THE EFFECT OF TURBULENCE ON LOCAL HEAT TRANSFER COEFFICIENTS AROUND A CYLINDER NORMAL TO AN AIR STREAM

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THE EFFECT OF TURBULENCE ON LOCAL HEAT TRANSFER
COEFFICIENTS AROUND A CYLINDER NORMAL TO AN AIR STREAM

INTRODUCTION

The effect of turbulence on the local heat transfer coefficient
distribution about a cylinder normal to an air stream at three
different Reynolds numbers is presented in this thesis. This is
a continuation of the work done on this subject by W. M. Bollen (1).

A number of investigators have determined the local heat
transfer coefficient distribution about a cylinder at undetermined
turbulence intensities. Several have shown that the intensity of
turbulence markedly affects the overall heat transfer coefficients.
However, to our knowledge, there has been no work published on
the effect of turbulence on the local coefficients.

Local heat transfer coefficients, expressed as dimensionless
Nusselt numbers, \( hD/k \), were determined by measuring the conden-
sation rate of steam in a thermally-isolated segment subtending an
angle of 16.8° on the circumference of a standard two-inch copper
pipe. The pipe was maintained at a constant temperature by the
passage of steam through it. The segment could be exposed to
different positions relative to the direction of the airflow
normal to it. This was accomplished by rotating the experimental
cylinder about its vertical axis in a wind tunnel.

The range of data covered in this investigation were obtained
by using three turbulence levels at each of three different
Reynolds numbers. The three turbulence levels used in this in-
vestigation were 0.9%, 3.0% and 11.5%. The first level was the
turbulence existing in the tunnel in the absence of any disturbing
members. The last two levels were obtained by placing two
different turbulence promoting grids eleven inches upstream from the experimental tube. The three Reynolds numbers used were 39,000, 71,500 and 110,000, corresponding to velocities of 33.1, 60.3 and 93.5 feet per second.

It should be emphasized here that the results of this investigation, as presented in the following paragraphs, are valid only in the range of Reynolds numbers covered, i.e., 39,000 to 110,000.

The distribution of heat transfer about the circumference of a cylinder at 0.9% turbulence has a maximum at the forward stagnation point, 0°, and then decreases to a sharp minimum at about 83° which is less than half of the value at 0°. The heat transfer then rises to a second maximum at 180° which is about 25% higher than that at 0°. With an increase of turbulence to the 3% level, the general trend in local coefficients is the same as at the 0.9% level except for a new maximum at about 110° which gradually develops with increasing velocity. At 11.5% turbulence, the shape of the distribution curves is similar to the high velocity curve at the 3% level. The new maximum, at 113°, is prominent at all velocities, becoming sharper as the Reynolds number is increased; it is approximately the same magnitude as the maximum at 0°. The minimum following this new maximum has about the same value as the first minimum, both of which are less than half of the value at 0°.

At each turbulence level used, the effect of increasing the Reynolds number was to increase uniformly the local heat transfer coefficients. Within the range of Reynolds numbers covered in this experiment, the local heat transfer generally increased considerably with the increase in turbulence. Another effect of increasing the
intensity of turbulence was a lowering of the critical Reynolds number, thus permitting the abnormal behavior occurring in the critical range to take place at a much lower Reynolds number.

A short historical introduction is offered as a background for the experimental work. This is followed by a discussion of the theories of heat transfer, fluid flow about a cylinder, the nature of turbulence, and the control of the intensity of turbulence in a wind tunnel. After an experimental section which includes a detailed description of the equipment, operating procedure and sample calculations, the results of this research are presented and discussed. An extensive bibliography is also given.
HISTORICAL INTRODUCTION

The investigation of local heat transfer coefficient distribution about a cylinder by several authors is discussed in this section in chronological order. Following this several references concerning the effect of turbulence on overall coefficients is given. No published work was found on the effect of turbulence on local transfer coefficients, although there was some preliminary work carried out on this subject by Bollen (1).

Lohrisch (4), as early as 1926, studied the effect of fluid velocity on the distribution around a cylinder of the rate of mass transfer. He converted his local mass transfer data into heat transfer data by means of the Chilton-Colburn analogy. Figure 9 shows the results obtained by Lohrisch and a number of other investigators.

