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THE EFFECT OF TURBULENCE ON HEAT TRANSFER FROM HEATED CYLINDERS

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Abstract—Data were obtained on the effect of free-stream turbulence on heat transfer from heated cylinders placed normal to an air stream. Tests were conducted for a Reynolds number range from 109 000 to 302 000 at values of turbulence intensity from 0·40 to 14·20 per cent. The diameters of the test cylinders used were 4·5 and 7·6 in respectively and the ratio of cylinder diameter to test section width were 0·075 and 0·127. The ratio of turbulence scale to cylinder diameter ranged from 0·015 to 0·095. The scale of turbulence varied from 0·113 to 0·426 in during the study. The structure of the free-stream is characterized and the effects of the artificially induced turbulence on both the local and overall heat transfer are determined.

The data are discussed and compared with results of previous studies where appropriate. The results are explained in part by available theory and also indicate areas that require further investigation. A correlation equation is presented that enables the stagnation point heat transfer as a function of turbulence intensity to be determined. The augmentation of heat transfer through the laminar boundary layer as a function of turbulence intensity is also determined. Attempts to determine the effect of turbulence intensity on the overall or average heat transfer were partially successful. A correlation equation for predicting the overall heat transfer as a function of turbulence intensity over a narrow range of correlation parameters is presented.

NOMENCLATURE

A, amplitude factor;
A_s, surface area [ft^2];
B, constant;
D, diameter [in];
h, local heat-transfer coefficient [Btu/h ft^2 °F];
I, current [A];
k, thermal conductivity [Btu/h ft °F];
L, scale of turbulence [in];
$\overline{Nu}$, average Nusselt number;
Nu, local Nusselt number;
n, exponent;
P, power to test strip [W];
P_B, barometric pressure [in Hg];
P_v, velocity pressure [in H_2O];
Pr, Prandtl number;
q, rate of heat transfer [Btu/h];
q'', heat flux [Btu/h ft^2];
R, resistance [Ω];
$R(y)$, correlation coefficient;
Sc, Schmidt number;
Sh, Sherwood number;
T, temperature [°R];
Tu, overall turbulence intensity [%];
t, temperature [°F];
U, mean flow velocity [ft/s];
u, instantaneous velocity component [ft/s];
$\bar{u}$, time-mean velocity component [ft/s];
u', RMS value of velocity component [ft/s];
δu, fluctuating velocity component [ft/s];
u'/U, component of turbulence intensity [%];
v, instantaneous velocity component [ft/s];
$\bar{v}$, time-mean velocity component [ft/s];
v', RMS value of velocity component, Appendix B [ft/s];
δv, fluctuating velocity component [ft/s];
v'/U, component of turbulence intensity [%];
w, instantaneous velocity component [ft/s];
$\bar{w}$, time-mean velocity component [ft/s];
w', RMS value of velocity component [ft/s];
δw, fluctuating velocity component [ft/s];
w'/U, component of turbulence intensity [%];
x, longitudinal spatial position [in];
y, horizontal spacial position [in];
z, vertical spacial position [in].

Greek symbols

ε, emissivity [ft^2/h];
θ, angular position [deg];
Λ, Taylor parameter;
ν, kinematic viscosity [ft^2/s];
ρ, density [lbm/ft^3];
σ, Stefan–Boltzmann constant;
ϕ, function;
ψ, function;
ψ^*, augmentation factor.

Subscripts

s, refers to surface;
∞, refers to conditions in free stream;
θ, refers to local value;
v, refers to velocity;
B, refers to barometric pressure.

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INTRODUCTION

THE ANALYSIS of forced convection heat transfer from cylinders has been the subject of numerous investigations over the last four decades. The differences in the published data and recent knowledge that two turbulent, or oscillating parallel streams are not similar when only the Reynolds numbers based on the average velocity are identical, have prompted new interest in the field. This study is concerned with the experimental determination of the influence of turbulence on the heat transfer from a cylinder in crossflow. All three components of intensity and the scale of turbulence were measured with the use of two hot wire anemometers and a correlator. The scale of turbulence to cylinder diameter ratio was held very small to minimize the influence of the scale on the heat transfer. The data indicate the influence of turbulence intensity on heat transfer and are compared with previous data. The data are discussed and correlation equations are presented for predicting the effect of turbulence intensity on heat transfer.

BACKGROUND AND RELATED STUDIES

In turbulent flow, the amplitudes and frequencies of the random fluctuating components of two parallel streams may differ. However, average values of intensity and scale of turbulence are adequate to establish similarity. Intensity of turbulence indicates the amplitude of the fluctuating components of velocity and the scale indicates the relative size of the turbulent eddies present. The overall turbulence intensity of a stream [1] is

$$Tu = \frac{\{\frac{1}{3}[\overline{(\delta u)^2} + \overline{(\delta y)^2} + \overline{(\delta w)^2}]\}^{\frac{1}{2}}}{U}. \qquad (1)$$

If the turbulence is isotropic, equation (1) becomes

$$Tu = \frac{[\overline{(\delta u)^2}]^{\frac{1}{2}}}{U} = \frac{u'}{U} \qquad (2)$$

where u' signifies the RMS value of the fluctuating component δu.

Dryden *et al.* [2] found the scale of turbulence can be determined by measuring the correlation coefficient of two longitudinal fluctuating components of velocity at two points in space separated a distance apart. That is, by specifying the quantity

$$R(y) = \frac{\overline{\delta u_1} . \overline{\delta u_2}}{[\overline{(\delta u_1)^2}]^{\frac{1}{2}}[\overline{(\delta u_2)^2}]^{\frac{1}{2}}} \qquad (3)$$

$R(y)$ decreases rapidly to zero as y increases and the scale of turbulence is given by

$$L \int_0^\infty R(y)\,\mathrm{d}y. \qquad (4)$$

The intensity and scale of turbulence of the free stream markedly affect heat-transfer rates in forced convection [3–11]. Unfortunately, the pioneer studies [3–11] on heat transfer from cylinders have failed to report the character of the turbulent flow although Reynolds numbers were varied from 5000 to 426000. Other studies [1–10] have added to the understanding

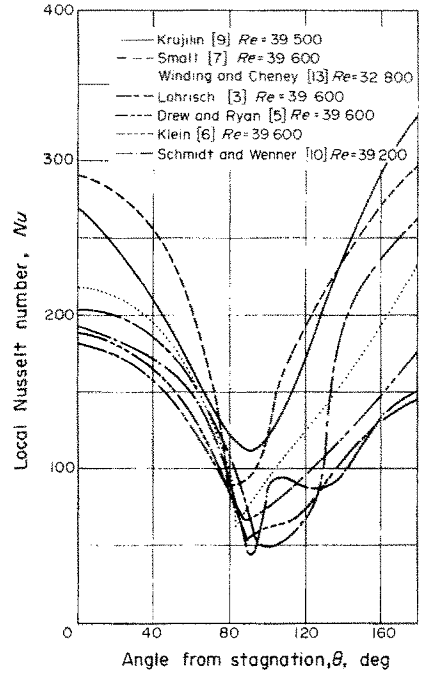

FIG. 1. Comparison of data of several investigators at one Reynolds number.

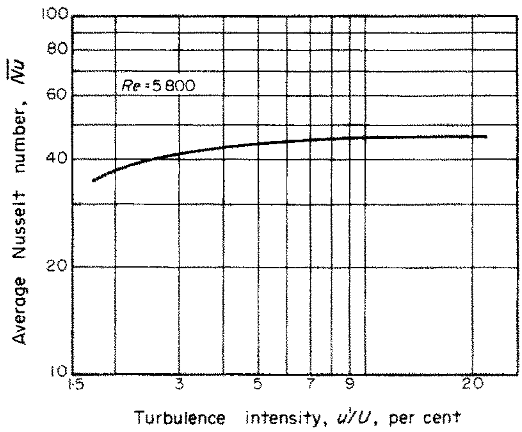

FIG. 2. Effect of turbulence intensity on heat transfer at constant Reynolds number [16].

