cylinder was not allowed to exceed 70°C, because above this value the epoxy resin became pliable and the thermocouples tended to detach themselves from the foil. To keep the errors as small as possible, in the region of 2 per cent, the minimum temperature difference was always greater than 8°C. These two restrictions required the heat flux to be within 630–1260 W/m^2. No variation in heat transfer coefficient was detected as a result of this. The local heat transfer distribution is plotted non-dimensionally as Nusselt number vs. θ. Typical data are shown in Fig. 5.

Although the general trend is similar to that of other workers it is clearly seen in Fig. 6 that the variation in heat transfer is not as great as reported elsewhere. A direct comparison is difficult, since seldom is the turbulence intensity quoted. In addition, the cylinder diameter is likely to have an effect via the turbulence scale, since this has been demonstrated by Zijnen [1] to be of importance. Apart from the vertical shift of the curves, which could be attributed to turbulence intensity, the greatest discrepancy appears to be the shape of the distribution. Since the present data agree in shape with pressure and velocity measurements in the same region (see Figs. 4 and 10), a small gradient appears

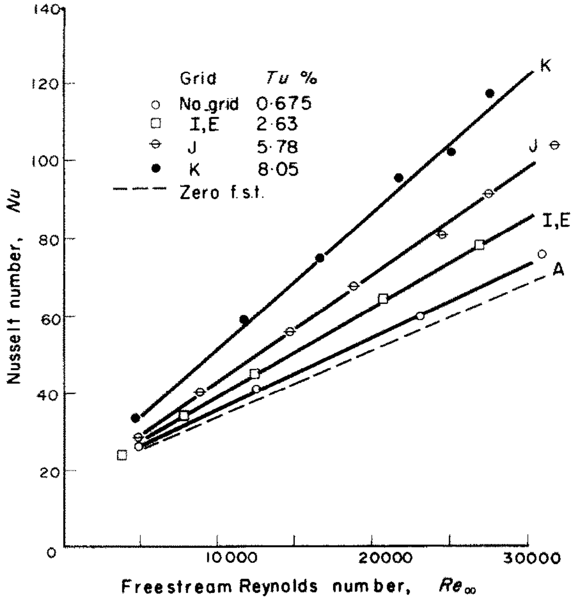

FIG. 7. Variation of heat transfer at rear stagnation point with freestream turbulence.

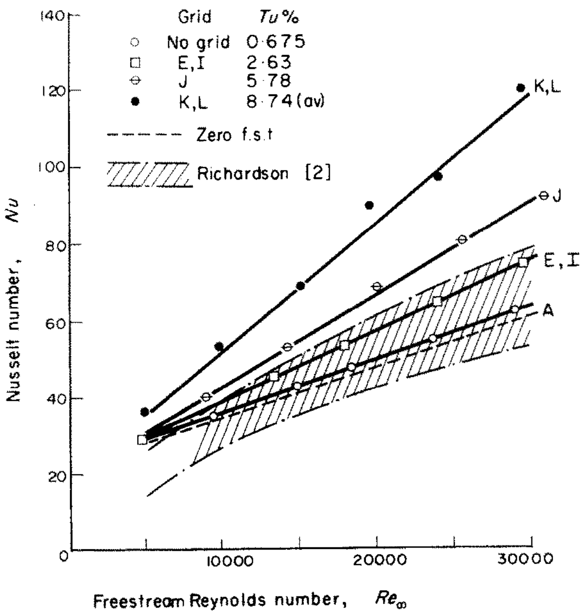

FIG. 8. Variation of overall heat transfer in the wake region.

the most satisfactory. Recently, similar trends in the wake region have been observed by Dyban and Epick [4] as shown in Fig. 6. A comparison of heat transfer at the rear stagnation point with the Fand and Cheng data at low turbulence intensities produces results of the same order of magnitude.

The data clearly demonstrate the sensitivity of heat transfer to freestream turbulence both on the front portion of the cylinder and in the wake region. Only the latter is of interest in this paper. It has already been stated elsewhere that a different mechanism is responsible for improvements in heat transfer on the front portion of the cylinder (see [14]).