A paper by Fage and Falkner (12) gives a mathematical theory of the rate of heat transfer from a surface of a cylinder over which the fluid flow in the boundary layer is laminar and two-dimensional when the heat flow is steady. "A general differential equation for heat transfer was obtained from a consideration of the heat balance at a point in the boundary layer; and after simplification, this equation was solved by artifices similar to those used to obtain a solution of the boundary layer equations." (12, p172) These theoretical relationships were compared with those obtained from measurements of the heat transfer from a nickel strip embedded in the surface of a cylinder and insulated from it. The agreement
between theory and experiment was good.

Drew and Ryan (4) in 1931, obtained actual heat transfer data in a manner somewhat resembling the work done for this thesis by condensing steam inside a tube sectioned with fins. The condensate from each section was collected and measured; from this the local heat transfer coefficients were calculated. The results obtained, which are similar to those of Lohrisch, are also shown in figure 9.

Small (23) measured local temperatures in a tube wall with a thermopile, from which he calculated the local heat flux and heat transfer coefficients.

By far the most carefully executed and extensive experiments on local heat transfer coefficients about a cylinder were carried out by Schmidt and Wenner (19) in 1940. Their experimental tube consisted of a brass pipe which was internally heated by steam. A strip of the surface was removed and replaced by a hollow copper bar heated internally by an electrical heating element. The surface of the bar, which was shaped so as to make the assembled experimental tube cylindrical, and the cylinder circumference were insulated from each other, thus forming a completely isolated heating element. The temperature of the element was maintained the same as the rest of the tube; from the power input required, the heat transfer coefficients were computed. The range of Reynolds numbers covered was from 8,000 to 426,000. Several of their curves are reproduced in figures 4, 5, and 6; their results in the ori-
tical range are used in the discussion of the results of this thesis.

In 1948, Winding and Cheney reported the results of their study on the effect of Reynolds number on local transfer coefficients. They observed the deformation, by sublimation, of cast naphthaline cylinders placed normal to an air stream. The resulting mass transfer data was converted to heat transfer data by Colburn's "j" factor.

The effect of turbulence on the overall heat transfer coefficient has been studied by Reiher who reported increases in heat flux of 50 to 100%, but he did not measure the turbulence levels used. Comings, Clepp and Taylor (3) in 1948, reported both the overall transfer coefficients for a cylinder normal to an air stream and the corresponding intensity of turbulence of the air stream. Their work stimulated an interest in the effect of turbulence on local heat transfer coefficients. This research project was undertaken to determine just what the effect of different turbulence levels was.
THEORETICAL CONSIDERATIONS

The problem of this thesis embody both heat transmission and fluid flow. They are discussed in this section under the general headings of heat transmissions, fluid flow, a correlation of these two, and turbulence. The first section briefly considers the material necessary to arrive at a formula for calculating heat transfer coefficients. The fluid flow section deals mainly with the theory of a boundary layer about a cylinder and how this layer behaves. The following section attempts to show the effect of the relationship between these two considerations and their bearing upon the subject of this research. The last section gives the definition of turbulence and how the intensity of turbulence was measured and controlled in this research.

HEAT TRANSMISSION

Considering the transfer of heat from steam condensing inside a cylinder to a fluid flowing outside, there are three resistances affecting the rate of heat transfer. These resistances are the steam film, the tube wall, and the boundary layer formed around the outside of the cylinder by the fluid flowing normal to it. Since they are in series, the reciprocal of the overall heat transfer coefficient, \( U \), is the sum of these resistances.

\[
\frac{1}{U} = \frac{1}{h_s} + \frac{1}{h} + \frac{1}{h_{b.l}}.
\]

Since the resistance of the steam film and of the tube wall are so small compared to that of the boundary layer, the overall
resistance may be assumed equal to the latter. Thus \( h_{b,l} \) at any point becomes equal to the ratio of the heat flux across the wall at that point to the total driving force between the condensing steam and the ambient air.

\[
h = \frac{q_{c}}{A} \frac{A}{\Delta t}
\]

There is also some heat transferred by radiation.

\[
gr = 0.173 A \epsilon \left[ \left( \frac{t_{h}}{100} \right)^{4} + \left( \frac{t_{w}}{100} \right)^{4} \right]
\]

The total heat transferred is the sum of the two; however the contribution of radiation in this research was very small.

**FLUID FLOW**

Prandtl's postulate, that the viscosity of a fluid plays an important role only in a thin "boundary layer" on the surface of a body in the moving fluid, has been a key to the understanding of many puzzling phenomena occurring in the flow of viscous fluids. Subsequent works, both experimental and theoretical, have verified the existence of a boundary layer and have given conclusive evidence that the radical difference between the behavior of an ideal fluid and that of a viscous fluid is due to the presence of this fluid film and the wake resulting from its separation from the surface of the body.