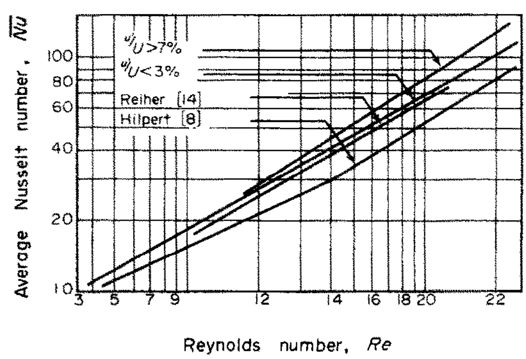

FIG. 3. Effect of Reynolds number on heat transfer at different turbulence intensities [16].

of heat transfer from cylinders in cross flow but the results present some conflict as seen in Fig. 1. The first quantitative investigation on the influence of turbulence intensity on heat transfer from a circular cylinder was reported in [16]. Figures 2 and 3 show typical results obtained in [16] and a comparison of data from other studies. Giedt [17] concluded from studies using a turbulence producing grid in the stream that the effect of the grid was to yield heat-transfer results characteristic of velocities higher than the measured free stream velocity. The work of [18] and [19] over a range of turbulence intensities and Reynolds numbers indicated that heat transfer increased with an increase in turbulence and that increasing turbulence intensity had the effect of lowering the critical Reynolds number thus permitting the behavior occurring in the critical range to take place at a much lower Reynolds number. The study of [20] also indicated increased heat transfer with increasing turbulence.

Recent studies [21–43] when taken together form a basis for a better understanding on the effect of turbulence on heat transfer from cylinders in cross-flow. Kestin and Maeder [21, 22] provide convincing proof that a change in the intensity of turbulence in the free stream markedly affects the local rate of heat transfer. Furthermore, their results showed the effect of the turbulence intensity was local and additional to the expected influence on transition and separation of the boundary layer. Van der Hegge Zijen [24] presented

$$\frac{(Nu)\text{ with turbulence}}{(Nu)\text{ without turbulence}} = 1+\phi \tag{5}$$

for the ratio of the heat-transfer rate with turbulence to the heat-transfer rate without turbulence for the same mean cross flow velocity. Van der Hegge Zijen suggested that the optiminium value of turbulence scale to diameter ratio corresponded to a condition of resonance wherein some "effective" frequency of turbulence coincides with the frequency of the eddies shed by the cylinder. Piercy and Richardson [25] studied velocity fluctuations near a cylinder in turbulent cross flow. Their data showed that the stagnation region is a region of maximum disturbance or turbulence which is damped out as the stream flows toward the rear of the cylinder. Kuethe *et al.* [26] found that low frequency components, in the free stream fluctuations, were amplified strongly in the region near the stagnation point. The low frequency fluctuations were attributed to vorticity fluctuations and amplification was attributed to the stretching of vortex filaments in the diverging flow near the stagnation point.

Kestin *et al.* [27] provided an understanding of the effects introduced by free stream oscillations on laminar boundary-layer profiles. Kestin *et al.* [28] presented evidence that the local effects turbulence had on laminar boundary layers were absent for the case of a flat plate at zero incidence and concluded that large effects from changes in free stream turbulence occur only in the presence of a favorable pressure gradient. Kestin *et al.* [29] demonstrated the local effect of turbulence intensity on the rate of heat transfer and thus

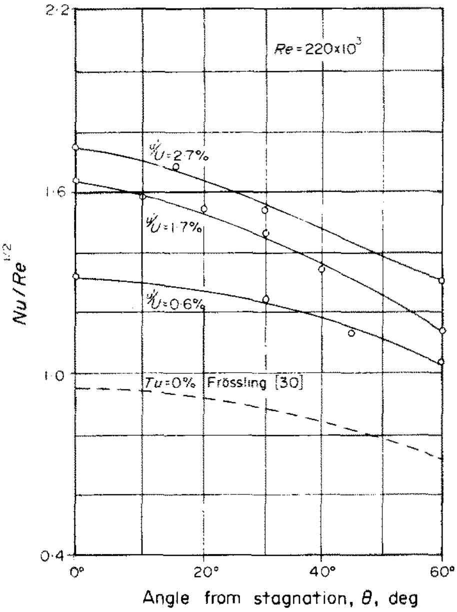

FIG. 4. Variation of local heat transfer around a cylinder with turbulence intensity for Re = 220 000 [29].

on the characteristics of the associated laminar boundary layer. Typical data from [29] is seen in Fig. 4. Also Fig. 4 shows Frössling's theoretical curve [30] for zero turbulence intensity. Heretofore, few studies other than [25, 26] have given an adequate explanation of the effect of free stream turbulence on heat transfer. Sutera *et al.* [31], following [26], theorized that vorticity amplification by stretching was a possible and perhaps the dominant underlying mechanism responsible for the effect. Their mathematical model did show that amplification of vorticity on a large scale can occur. They found the thermal boundary layer to be more sensitive than the velocity boundary layer to the stretching of vortex filaments and that the thermal boundary-layer sensitivity increases with Prandtl number. Sutera [32] continued this study with similar results.

Smith and Kuethe [33] developed a relation for heat transfer and skin friction near the stagnation point. The Reynolds stresses were assumed proportional to the imposed turbulence level in the ambient flow and to the distance from the wall. The theory is compared with experiment in Fig. 5. The data exhibit a Reynolds number effect for Reynolds numbers less than 10^5 that is not accounted by the theory.

The presence of three dimensional vortex structures in the boundary layer near the stagnation point of a cylinder was shown by Brun *et al.* [35]. This study allowed examination of the vortex theory proposal in [31]. Calculations were performed for different values of the amplitude factor. The results indicated a disagreement with the result of [32] for heat transfer at the stagnation point for fluid with a Prandtl number equal to 100. Sutera and Williams [37] examined solutions to the vortex amplification model as Prandtl $\rightarrow \infty$ and the results indicated 95 per cent of the asymptotic value of the heat-transfer coefficient is obtained at Prandtl number equal to 100.

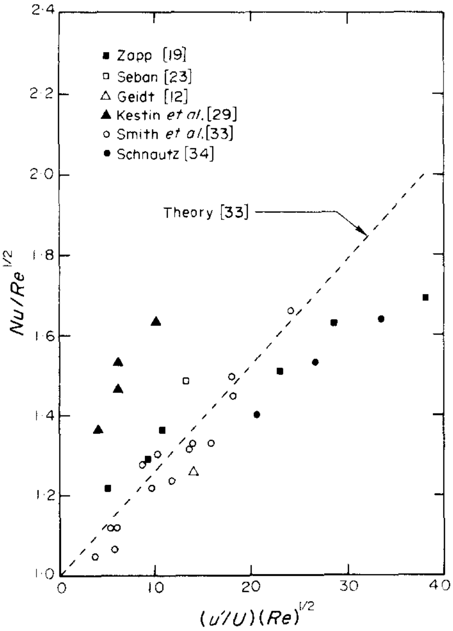

FIG. 5. Comparison of heat-transfer data at the stagnation line of cylinders in cross flow with the theory of [33].

More recently, Kestin and Wood [38–40] demonstrated that the flow field in the laminar boundary layer formed at the front of a cylinder is essentially three dimensional in character. Their results are compared with those of others in Fig. 6. The authors mentioned that it appears that Smith and Kuethe's theory should have been based on Frössling's [30] theoretical (zero turbulence) calculations $Nu/Re^{\frac{1}{2}} = 0{\cdot}945$ rather than a value of 1·00 [44].