The effect of freestream turbulence on heat transfer at the rear stagnation point is shown in Fig. 7 where improvements up to to 80 per cent were obtained when the turbulence increased from 0·675 per cent to 8·05 per cent. To provide a basis for comparison the Nusselt number at zero freestream turbulence was obtained by extrapolation of the data and is indicated by the chain line in Fig. 7. For the purposes of comparison with other reported data, the overall heat transfer in the region from 90° to 180°, that is the heat transfer beyond the separation point, is plotted in Fig. 8. Good agreement is obtained with the data of Richardson [2]. Although no effect of turbulence is mentioned it is likely that intensity levels less than 5 per cent were involved.

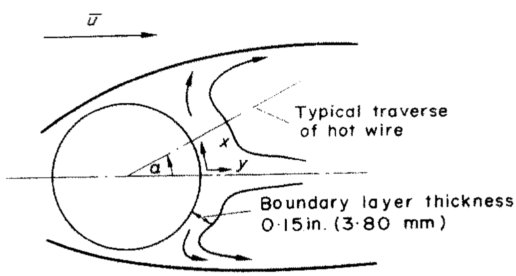

FIG. 9. Location of hot wire traverse.

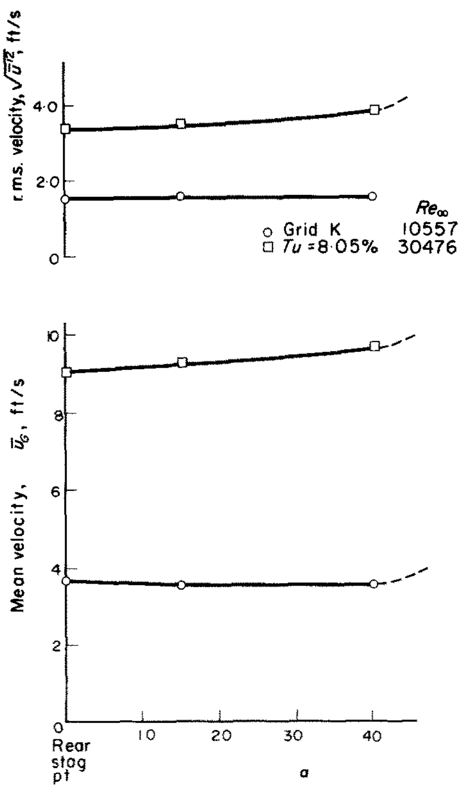

FIG. 10. Variation of mean and fluctuating velocity at the edge of boundary layer in the wake region of the cylinder.

3.4 *Hot wire measurements in the wake region*

The primary purpose of hot wire measurements was to establish at least a preliminary understanding of the flow in the vicinity of the wall and how it effects the heat transfer. For this purpose a knowledge of the boundary layer thickness and variation of mean and fluctuating components at the edge of the boundary layer was desired. The edge of the boundary layer was located by recording radially from the wall the variation of mean and fluctuating velocities at several locations around the wall, indicated in Fig. 9. The layer was found to be approximately 3·8 mm thick and this remained constant for the range of freestream velocities and turbulence intensities tested. No variation in thickness, mean and fluctuating velocities was observed for at least 40° from the rear stagnation point, as can be seen in Fig. 10. For values of α greater than 40° the probe was very close to the separated shear layer and was therefore extremely sensitive to the radial location. The flow could now be considered to be swept into the freestream at these positions and therefore the assumption of flow parallel to the wall was no longer valid. The velocity gradients around the wall $d\bar{u}/dx$ and $d\sqrt{\overline{u'^2}}/dx$ within the 40° region were zero and are in agreement with the pressure and temperature gradients in this region. The mean velocity $\bar{u}$ varied between one fifth and one tenth of the freestream velocity and the ratio $(\sqrt{\overline{u'^2}})/\bar{u}$ was of the order of 0·3 in this region.

The mean velocity profile at the rear stagnation point was found to be representative of the profiles found in the 0°–40° region and is reported here. The data are plotted in non-dimensional form in Fig. 11 where the subscript "G" refers to conditions at the edge of the boundary layer. The data appear to collapse fairly well onto the profile indicated in Fig. 11 although the scatter of data increases rapidly close to the wall. Measurements were only made in the region $y/y_G > 0{\cdot}2$ due to the complexities of the hot wire measurements extremely close to walls. The relationship $\bar{u}/\bar{u}_G = (y/y_G)^n$ has an index n of about 1/5 as shown in Fig. 11. It is interesting to note that this profile lies between the laminar profile and the well known "1/7th Power Law" for turbulent flows. Due to the scatter of data no effects of freestream turbulence on this profile could be detected.