The thickness of this boundary layer is quite thin; its outer edge usually being defined as that point where its velocity is the same as that of the main stream. The velocity gradient within the
boundary layer is quite steep, varying from zero velocity at the surface of the immersed body to the main stream velocity at the outer edge. For laminar layers, the velocity distribution is approximately parabolic. The presence of turbulence in the boundary layer considerably modifies the velocity distribution. In other words, the resulting process of mixing tends to make the velocity more uniform throughout the greater part of the layer, yet at the same time produces a very rapid change in velocity near the wall. If the surface is very smooth, laminar motion will still persist over a small zone known as the laminar sublayer even though there is turbulent flow in the main boundary layer. In the turbulent part of the layer the velocity distribution is approximately logarithmic. Another interesting fact concerning velocity distribution in the boundary layer is that the temperature field and the velocity field in the fluid flowing along a surface are almost identical.

On cylinders all boundary layers begin at the stagnation point and remain laminar for a certain distance downstream, whatever may be the character of the exterior flow on the surface. If the prevailing velocity is sufficiently high and the surface sufficiently large, however, the layer will become turbulent. The chief characteristic which distinguishes the turbulent flow from the laminar is its much greater ability to transport momentum laterally, and hence, also heat.

When the pressure outside the boundary layer increases in the
downstream direction, the layer thickens rapidly. The particles within the boundary layer are retarded by the surface friction and the adverse pressure gradient, but are forced ahead by their own kinetic energy and the momentum transferred to them by the friction of the neighboring, faster moving particles. As the layer thickens, the retarding influence becomes predominant, the particles lose energy and finally reversal of the flow occurs. The reversal of flow, on account of the consequent accumulation of fluid, separates the flow from the surface. This happens even when the flow in the boundary layer is laminar. "At low values of the Reynolds number of a cylinder, the flow in the boundary layer is laminar and separation takes place in a manner governed by the laws of laminar flow, namely at a definite location dependent on the shape but not on the Reynolds number". (6, p. 72) Visual observations with smoke show that the air motion in the wake fluctuates greatly and that only near the separation point is thin reversed flow at all regular.

Experiments by Fage and Falkner (10, 11) showed that the maximum negative pressure normal to the surface of a cylinder at Reynolds numbers below the critical value occurred at about 70 degrees from the stagnation point. The point of separation was then indicated between 80 and 85 degrees for all Reynolds numbers below the critical value by a leveling off of the normal pressure curves. This observation of the separation point was also indicated by their surface friction measurements. Their experiments indicated that below the critical Reynolds number the boundary layer left the surface before
transition to turbulent flow occurred. Consequently, they reasoned, the flow characteristics, with the exception of surface friction are not sensibly affected by a change in the value of the Reynolds number.

There is a particular range of Reynolds numbers, from about 150,000 to 300,000, in which the character of the flow around a circular cylinder undergoes a marked change. With increasing Reynolds numbers in this critical range, Fage and Balkner noted that the point of maximum negative normal pressure moved further around the cylinder. They explained that, in this range, a transition from laminar to turbulent flow, instead of separation, occurred about 10 degrees beyond this point of maximum negative pressure. The local conditions were such that no abrupt separation of the boundary layer from the surface took place. In fact, they claimed, the turbulent layer clings to the surface for an appreciable distance around the back of the cylinder, but eventually the retarding influence of the opposing pressure led to a complete separation of the layer from the surface and the formation of a stagnant region at the back of the cylinder.

Schmidt and Wenner (19, p. 16) noted that their lowest local heat transfer coefficients in the critical region of flow occurred at about the same position as the minimum in Fage and Balkner's frictional intensity curves. The second maximum in Schmidt and Wenner's heat transfer curves at 115 degrees also corresponded to a second maximum at this position in Fage and Balkner's frictional
intensity curves. They agreed with Fage and Falknor's explanation of the flow characteristics in this region and added that through the turbulent interchange of momentum, the boundary layer behind the minimum position of the frictional intensity curves is nearly accelerated. Not until the second maximum in these curves is reached does the orderly boundary layer flow cease and the layer separate.

Although the explanation of the behavior of the boundary layer in the critical range of flow by the four authors above seems quite logical, a somewhat different explanation has been advanced and backed up experimentally which agrees with the results of this research much better. This theory will be discussed in the following paragraphs.