The related studies indicate turbulence does influence the heat transfer from cylinders in cross flow. This influence has been attributed to the increase in heat transfer in the laminar portion of the boundary layer, but certainly turbulence affects the transfer in the turbulent and separated regions though as yet an unexplained way. Perhaps, it is due solely to the influence on the flow field. The vorticity amplification theory tends to explain the increased heat transfer in the stagnation region although the theory is not explicit in that it relies on experimental determinations of certain parameters. The vorticity amplification theory and associated work has contributed greatly to a better understanding of heat transfer through laminar boundary layers in the presence of a turbulent free stream.

EQUIPMENT

An open circuit wind tunnel with a contraction ratio of 9:1 and rectangular test section of 5 × 3 ft and two cylinders (4·5 and 7·6 in in diameter) with heated test strips were used to obtain the data. The coordinate system used to locate stations in the tunnel test section had an origin at the longitudinal center of the test section on the tunnel geometric centerline. The test cylinders were mounted to permit rotation of the cylinder around the longitudinal center to position the test strips at a given angle θ. A honeycomb structure at the inlet of the tunnel, plus provisions for installing damping screens and turbulence inducing grids at various locations upstream of the test cylinders, provided variations in free stream conditions. Turbulence characteristics were measured with two Disa Anemometers, 55A01, and a Disa Signal Indicator and Correlator, 55A06. Probe transducers were a Disa single wire probe, 55A22, and an x–y probe, 55A32.

Three types of turbulence inducing devices were used:

1. Fine mesh damping screens;
2. Large mesh woven wire grid;
3. Wooden grid constructed of dowels.

Table 1. Fine mesh damping screens

Screen number	Mesh size	Wire diameter (in)	Wire spacing (in)	Open area (%)
1	24	0·0075	0·0342	69·7
2	26	0·0075	0·0310	64·8

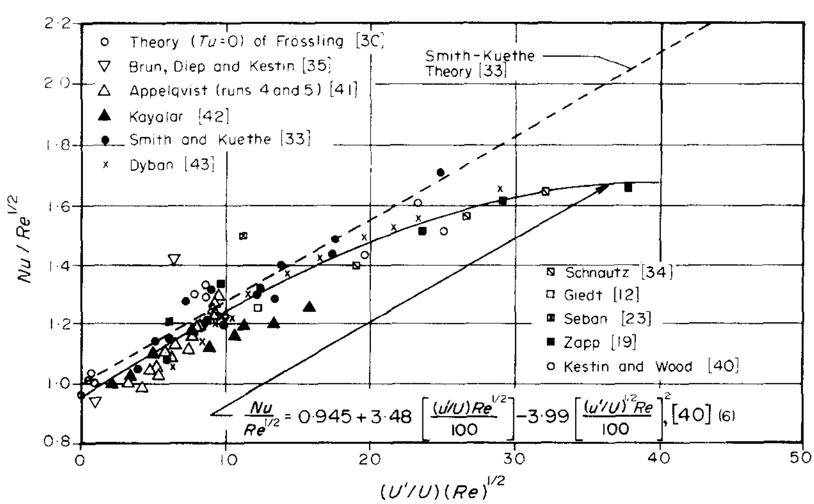

FIG. 6. Comparison of heat- (mass-) transfer data at the stagnation line of cylinders in cross flow.

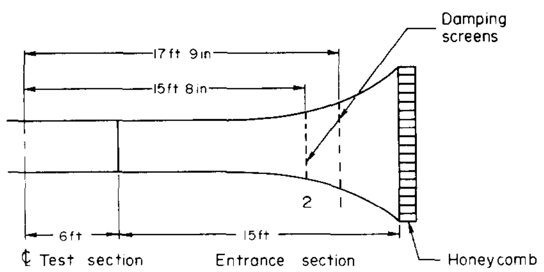

FIG. 7. Location of damping screens in entrance section of forced convection facility.

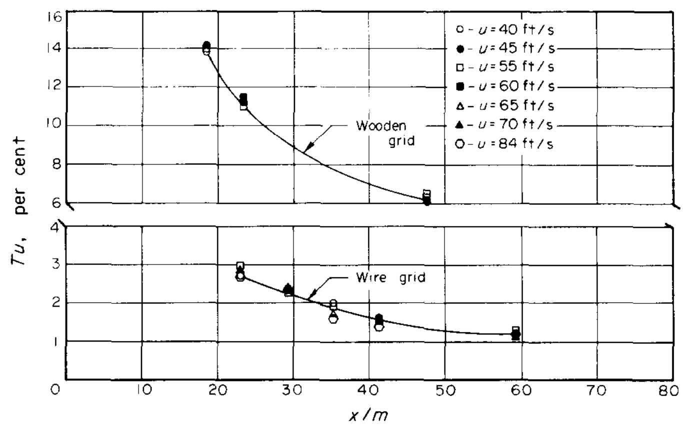

FIG. 8. Variation of turbulence intensity with grids at selected distances, X/M, upstream from centerline of test section.

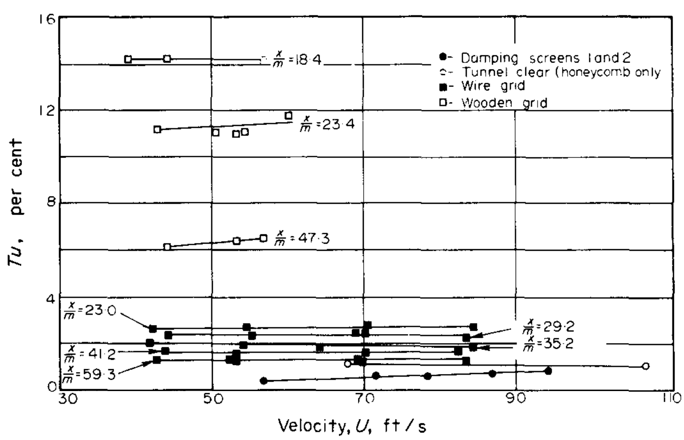

FIG. 9. Variation of turbulence intensity with velocity with grids at selected distances, X/M, upstream from centerline of test section.

Table 1 describes the fine mesh screens and Fig. 7 shows the location of the screens and the honeycomb relative to the center of the test section. Installation of both fine mesh screens reduced turbulence intensities from about 1 per cent to as low as 0·36 per cent. One large mesh wire grid (0·08 in dia with 1·0 in spacing on centers) was used to induce turbulence intensities from 1 to 3 per cent. This grid was placed from 0·5 to 5·5 ft upstream of the geometric center of the test section. The grid of wooden dowels (0·625 in in diameter with 1·25 in spacing on center) produced turbulence intensities from 5 to 14 per cent. The wooden grid was positioned from 0·5 to 5·5 ft upstream of the geometric center of the test section.

Figures 8 and 9 give an indication of the control obtained with the screens and grids. Figure 8 shows turbulence intensity (equation 1) vs the ratio of x to m, the wire spacing. It should be noted that differences of up to 30 per cent were determined for the intensity components. Figure 9 indicates that the turbulence level is not a strong function of velocity over the range of air speeds studied.

Figures 10–12 show the variation of correlation coefficient R, with probe separation y. Also shown are the scale values L obtained by integrating the areas under the curves. Scale data were taken for test conditions and ranged from 0·1133 to 0·4255 in. The data in Fig. 11 indicate the scale of turbulence is independent of velocity.

Two similarly constructed cylinders were used in the tests. A 4·5 in dia cylinder [46] was used to provide checkpoints for the data taken with the main 7·6 in dia cylinder. Figure 13 shows the construction details of

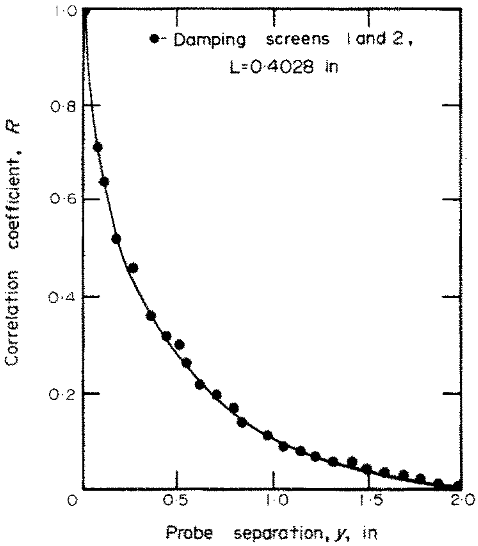

FIG. 10. Variation of correlation coefficient with probe separation (damping screens 1 and 2 in tunnel).