The variation of the r.m.s. velocity fluctuations within the boundary layer is recorded in Fig. 12. Although the scatter of data is much greater a power law relationship with an index m of one seventh appears to be a satisfactory description.

In order to examine the mechanism of interaction between the freestream fluctuations and the wake motion it is necessary to look at the frequency spectrum of the r.m.s. fluctuation. Analyses of this kind rapidly accumulate data and therefore spectrum surveys were restricted

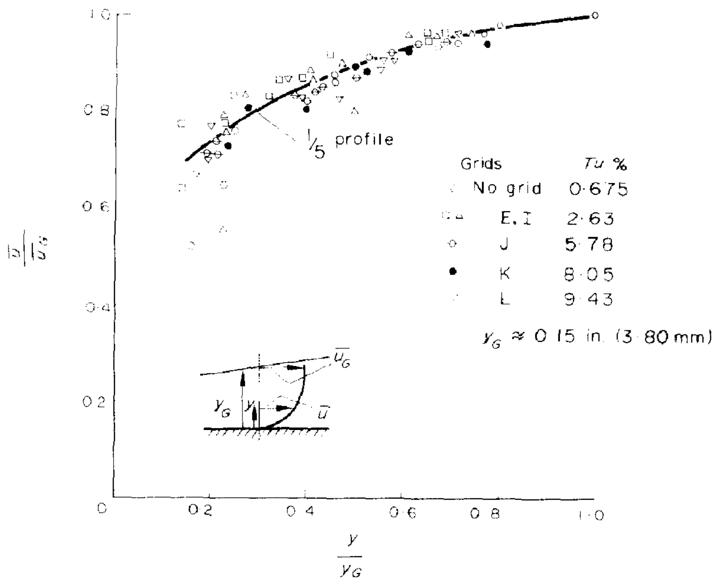

FIG. 11. Non-dimensional mean velocity profiles at rear stagnation point.

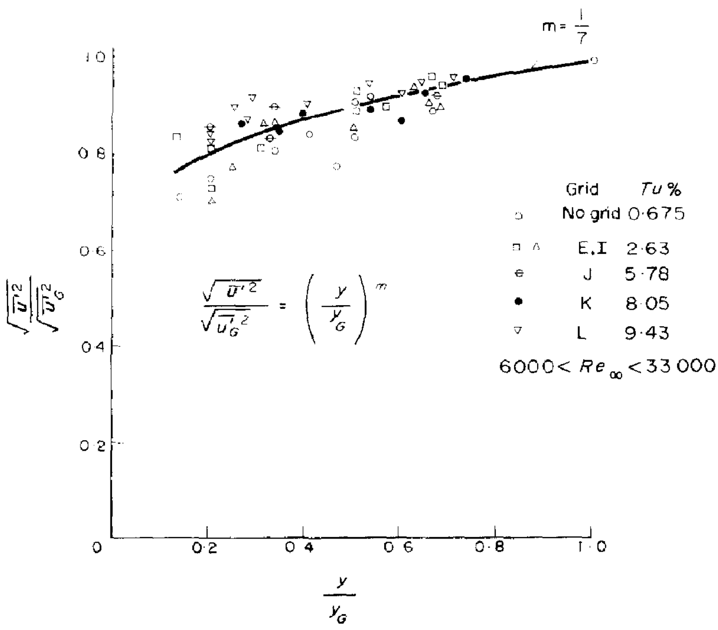

FIG. 12. Non-dimensional fluctuating velocity profile at rear stagnation point.

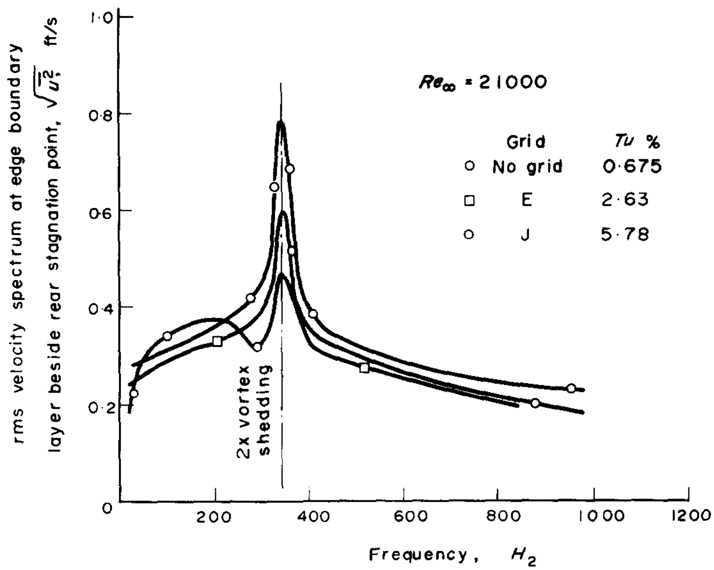

FIG. 13. Effect of freestream turbulence on the vortex shedding process in the wake region.