Hinke (14) showed that a laminar boundary layer could exist as a free laminar layer following separation and that, as the free layer thickened, transition to turbulent flow finally occurred. This transition was affected by the turbulence of the air stream.

The existence of a free laminar boundary layer was also proved by Schubauer (20) in 1939. To do this he made velocity explorations in the boundary layer with a very sensitive hot wire anemometer and made photographs of kerosene and lampblack flow patterns on an elliptic cylinder. Flow patterns were also indicated by litmus solution dried on the surface of the cylinder followed by the release of a small amount of ammonia gas at the surface through a hypodermic needle. His findings will be used in the Results and
Discussion section to explain the shape of the local heat transfer curves in the critical range of Reynolds numbers.

A double separation of the boundary layer was indicated by Schubaer's photographs of flow patterns above a certain critical Reynolds number. This double separation was a characteristic of the flow in the critical region and always occurred at the same points on the surface. The first separation was that of a nearly laminar layer and the second was that of a turbulent layer. A complete transition occurred in the free laminar layer between the two points and the resulting turbulent layer, due to the fanning out of the layer that always accompanies transition, re-attached itself to the surface. Below this critical range of Reynolds numbers, only one separation was detected. The latter agreed the results of Fage and Falkner (11).

**CORRELATION OF HEAT TRANSFER AND FLUID FLOW**

The heat transferred in any forced convection process involving a heating element in a stream of flowing fluid is dependent upon how much heat can be transferred through the boundary layer that is formed on the element. The boundary layer thus exerts a retarding influence on the heat transfer process. The magnitude of this retarding influence depends upon the thickness of the layer and its type of flow. The thicker the layer builds up, the larger the resistance becomes and, consequently, the lower the heat flux. In laminar flow, momentum is transferred from the exterior stream by molecular exchange only; in like manner, heat is transferred to
the main stream by molecular exchange only. In order for the heat to get from its source to the main airstream it must pass through the laminar layer by molecular conduction, hence the transfer process in such flow is low. In turbulent flow, momentum and heat are transferred by relatively intense eddy motion. This enables the heat picked up at the surface of the heating element by conduction to be transferred through the boundary layer at a much faster rate than the molecular diffusional process occurring in the laminar flow.

Another fact to be remembered about the boundary layer is that as it proceeds around the hot tube from the stagnation point it becomes warmer. The actual temperature difference between the tube and the boundary layer on its surface is thus smaller than the apparent temperature difference between the tube and the ambient air as used in calculating the local heat transfer coefficients. Since a turbulent layer can dissipate its heat more readily than a laminar one, it will not have its temperature increased as much as a laminar layer would as it proceeds around the tube. Consequently, it should exhibit a larger actual driving force and heat flux.

**TURBULENCE**

Definition: Turbulence connotes fluctuation of velocity with time; the speed at any point fluctuating from instant to instant about a mean value. The comparatively slow fluctuations which can be detected by ordinary measuring instruments are not included in the concept of turbulence. The frequencies included are those above
a rather indeterminate limit of ten or more per second.

Turbulence is usually defined by both its scale and its intensity. Mathematically, the intensity of turbulence is defined as the square root of the mean square of the deviations from the mean velocity over a period of time at one point, divided by the mean velocity. That is, it is the ratio of the root mean square deviating velocity to the mean velocity. If the wave form were sinusoidal, a turbulence of 1% would mean fluctuations of \( \pm 1.4\% \) from the mean velocity.

The scale of turbulence is a measure of the relative size of the eddies in the flow. It is defined by \( L = \sqrt{\frac{\sum R_z^2}{\sum z}} \), where \( R_z \) is the coefficient of correlation between instantaneous values of two velocity fluctuations separated by the cross stream distance \( z \). Since the scale is not a determining factor in this experiment, it will not be further discussed.

Measurement: There are at present three methods of measuring the intensity of turbulence. The first method of comparing turbulence of different air streams is by the observation of the resistance of spheres. The most accurate, and actually the reference against which all other methods are compared, is a special form of the hot-wire anemometer developed by Dryden (16) of the U.S. Bureau of Standards. This equipment is cumbersome and expensive, hence the third method, which uses relatively inexpensive, portable equipment, was selected for use in this research. This hot-wire, wake-angle instrument was first developed by Schubauer (22). Figure 1 is a
sketch of the instrument probe as modified by J. F. Taylor of the University of Illinois and constructed at Oregon State College by R. C. Meng. This probe was mounted in a ball bearing support for rotating in the tunnel. Page 29 shows a photograph of the assembled instrument. Auxiliary equipment required is a potentiometer and a means of applying a small, steady current to the heating element.