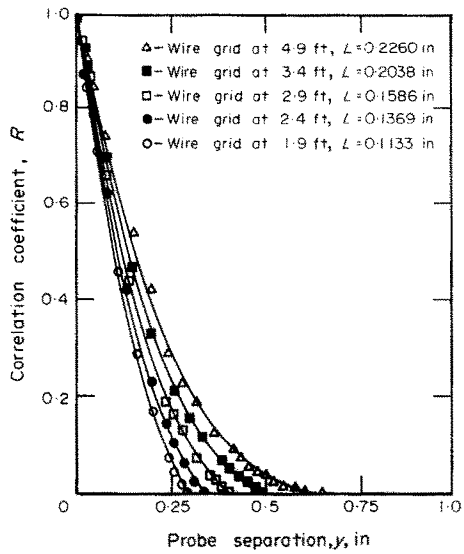

FIG. 12. Variation of correlation coefficient with probe separation (wire grid in tunnel).

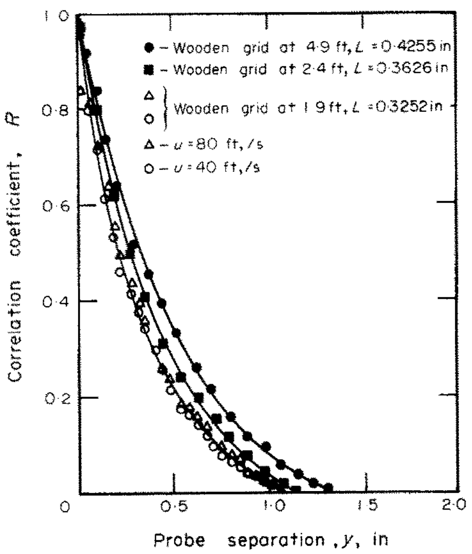

FIG. 11. Variation of correlation coefficient with probe separation (wooden grid in tunnel).

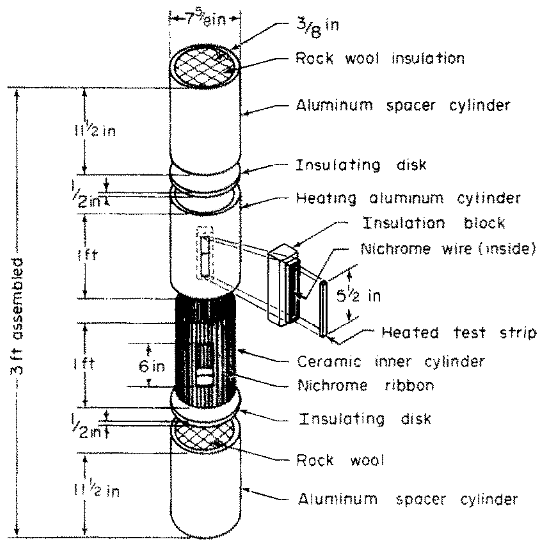

FIG. 13. Details of 7·6 in dia test cylinder.

the 7·6 in cylinder. As is seen, data were taken from a heated test strip imbedded in an insulating block of foamed fused silica with a conductivity of approximately 0·07 Btu/h ft °F [47]. The test strip was heated by a nichrome ribbon on the back surface and thermal control of the main cylinder assembly was obtained with a resistance heater placed inside the cylinder. A 30 gage copper–constantan thermocouple was placed at the mid-point of the test strip and similar thermocouples were positioned on each side of the test strip insulation at the surface of the main cylinder.

At each angular position, the power input to the test section heater was adjusted until the test strip surface temperature was equal to the main cylinder surface temperature indicated by the thermocouples mounted on each side of the test strip. It was assumed that all power initiated into the test strip was transferred from the surface of the test strip. Thus, after correcting the data for radiation loss, the local heat-transfer coefficient was calculated directly by knowing the power input to the test strip, test strip surface area, and the temperature difference between the cylinder surface and free stream. Additional details concerning the construction and operation of the test cylinders are given in [46, 53].

TEST RESULTS AND ANALYSIS

Twenty-four tests were run with the 7·6 in cylinder and nine were run with the 4·5 in cylinder. The range of variables was

Re,	Reynolds number	109 107–302 290
Tu,	turbulence intensity	0·4–14·2 per cent
L,	scale of turbulence	0·1130–0·4255 in.

Turbulence was measured without the test cylinders in the tunnel. The data indicated strong anisotropic conditions of turbulence; thus the overall turbulence intensity Tu was used as the correlating parameter. Scale effects on heat transfer were not evaluated due to the limited range of scale values and the small ratio of turbulence scale to cylinder diameter which is in keeping with [41]. Blockage effects were not accounted for since the ratios of test section width to cylinder diameters were large. Blockage effects on the Reynolds numbers were estimated to be less than 0·5 per cent.

The local measurements indicated that increasing the turbulence intensity for a given Reynolds number increased the heat transfer at the stagnation point and through the laminar boundary layer. Typical data are shown in Figs. 14 and 15 along with Frössling's curve [30]. The figures show that for a given Reynolds number increasing the turbulence intensity increases the heat transfer over the theoretical curve, ($Tu = 0$), for $0 < \theta < 40$ degrees. Analysis of all results at the stagnation point are presented in Table 2 and one may interpret the result as applicable over the range $0 < \theta < 40$ degrees. The calculations in Table 2 were based on the measured values of $Nu/Re^{\frac{1}{2}}$ at $\theta = 0$ and Frössling's zero turbulence value of $Nu/Re^{\frac{1}{2}} = 0{\cdot}945$ at $\theta = 0$. Thus the table really indicates the increase in the rates of heat transfer over the theoretical value only for the stagnation point, but since the increase is almost constant over the range $0 < \theta < 40$ degrees, the results may be interpreted as applying over the range $0 < \theta < 40$ degrees.

Figure 16 is a graphical display of the results of Table 2. The figure leads one to conclude that the largest increases occur for small increases in turbulence intensity at the smaller values of turbulence intensity. No explanation can be given for the spike in the intensity range 2·77–2·85. The percent increase seems to approach an asymptotic value of 60 per cent for intensity approaching 14 per cent but this should be checked by further experimentation. Figure 17 is a display of the data as suggested by Kestin and Wood [40]. The augmentation factor describes the effect of turbulence intensity on the heat transfer through the laminar boundary layer. The figure shows no discernible effect of Reynolds number although some scattering of the data is observed. These results are in conflict with those of [40] as can be seen by contrasting Figs. 17 and 18. The disagreement could be due to the fact that the data of [40] were limited to a range of Reynolds numbers terminating at 125 000. The majority of the data in Fig. 18 are for Reynolds numbers greater than 150 000 indicating that any Reynolds number effect is not present for values of the Reynolds number higher than those of [40].