to the edge of the boundary layer at the rear stagnation point. It is evident from Fig. 13 that the increase in r.m.s. fluctuation level is due to an amplification of the periodic component of the spectrum, only small increases in the turbulent component being observed. The periodic component was observed to occur at twice the vortex shedding frequency which is a direct result of the measurements being made at the rear stagnation point which experiences the effect of vortices shed from both sides of the cylinder. This amplification is analogous to the simple mechanical vibration of spring/mass system with an oscillating force applied. The amplitude of motion of the mass is dependent upon the magnitude and frequency of the forcing function and hence depending upon the amount of damping in the system conventional resonance curves can be drawn. In this problem the freestream fluctuation represents the forcing function. With the available data it is difficult to separate the effect of magnitude (turbulence intensity) and length scale of turbulence and therefore it is not known whether the maximum amplification has been achieved. However, it was observed, although no details are recorded here, that most of the freestream turbulent energy occurred over a low frequency range which included the periodic frequency. It is therefore likely that an amplitude effect is being observed and this is also likely to be near the maximum attainable.

4. DISCUSSION

4.1 *General*

It is now evident from Fig. 7 that the heat transfer from a cylinder in cross flow is extremely sensitive to freestream fluctuations, in the Reynolds number range 5000–35 000. It is also shown in Fig. 6 that a wide variation in the heat transfer exists in the data reported by different investigators. This we attribute to the freestream turbulence effect, which is not normally measured or stated by these investigators. This effect has already been demonstrated by Zijnen [1] when measuring overall heat transfer. Another factor in considering other data in the literature is that the form of surface heating employed is often not clearly described and this can affect the temperature gradient around the cylinder wall.

The present series of tests have shown that all regions of the cylinder are sensitive to the freestream fluctuations. The overall heat transfer is found to be in agreement with that quoted by Richardson as shown in Fig. 8 and the similarity between temperature, mean and fluctuating velocity gradients around the cylinder wall in the region of $\pm 40°$ from the rear stagnation point, as shown in Fig. 5 and 10, suggest that the present data are consistent. It therefore appears that for this range of Reynolds number, increases in freestream turbulence up to 10 per cent are likely to improve heat transfer in the wake region by approximately 100 per cent.

The main difficulties encountered when making hot wire measurements in low mean velocity, highly turbulent flows have already been discussed together with the reasons for using a single wire probe. Despite these difficulties and the errors which are likely to be introduced, the hot wire anemometer provides useful information regarding the flow around the wall. As shown in Fig. 10 both mean velocities and r.m.s. velocity fluctuations at the edge of the boundary layer were found to remain constant for approximately 40° on either side of the rear stagnation point and within experimental error, the boundary layer thickness remained constant at 3·81 mm. This information justified the restriction of further measurements to the rear stagnation point. As shown in Fig. 11 the mean velocity profiles obeyed a 1/5th power law relationship. Of more interest is the r.m.s. velocity shown in Fig. 12 which followed a 1/7th power law, although the experimental scatter is considerable. The lack of an observable maximum near the wall in this curve suggests that the turbulence generated near the wall is small in comparison with that generated by the vortices. This suggests that the laws governing the decay of turbulence in this region will be primarily dependent on diffusion and dissipation.

Examination in Fig. 13 of the frequency spectrum in the wake region shows a predominantly periodic motion for the range of Reynolds numbers tested and this periodic component is amplified by the freestream oscillations. This apparent resonance effect is likely to be the basis of the interaction between the wake and the freestream. This resonance effect decreases with increasing Reynolds number and this is attributed to the increase in apparent viscosity as the turbulent component of the spectrum increases. This is in agreement with the work of Kestin [14] who claimed no improvements in heat transfer in the wake region at a Reynolds number of 200 000.