Smith and Kuethe [33] assumed a linear relation between the stagnation point Nusselt number and the free stream turbulence level for a constant Reynolds number as seen in Fig. 19. The results of this study are seen in Fig. 20 and it is seen that the Nusselt number is not a linear function of turbulence intensity in the range, $0 < Tu < 16$ per cent. In fairness to [33], when

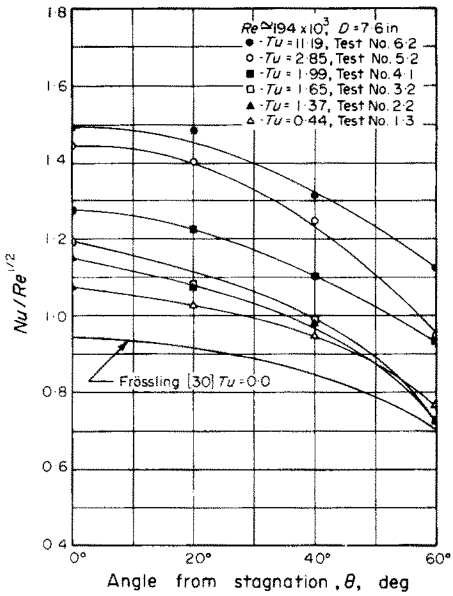

FIG. 14. Effect of turbulence intensity on heat transfer through the laminar boundary layer for $Re = 194 \times 10^3$

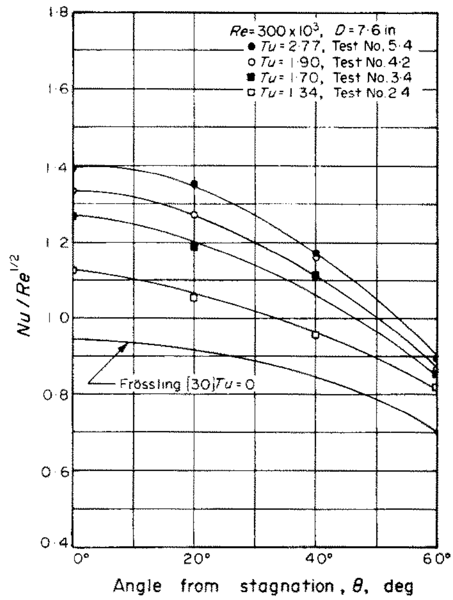

FIG. 15. Effect of turbulence on heat transfer through the laminar boundary layer for $Re = 300 \times 10^3$.

the present results are graphed for $0 < Tu < 6$ per cent in Fig. 21, it is easy to see how a linear relation might be assumed.

The stagnation point and overall heat-transfer data are listed in Table 3. Figure 22 shows the stagnation point data displayed as suggested by [33]. The solid line represents the least squares curve fit given by

$$Nu/Re^{\frac{1}{2}} = 1{\cdot}010 + 2{\cdot}624\left[\frac{TuRe^{\frac{1}{2}}}{100} - 3{\cdot}070\frac{TuRe^{\frac{1}{2}}}{100}\right]^2 \qquad (7)$$

in the range $0 < TuRe^{\frac{1}{2}} < 64$ where Tu is an absolute

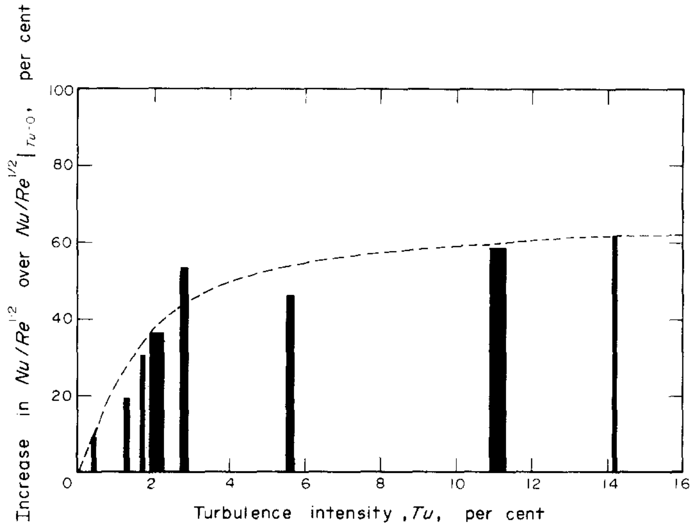

FIG. 16. Overall effect of turbulence intensity on the rate of heat transfer through the laminar boundary layer.

Table 2. Effect of turbulence on heat transfer through the laminar boundary layer

	Range of turbulence intensity (%)								
	0·40 to 0·47	1·23 to 1·28	1·30 to 1·37	1·65 to 1·70	1·90 to 2·23	2·77 to 2·85	5·57 to 5·68	10·88 to 11·28	14·15 to 14·20
Re	Increase in $Nu/Re^{\frac{1}{2}}$ over theoretical zero turbulence value of Frössling [30] (%)								
109×10^3							44·9	58·3	
114		19·4					47·0	58·3	
142									60·7
151	6·6	19·6			32·8				
157	8·1			30·3				58·1	
162	10·2								66·2
194	13·8		21·9	26·0	35·0	52·7		58·1	
203									56·4
251	6·8		17·7	31·0		60·7			
300			19·0	34·0	41·1	48·0			
Average % increase	9·1	19·5	19·5	30·3	36·3	53·8	46·0	58·2	61·1

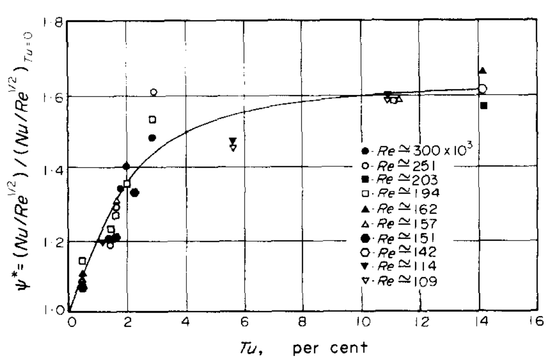

FIG. 17. The augmentation factor ψ^* as a function of turbulence intensity.

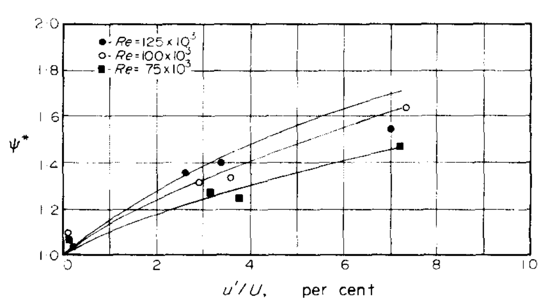

FIG. 18. The augmentation factor ψ^* of Kestin and Wood as a function of turbulence intensity [40].

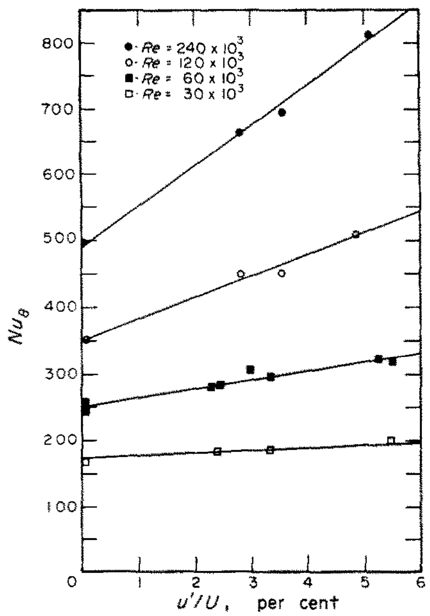

FIG. 19. Variation of the stagnation point Nusselt number turbulence intensity, $0 < Tu < 6\%$ [33].

fraction. The standard deviation of the fit is 0·0554. The datum point at $TuRe^{\frac{1}{2}} = 14{\cdot}2$ was rejected by applying Chauvenet's criterion [51]. The maximum deviation of any data point from the curve is 10·5 per cent which is within the external error estimate of 11·8 per cent [53]. Eighty-seven and a half per cent of the data points are within 6·1 per cent of the curve. The curve was not forced to go through $Nu/Re^{\frac{1}{2}} = 0{\cdot}945$ [30] or $Nu/Re^{\frac{1}{2}} = 1{\cdot}00$ [44] at $TuRe^{\frac{1}{2}} = 0$. Kestin and Wood [40] have concluded that it is doubtful that the limit $Nu/Re^{\frac{1}{2}} = 0{\cdot}945$ can be obtained in the laboratory and they suggest the procedure used herein. The curve fitted equation for the data of this study and the results of [33] and [40] are shown in Fig. 23.