4.2 *An empirical correlation for the wake region*

A useful relationship can be derived between the local heat transfer and the freestream Reynolds number *Re* and turbulence intensity Tu_∞ by considering the fluid motion in the wake. Since it is expected that the fluctuating motion rather than the mean motion is the most

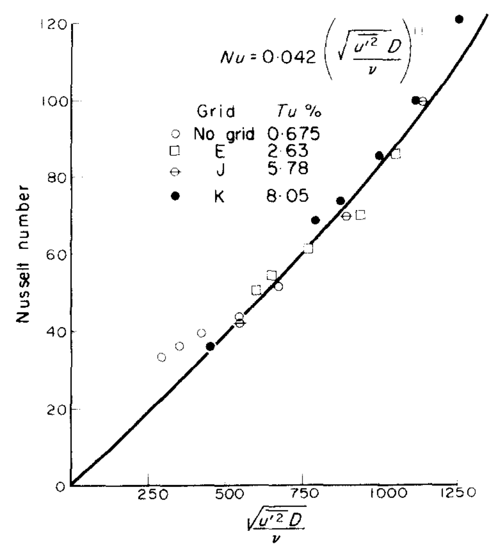

FIG. 14. A relationship between experimental heat transfer and turbulence at rear stagnation point.

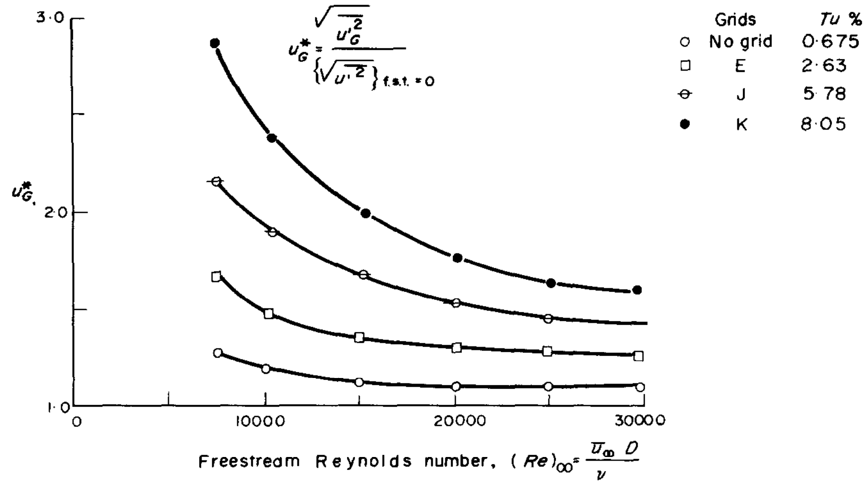

FIG. 15. Effectiveness of freestream turbulence on rear stagnation point fluctuations.

important component in this region the turbulent energy k is a useful parameter in any correlation. It has the additional advantage of being non-directional. Although only one component $\overline{u'^2}$ has been measured this is regarded as sufficiently accurate at this stage of the investigation. A good correlation is in fact obtained in Fig. 14 which suggests that the Nusselt number Nu is dependent solely on the turbulent energy measured by the fluctuating Reynolds number $(\sqrt{\overline{u_G'^2}})D/\nu$. As the turbulent energy decreases to zero at low Reynolds number so will the Nusselt number approach the limit of that obtained by natural convection, negligibly small.

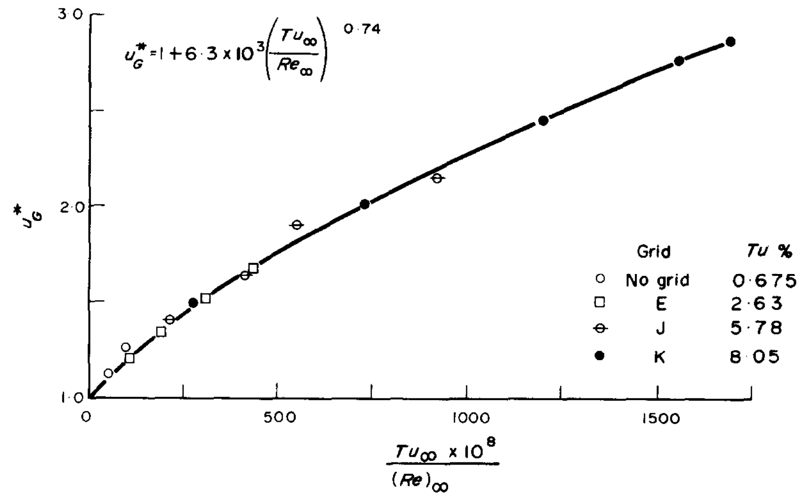

FIG. 16. A correlation between freestream turbulence and the wake fluctuations.