Table 4 shows the effect of turbulence intensity on the overall or average Nusselt number. Increases and decreases in heat transfer are observed and it is seen that the largest increases occur for small increases in turbulence intensity at the lower values of turbulence intensity. More data should be obtained in the intermediate range of intensities before a generalization can be made.

Since it was concluded that increases in the heat-transfer rate through the laminar boundary layer

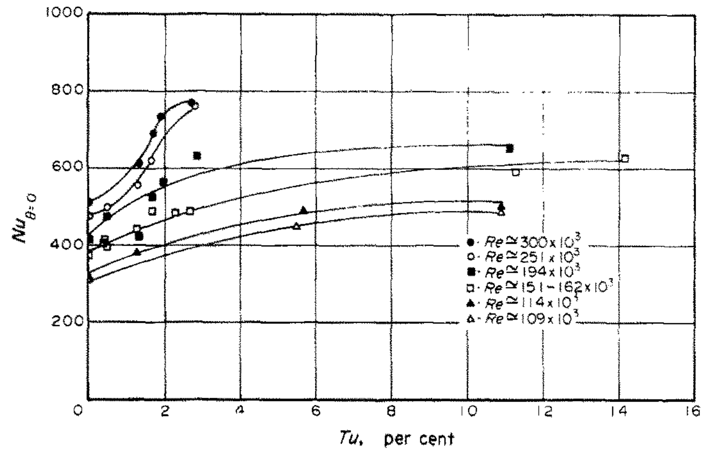

FIG. 20. Variation of the stagnation point Nusselt number with turbulence intensity, $0 < Tu < 16\%$.

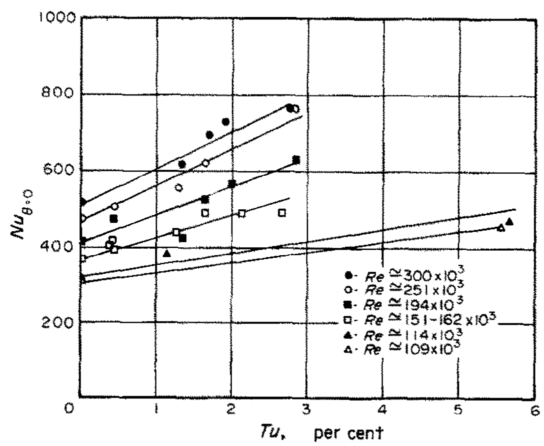

FIG. 21. Variation of the stagnation point Nusselt number with turbulence intensity, $0 < Tu < 6\%$.

always occurred for increased turbulence intensity, local plots of the variation of heat transfer with angular position were made to determine where the decreases in overall heat-transfer rate occurred. A typical plot is shown in Fig. 24. It is seen that for a change in turbulence intensity from 1·34 to 2·77 per cent at an approximately constant Reynolds number of 300000, the net change in overall Nusselt number was a decrease of 2·6 per cent. The figure shows increased local heat-transfer rates were obtained over the forward portion of the cylinder (near the stagnation point), but large decreases were measured toward the rear of the cylinder. The increases in heat-transfer rates over the laminar boundary layer are more than offset by decreases in the separated regions. Thus in addition to the need for a better understanding of the mechanisms by which increased turbulence intensity affects

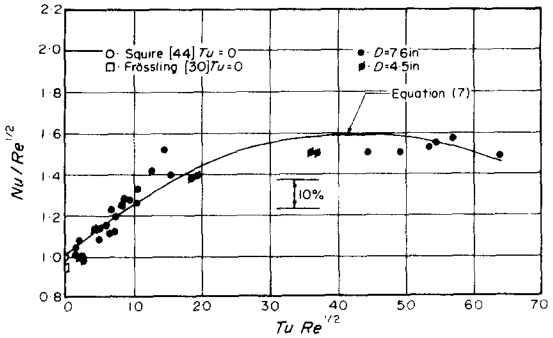

FIG. 22. Correlation of the heat-transfer data for the stagnation line of cylinders in cross flow.

laminar heat-transfer rates, more knowledge is required on the effect of turbulence intensity on the flow field in the transitional and separated flow regions.

The overall Nusselt number, $\overline{Nu}$, vs Reynolds number is seen in Fig. 25. Also, shown is Hilpert's [8] experimentally determined curve. Hilpert's curve corresponds closely with the data of this study taken at turbulence intensities between 1·23 and 1·37 per cent. The data for turbulence intensities above approximately 1·37 per cent show increases and decreases and display no systematic behavior with turbulence intensity or Reynolds number. These results are indicative of flows in which transition and separation occur as discussed previously. The data for turbulence intensity from 0·40 to 0·60 per cent display an approximate linear relationship between overall Nusselt number and Reynolds number and are indicative of subcritical flow.

Attempts were made to correlate the overall heat-transfer data as a function of turbulence intensity, although the objective of the study was to determine experimentally the effects of turbulence on local heat transfer. The most successful correlation obtained is shown in Fig. 26 and the data display an approximate linear relationship between $\overline{Nu}/Re^{\frac{1}{2}}$ and $TuRe^{\frac{1}{2}}$ for values of $TuRe^{\frac{1}{2}} < 10{\cdot}5$, but deviate substantially for

Table 3. Summary of stagnation point and overall heat-transfer data

Test No.	Grid condition and location (ft)	Scale (in)	Tu (%)	$Nu_{\theta=0}$	$\overline{Nu}$	Re	$TuRe^{\frac{1}{2}}$	$\frac{Nu_{\theta=0}}{Re^{\frac{1}{2}}}$
7·6 in dia cylinder								
1.1	Damping screens 1 and 2	0·403	0·40	406	293	157813	1·6	1·022
1.2	Damping screens 1 and 2	0·403	0·41	419	305	161877	1·7	1·041
1.3	Damping screens 1 and 2	0·403	0·44	477	364	197127	2·0	1·075
1.4	Damping screens 1 and 2	0·403	0·47	505	403	250780	2·3	1·009
2.1	Wire grid at 4.9	0·226	1·31	424	313	154358	5·1	1·080
2.2	Wire grid at 4.9	0·226	1·37	506	419	192672	6·0	1·152
2.3	Wire grid at 4.9	0·226	1·30	557	493	251220	6·5	1·112
2.4	Wire grid at 4.9	0·226	1·34	617	592	300399	7·3	1·125
3.1	Wire grid at 3.4	0·204	1·68	488	419	156993	6·7	1·231
3.2	Wire grid at 3.4	0·204	1·65	523	457	192783	7·2	1·191
3.3	Wire grid at 3.4	0·204	1·65	620	554	251282	8·3	1·238
3.4	Wire grid at 3.4	0·204	1·70	693	581	299961	9·3	1·266
4.1	Wire grid at 2.9	0·159	1·99	563	465	194595	8·8	1·276
4.2	Wire grid at 2.9	0·159	1·90	733	633	302290	10·5	1·333
5.1	Wire grid at 1·9	0·113	2·68	487	386	151300	10·4	1·252
5.2	Wire grid at 1.9	0·113	2·85	636	461	194279	12·6	1·443
5.3	Wire grid at 1.9	0·113	2·82	762	519	251840	14·2	1·519
5.4	Wire grid at 1.9	0·113	2·77	766	570	299271	15·1	1·399
6.1	Wooden grid at 2.4	0·363	11·28	591	396	156442	44·6	1·494
6.2	Wooden grid at 2.4	0·363	11·19	659	459	194562	49·3	1·494
6.3	Wooden grid at 2.4	0·363	11·74	719	505	215810	54·6	1·547
7.1	Wooden grid at 1.9	0·325	14·15	574	345	142732	53·5	1·519
7.2	Wooden grid at 1.9	0·325	14·19	631	384	161478	57·0	1·571
7.3	Wooden grid at 1.9	0·325	14·20	666	412	203135	64·0	1·478
4·5 in dia cylinder								
8.1	Damping screens 1 and 2	0·403	0·46	392	294	151431	1·8	1·007
8.2	Damping screens 1 and 2	0·403	0·60	409	318	173833	2·5	0·981
9.1	Wire grid at 4.9	0·226	1·23	381	271	114366	4·1	1·128
9.2	Wire grid at 4.9	0·226	1·28	440	350	151631	5·0	1·130
10.1	Wire grid at 2.4	0·137	2·23	487	436	150938	8·6	1·255
11.1	Wooden grid at 4.9	0·426	5·57	452	366	109107	18·4	1·369
11.2	Wooden grid at 4.9	0·426	5·68	470	371	114308	19·2	1·389
12.1	Wooden grid at 2.4	0·363	10·89	494	367	109208	36·0	1·496
12.2	Wooden grid at 2.4	0·363	10·88	506	375	114518	36·8	1·496

Table 4. Effect of turbulence intensity on overall or average Nusselt number

Re	Turbulence intensity Tu (%) Lowest	Highest	Average Nusselt number $\overline{Nu}$ Lowest	Highest	Increase or decrease in $\overline{Nu}$ (%)
109×10^3	5·57	10·89	366	367	+0·3
114×10^3	1·23	5·68	271	371	+36·9
	5·68	10·88	371	375	+1·1
	1·23	10·88	271	375	+38·4
151×10^3	0·45	1·28	294	350	+19·0
	1·28	2·23	350	436	+24·6
	0·45	2·23	294	436	+48·3
157×10^3	0·40	1·68	293	419	+43·0
	1·68	11·28	419	396	−5·8
	0·40	11·28	293	396	+35·2
162×10^3	0·41	14·19	305	384	+25·9
194×10^3	0·44	1·37	364	419	+15·1
	1·37	1·65	419	457	+9·1
	1·65	1·99	457	465	+1·8
	1·99	2·85	465	461	−0·9
194×10^3	2·85	11·19	461	459	−0·4
	0·44	11·19	364	459	+26·1
251×10^3	0·47	1·30	403	493	+22·3
	1·30	1·65	493	554	+12·4
	1·65	2·82	554	519	−6·7
	0·47	2·82	403	519	+28·8
300×10^3	1·34	1·70	592	581	−1·9
	1·70	1·90	581	633	+9·0
	1·90	2·77	633	577	−9·7
	1·34	2·77	592	577	−2·6

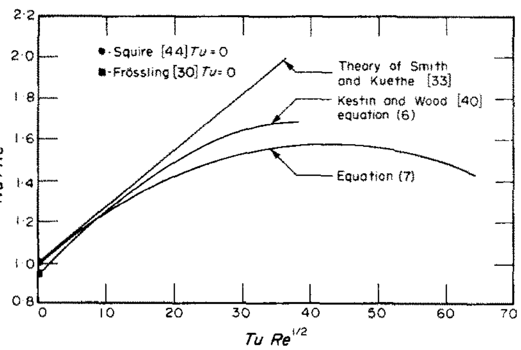

FIG. 23. Comparison of correlation equation with the theory of Smith and Kuethe [33] and with the correlation of Kestin and Wood [40].

values of $TuRe^{1/2} < 10·5$ with considerable data scatter at the higher values of $TuRe^{1/2}$.

A least squares fit of the data for $TuRe^{1/2} < 10·5$ yielded the equation

$$Nu/Re^{1/2} = 0·686+0·043\,TuRe^{1/2} \tag{8}$$

shown by the solid line in Fig. 26. Equation (8) has a standard deviation of 0·0582 and the correlation is definitely limited to the range, $0 < TuRe^{1/2} < 10·5$. At values of $TuRe^{1/2}$ approaching 10·5 and higher, the correlation parameter $\overline{Nu}/Re^{1/2}$ deviates substantially from the linear relationship. Because of the unsystematic behavior of the parameter $\overline{Nu}/Re^{1/2}$ at values of $TuRe^{1/2} > 10$, no definite conclusions can be made on the behavior of the correlation function at larger values of $TuRe^{1/2}$. This clearly points out the need for more

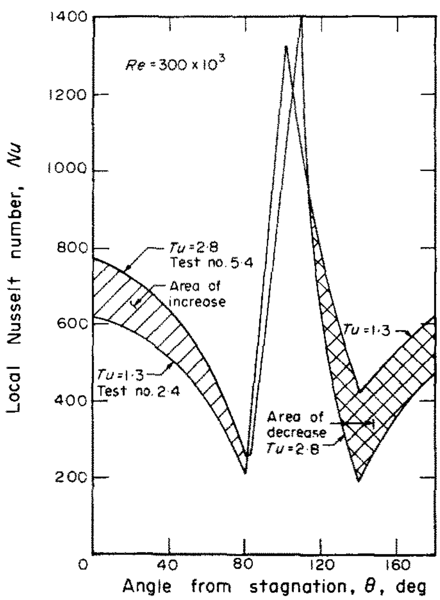

FIG. 24. Comparison of the local variation of heat transfer around a cylinder for test numbers 2·4 and 5·4.

detailed data on the effect of turbulence intensity on overall heat-transfer rates and on the effect of turbulence intensity on the flow field for higher Reynolds number flows. A dashed line is drawn on Fig. 26 to show a possible behavior of the correlation parameter $Nu/Re^{1/2}$ at values of $Tu/Re^{1/2} > 10$, but is not intended to suggest the actual behavior.

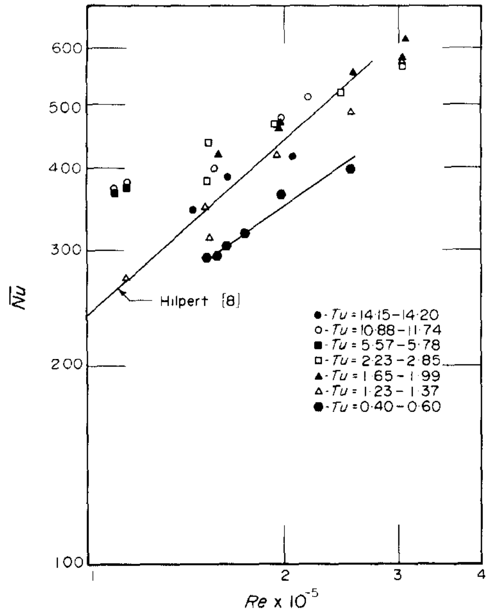

FIG. 25. Effect of Reynolds number on overall heat transfer at various turbulence intensities.

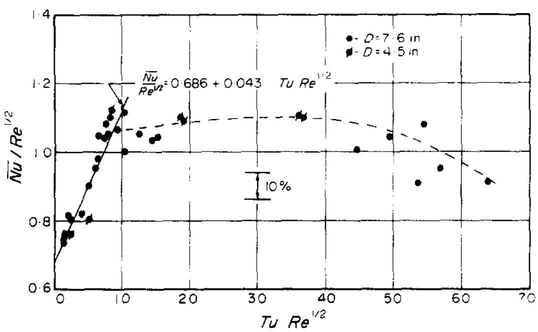

FIG. 26. Effect of turbulence intensity on overall heat transfer.

SUMMARY

Specifically, it can be stated from this study that:

1. In the laminar boundary layer, small increases in turbulence intensity (0·4–1·2 per cent) markedly increased the local rate of heat transfer. Increases in heat transfer over the theoretical zero turbulence value were on the order of 20 per cent. Increases in heat transfer over the zero turbulence value up to approximately 60 per cent were obtained at the highest values of turbulence intensity.