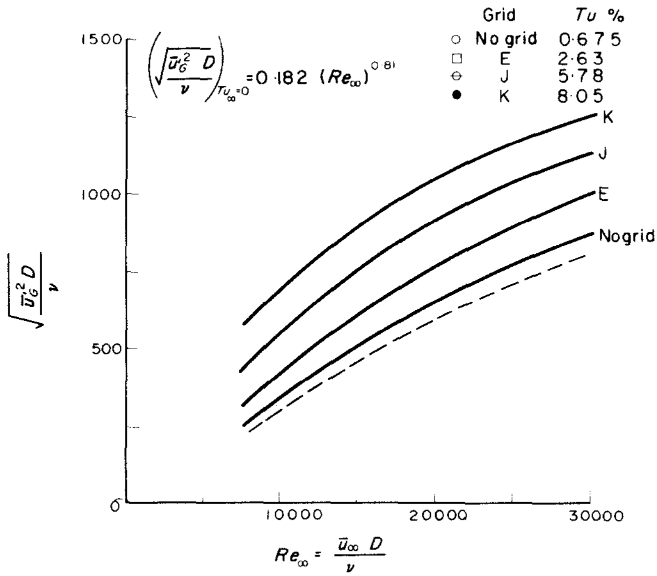

FIG. 17. Effect of freestream turbulence on fluctuations at rear stagnation point.

It is therefore adequate to relate Nusselt number to turbulent energy by an expression of the form,

$$Nu = 0{\cdot}042\left(\frac{(\sqrt{u_G'^2})D}{\nu}\right)^{1\cdot1}. \tag{5}$$

The closeness of fit of this expression to the data is shown in Fig. 14.

The turbulent energy in the wake can be related to the freestream motion by considering the effectiveness of the freestream turbulence shown in Fig. 15. It was found possible to correlate these data by plotting the ratio u_G^* of the fluctuations at the rear stagnation point to the fluctuations obtained with no freestream turbulence against the ratio of the mainstream turbulence to the mainstream Reynolds number $(Tu)_\infty/(Re)_\infty$, as shown in Fig. 16. A good fit to the data was obtained by the equation

$$u_G^* = 1 + 6{\cdot}3 \times 10^3\left(\frac{Tu_\infty}{Re_\infty}\right)^{0\cdot74}. \tag{6}$$

The data for the limiting case of zero freestream turbulence was obtained by extrapolation in Fig. 17 and is described by the relationship

$$\left(\frac{(\sqrt{\bar{u}'^2})D}{\nu}\right)_{Tu_\infty=0} = 0{\cdot}182\,(Re_\infty)^{0\cdot81}. \tag{7}$$

By combining equations (5)–(7) the following expression for the Nusselt number at the rear stagnation point is obtained

$$Nu = 0{\cdot}0065\,(Re_\infty)^{0\cdot89} + \left[1 + 6{\cdot}3 \times 10^3\left(\frac{Tu_\infty}{Re_\infty}\right)^{0\cdot74}\right]^{1\cdot1}. \tag{8}$$

When there is zero freestream turbulence this expression reduces to the form $Nu \propto Re_\infty^{0\cdot89}$ which is similar to that obtained by Richardson [2] experimentally and Spalding [15] theoretically.

4.3 *A simple theory for the wake region*

The relationship between heat transfer and turbulent energy in equation (5) can be justified by the following simple one dimensional theory.