2. A Reynolds number effect was not discernible when the data was plotted as $Nu/Re^{\frac{1}{2}}$ vs some parameter.

3. In the laminar boundary-layer range, $0 < \theta < 40$ degrees, the effect of increased turbulence intensity is to always increase the local rates by a constant factor, ψ^*, which depends on the turbulence intensity and appears to be independent of Reynolds number over the range of this investigation. The augmentation factor, ψ^* seems to approach a value of approximately 1·6 for values of turbulence intensity around 14 per cent. It is recommended that more data be taken at $Tu > 14$ per cent to verify this observation.

4. The stagnation point data were successfully correlated against the parameter $TuRe^{\frac{1}{2}}$ as suggested by [33]. The correlation equation was obtained by a least squares curve fit of the experimental data and extends the range of existing correlations from a value of $TuRe^{\frac{1}{2}} = 40$ to 64. The data exhibit good agreement with the results of previous investigations for values of $TuRe^{\frac{1}{2}} < 20$, but deviate significantly for values of $TuRe^{\frac{1}{2}} > 20$. No data were found for values of $TuRe^{\frac{1}{2}} > 40$ and meager data were found in the range $20 < TuRe^{\frac{1}{2}} < 40$.

5. Both increases and decreases in overall heat transfer were obtained for increased turbulence intensity. In the majority of the cases, decreases in heat transfer were obtained with increased turbulence intensity above approximately 1·3 per cent. This indicates that decreases occurred in flows for which transition and separation occurred. Comparisons of the local variation of heat transfer around a cylinder at different turbulence intensities showed that large decreases in local rates of heat transfer occurred toward the rear of the cylinder indicating an effect on the flow and corresponding temperature field due to increased turbulence.

6. Partial success was obtained in correlating the overall rate of heat transfer against the parameter $TuRe^{\frac{1}{2}}$. The data exhibited a linear relationship for values of $TuRe^{\frac{1}{2}} < 10·5$, but showed no systematic behavior for values of $TuRe^{\frac{1}{2}} > 10·5$. It is suspected that the correlation holds only for subcritical flows; that is, for flows that are not of a transitional nature.

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INFLUENCE DE LA TURBULENCE SUR LE TRANSFERT THERMIQUE AUTOUR DE CYLINDRES CHAUFFES

Résumé—Des résultats ont été obtenus sur l'effet de la turbulence du courant extérieur sur le transfert de chaleur autour de cylindres chauffés placés normalement à un écoulement d'air. Les expériences ont été menées pour des nombres de Reynolds allant de 109 000 à 302 000 et des valeurs de l'intensité de turbulence allant de 0,40 à 14,20 pour cent. Les diamètres des cylindres utilisés étaient de 11,45 et de 19,3 cm et le rapport du diamètre du cylindre à la largeur de la section d'étude de 0,075 et 0,127 respectivement. Le rapport de l'échelle de turbulence au diamètre du cylindre se situe entre 0,015 et 0,095. L'échelle de la turbulence a varié de 0,29 à 1,08 cm au cours de l'étude. On a caractérisé la structure du courant extérieur libre et on a déterminé les effets de la turbulence produite artificiellement sur les transferts de chaleur local et global.

Les données sont discutées et comparées aux résultats d'études antérieures dans les domaines appropriés. Les résultats sont partiellement expliqués par la théorie disponible et font apparaitre des zones qui nécessitent une étude plus poussée. Une équation de corrélation est présentée qui permet de déterminer le transfert de chaleur au point d'arrêt en fonction de l'intensité de turbulence. L'augmentation du transfert de chaleur à travers la couche limite laminaire en fonction de l'intensité de turbulence est aussi déterminée. Les tentatives de détermination de l'effet de l'intensité de turbulence sur les transferts de chaleur global ou moyen ont partiellement réussi. Une équation de corrélation est présentée qui permet de prévoir le transfert de chaleur global en fonction de l'intensité de turbulence dans un domaine étroit de variation des paramètres.

DIE WIRKUNG DER TURBULENZ AUF DEN WÄRMEÜBERGANG VON BEHEIZTEN ZYLINDERN

Zusammenfassung—Über die Wirkung der Turbulenz der freien Strömung auf den Wärmeübergang von beheizten Zylindern, die senkrecht in einem Luftstrom angeordnet sind, werden Versuchsergebnisse gewonnen. Die Versuche werden für einen Bereich der Reynolds-Zahl von 109 000 bis 302 000 bei Werten für den Turbulenzgrad von 0,40 bis 14,20% durchgeführt. Die Durchmesser der benutzten Versuchszylinder betrugen 4,5 bzw. 7,6 in, und das Verhältnis von Zylinderdurchmesser zur Breite des Versuchsbereichs war 0,075 und 0,127. Das Verhältnis von Turbulenzgrad zu Zylinderdurchmesser reichte von 0,015 bis 0,095. Der Turbulenzgrad änderte sich von 0,113 bis 0,426 in während der Untersuchungen. Die Struktur der freien Strömung wird gekennzeichnet, und die Wirkungen der künstlich hervorgerufenen Turbulenz sowohl auf den örtlichen als auch den gesamten Wärmeübergang werden bestimmt. Die Versuchsdaten werden diskutiert und mit den Ergebnissen früherer Untersuchungen, soweit sie zugänglich waren, verglichen. Die Ergebnisse werden teilweise durch gültige Theorien erklärt und zeigen auch Gebiete auf, die noch weitere Untersuchungen erfordern. Es wird eine Bestimmungsgleichung angegeben, die es ermöglicht, den Wärmeübergang im Staupunkt als eine Funktion des Turbulenzgrads zu bestimmen. Die Zunahme des Wärmetransports durch die laminare Grenzschicht als eine Funktion des Turbulenzgrads wird ebenfalls bestimmt. Teilweise erfolgreich waren Versuche, die Wirkung des Turbulenzgrads auf den gesamten oder mittleren Wärmeübergang zu bestimmen. Es wird eine Bestimmungsgleichung angegeben, die es erlaubt, den gesamten Wärmeübergang als eine Funktion des Turbulenzgrads über einen engen Bereich der bestimmenden Parameter vorauszusagen.

ВЛИЯНИЕ ТУРБУЛЕНТНОСТИ НА ТЕПЛОПЕРЕНОС ОТ НАГРЕТЫХ ЦИЛИНДРОВ

Аннотация — Получены данные о влиянии турбулентности свободного потока на теплоперенос от нагретых цилиндров, перпендикулярных потоку воздуха. Эксперименты проводились при числах Рейнольдса, изменяющихся от 109 000 до 302 000 при интенсивности турбулентности от 0,40 до 14,20%. Диаметры рабочих цилиндров составляли 4,5 и 7,6 дюйма, соответственно, а отношение диаметра цилиндра к ширине экспериментального участка — 0,075 и 0,127. Отношение масштаба турбулентности к диаметру цилиндра изменялось от 0,015 до 0,095. Масштаб турбулентности во время исследований изменялся от 0,113 до 0,426 дюйма. Описана структура свободного потока и определено влияние искусственно индуцированной турбулентности как на локальный, так и полный теплоперенос.

Полученные данные обсуждаются и сравниваются с результатами предыдущих исследований. Дается частичное объяснение этих результатов с помощью элементарной теории и указывается также на области, требующие дальнейшего исследования. Представлено обобщающее уравнение, которое позволяет определить теплоперенос в критической точке в зависимости от интенсивности турбулентности. Определяется также увеличение теплопереноса через ламинарный пограничный слой в зависимости от интенсивности турбулентности. Попытки определить эффект интенсивности турбулентности на полный или средний теплоперенос частично увенчались успехом. Представлено обобщающее уравнение для расчета полного теплопереноса в зависимости от интенсивности турбулентности в узком диапазоне изменения рассматриваемых параметров.