Starting from the familiar Reynolds analogy

$$\frac{q_w}{\tau} = \frac{c_p}{(Pr)} \frac{\frac{dT}{dy}}{\frac{d\bar{u}}{dy}}. \quad (9)$$

and integrating across the boundary layer,

$$\Delta T = \frac{q_w(Pr)}{c_p} \int_0^{y_G} \frac{1}{\tau} \frac{d\bar{u}}{dy} dy. \quad (10)$$

The integral on the right hand side can be divided into two parts, the laminar region $0 < y < y_0$ and a turbulent region $y_0 < y < y_G$. With $m = \frac{1}{7}$ and $n = \frac{1}{5}$, the latter integral can be shown to amount to about 30 per cent of the total integral [16]. As a crude approximation we shall neglect it. Then

$$\Delta T = \frac{q_w(Pr)y_0}{c_p \rho v}. \quad (11)$$

There are several expressions given in the literature which relate shear stress to turbulent energy. One, which is discussed in [15], is

$$\frac{\tau}{\rho} = ck^{\frac{1}{2}} y \frac{d\bar{u}}{dy}. \quad (12)$$

By changing the constant c to a variable which is a function of the turbulent energy at the edge of the boundary layer k_G a suitable relationship can be written

$$c = B\left(\frac{k_G^{\frac{1}{2}} D}{v}\right)^p. \quad (13)$$

These expressions may then be used to relate the thickness y_0 to condition at the edge of the boundary layer using the experimental relationship indicated in Fig. 12. Thus

$$y_0 = \left[\frac{v^{p+1} y_G^m}{B(k_G^{\frac{1}{2}})^{p+1} D^p}\right]^{\frac{1}{m+1}}. \quad (14)$$

By substituting this relationship into equation (11) the following expression is obtained

$$Nu = (B)^{\frac{1}{m+1}} \left(\frac{Dk_G^{\frac{1}{2}}}{v}\right)^{\frac{p+1}{m+1}} \left(\frac{D}{y_G}\right)^{\frac{m}{m+1}}. \quad (15)$$

From the experimental evidence, the boundary-layer thickness was constant; thus the second term is the only variable on the right-hand side. The velocity profiles of Fig. 12 suggest that $m = \frac{1}{7}$ and the correlation of equation (5) shows that $(p + 1)/(m + 1) = 1{\cdot}1$. This implies that $p = 0{\cdot}26$. A more elaborate two dimensional theory of the heat transfer at the rear stagnation point suggests that $p = 0{\cdot}5$ [16]. It is hoped to publish this theory more widely at a later date.

The above is not a satisfactory theory in the sense of predicting equation (5) exactly, but it does show how the Nusselt number is determined by the Reynolds number $\left(\frac{Dk_G^{\frac{1}{2}}}{v}\right)$, involving the local turbulent energy k_G. To obtain the index 1·1, however, the constant c in equation (12) must also be allowed to vary with this Reynolds number.

5. CONCLUSIONS

The results of this paper are based on experiments carried out on a single tube of diameter $\frac{3}{4}$ in. (19 mm) heated uniformly to produce temperature differences in the range 8–15°C. With this restriction the following conclusions can be drawn.

1. The heat transfer in the wake region is strongly dependent upon the turbulent energy and is described by equation (8). In the Reynolds number range 5000–35 000, increases in freestream turbulence intensity of 10 per cent result in improvements in heat transfer up to 100 per cent.
2. The boundary layer thickness in the region ±40° of the rear stagnation is of constant thickness for this range of Reynolds number.
3. Within experimental error the mean velocity and r.m.s. velocity fluctuation profile at the rear stagnation point region obey a 1/5th and 1/7th power law relationship, respectively.

4. The improvements in heat transfer in the wake are the result of a resonant interaction between the wake motion and the freestream fluctuations at twice the vortex shedding frequency.

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ETUDE EXPÉRIMENTALE DE LA SENSIBILITÉ DU TRANSFERT THERMIQUE À LA TURBULENCE EXTERNE DANS LES SILLAGES DE CYLINDRES PLACÉS FRONTALEMENT À L'ÉCOULEMENT

Résumé—On présente une étude expérimentale de l'effet de la turbulence du courant principal sur le transfert thermique à partir d'un cylindre. On chauffe électriquement le cylindre et on s'intéresse à l'effet de la turbulence sur la distribution de pression et sur le coefficient de transfert thermique local autour du cylindre; les derniers résultats sont comparés avec une autre information publiée.

Des mesures de la vitesse moyenne et de la valeur efficace des fluctuations de vitesse dans la région $\pm 40°$ sur la face arrière du cylindre ont été faites à l'aide d'un anémomètre à fil unique et des profils réduits de vitesse ont été représentés par des lois de puissance simples. Le spectre de fréquence à l'arrière du cylindre est dominé par la fréquence double de celle d'échappement des tourbillons.

Les résultats de transfert thermique à l'arrière du cylindre sont unifiés en fonction du nombre de Reynolds d'énergie de turbulence et cette énergie de turbulence est rapportée au niveau de turbulence du courant principal et au nombre de Reynolds du courant principal pour obtenir une relation empirique entre le transfert thermique à l'arrière du cylindre, ce dernier nombre de Reynolds et le niveau de turbulence. On donne une interprétation théorique monodimensionnelle simple des résultats. Ceux-ci montrent que dans le domaine du nombre de Reynolds compris entre 5 000 et 35 000 des accroissements de 10% de l'intensité de turbulence du courant libre provoquent des augmentations du transfert thermique à l'arrière du cylindre atteignant 100%.

EXPERIMENTELLE UNTERSUCHUNG DER EMPFINDLICHKEIT DER FREISTROMTURBULENZ UND DES WÄRMEÜBERGANGS IM SOG VON QUERANGESTRÖMTEN ZYLINDERN

Zusammenfassung—In einer experimentellen Untersuchung wird über die Wirkung der Hauptstromturbulenz auf den Wärmeübergang von einem Zylinder berichtet. Der Zylinder wird elektrisch beheizt. Der Einfluss der Turbulenz auf die Druckverteilung und den lokalen Wärmeübergangskoeffizienten um den Zylinder wird untersucht. Diese Daten werden mit anderen Veröffentlichungen verglichen.

Zur Messung der mittleren Geschwindigkeit und der Geschwindigkeitsschwankungen nach dem mittleren Fehlerquadrat im Bereich $\pm 40°$ von der Zylinderrückseite aus gemessen, dient ein Einzeldraht-Anemometer. Für die reduzierten Geschwindigkeitsprofile gilt eine einfache Potenzbeziehung. Es wird gezeigt, dass das Frequenzspektrum an der Zylinderrückseite hauptsächlich von der doppelten Wirbelablösungsfrequenz bestimmt wird.

Die Wärmeübergangsdaten werden auf die Zylinderrückseite bezogen und mit Hilfe einer Reynoldszahl der Turbulenzenergie miteinander verknüpft. Diese Turbulenzenergie wird auf die Turbulenzebene des Hauptstroms bezogen. Die Reynoldszahl der Hauptströmung dient dazu, eine empirische Beziehung aufzustellen, die den Wärmeübergang an der Zylinderrückseite mit der Reynoldszahl der Hauptströmung und der Turbulenzebene verbindet. Für diese Ergebnisse wird eine einfache eindimensionale theoretische Darstellung gegeben. Die Ergebnisse zeigen, dass in einem Bereich der Reynolds-Zahlen von 5000–35000 ein Zuwachs der Stärke der Freistromturbulenz von 10%, für den Wärmeübergang an der Zylinderrückseite eine Erhöhung bis zu 100% bedeutet.

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ВЛИЯКИЯ ТУРБУЛЕНТНОСТИ НАБЕГАЮЩЕГО ПОТОКА НА ТЕПЛООБМЕН В СЛЕДАХ ЦИЛИНДРОВ ПРИ ПОПЕРЕЧНОМ ОБТЕКАНИИ

Аннотация—В статье описывается экспериментальное исследование влияния турбулентности набегающего потока на теплообмен цилиндра, нагреваемого электрическим током. Сообщается о влиянии турбулентности на распределение давления по периметру и на локальный коэффициент теплообмена; дается сравнение полученных результатов с данными, опубликованными другими авторами.

Измерения средней скорости и среднеквадратичных флуктуаций скорости в следе за цилиндром производились с помощью анемометра с однониточным датчиком; выведены простые степенные зависимости для профилей скорости. Показано, что в спектре частот в следе за цилиндром доминирует удвоенная частота срыва вихрей.

Данные по теплообмену в кормовой области цилиндра обобщаются с помощью числа Рейнольдса для турбулентной энергиий приводится отношение этой турбулентной энергии к степени турбулентности набегающего потока и числу Рейнольдса основного потока для того, чтобы получить эмпирическую зависимость, связывающую перенос тепла в следе за цилиндром с числом Рейнольдса основного потока и степенью турбулентности. Дается простая одномерная трактовка этих результатов.

Результаты показывают, что в диапазоне чисел Рейнольдса набегающего потока от 5 000 до 35 000 повышение интенсивности турбулентности в свободном потоке на 10% приводит к увеличению интенсивности теплообмена в кормовой зоне цилиндра на величину до 100%.