The theory of this instrument as outlined below is taken directly from references 1 and 22.

It has been found that with increasing turbulence the width of a heated wake from a hot wire increases. For laminar, uniform flow, assuming the heated wire's diameter is so small as to produce a negligible disturbance and acts as a line source of heat, the temperature distribution in the wake at points not too close to the wire is given by the equation:

\[ \Delta T = \Delta T_{\text{max}} e^{-\frac{y^2 c V \rho}{4 k x}} \]

where

- \( y \) = distance across wake measured from center
- \( x \) = distance downstream from wire
- \( \Delta T \) = temperature rise at any point
- \( \Delta T_{\text{max}} \) = maximum temperature rise at the point \((x, 0)\)
- \( V \) = air speed past the wire
- \( \rho \) = density of air
- \( c \) = specific heat of air
- \( k \) = thermal conductivity of air
The finite size of the wire, the conduction of heat in the $x$ direction, the retardation of the flow by the wire and the stirring action due to whatever eddies are formed in the wake of the wire have been neglected.

If the stream is turbulent, there is an apparent conductivity, $(k + B_{\phi})$ where $B$ is the eddy conductivity due to the turbulence. Hence the above equation becomes:

$$\Delta T = \Delta T_{\text{max}} e^{\frac{-\gamma^2 eV\rho}{4(k + B)\alpha}}$$

The width of the heated wake at half the maximum temperature change is selected to represent the width characteristic of the wake. Hence:

$$\frac{\Delta T}{\Delta T_{\text{max}}} = \frac{1}{2} = e^{\frac{-\gamma^2 eV\rho}{4k\alpha}}$$

where $\gamma$ represents the half width, and $2\gamma$ then represents the width characteristic. Changing to angular measure and solving for $\alpha_{l}$, the total angle subtended at the wire by the wake width at

$$\frac{\Delta T}{\Delta T_{\text{max}}} = \frac{1}{2} \text{ for laminar flow;}$$

$$\alpha_{l} = 190.8 \sqrt{\frac{k}{\rho eV \alpha}}$$

Similarly, for turbulent flow:

$$\alpha = 190.8 \sqrt{\frac{k + B}{\rho eV \alpha}}$$

where $\alpha$ is the angular width for turbulent flow.

$$\alpha^2 = \frac{36,400 k}{\rho eV \alpha} + \frac{36,400 B}{\rho eV \alpha} = \alpha_{l}^2 + \alpha_{t}^2$$
where \( \alpha^2_{\text{ct}} \) is the angular width due to turbulence alone. Hence:

\[
\alpha^2_{\text{ct}} = \alpha^2 - \alpha^2_o
\]

Since the turbulence level \( Z \) is proportional to

\[
Z = c\alpha^2_{\text{ct}}
\]

The constant, \( c \), was found by Schubauer (22, p. 4) to equal \( \frac{1}{1.53} \).

Therefore:

\[
Z = \frac{\alpha^2_{\text{ct}}}{1.53}
\]

CONTROL OF TURBULENCE IN A WIND TUNNEL

The first problem involving turbulence when this research was undertaken concerned itself with reducing the turbulence level that was characteristic of the existing wind tunnel. After reading a number of papers by Dryden et al (5, 6, 7, 8, 18) it was decided that this could be accomplished by smoothing down the entrance to the working section and using damping screens. A summary of the effect of damping screens in reducing turbulence is now presented.

The effect of a damping screen in reducing the turbulence of an air stream is found to depend on the pressure drop coefficient, \( k_p \), of the screen. According to a theory which has been experimentally confirmed, the intensity of turbulence is reduced in the ratio \( 1/(1 + k)^{1/2} \) by a single screen, and in the ratio \( 1/(1 + k)^{2/n} \) by \( n \) identical screens in series. These relations show that for the same power expenditure a greater turbulence reduction can be obtained with a number of screens of small "\( k \)" than with a single screen of coefficient "\( n k \)". Factors that place a lower limit on velocity fluctuations are turbulence produced by the screens themselves and tunnel noise.
It may seem contradictory that screens are used in this experiment both to increase and to decrease the intensity of turbulence. A screen smooths an air stream to the extent of decreasing turbulent motions of larger scale than its mesh size, while at the same time introducing turbulent motions of smaller scale. The small-scale turbulence decays much more rapidly than the large-scale turbulence. Hence, at some distance from the screen, the overall effect is a smoothing of the airstream in which both the intensity and the scale of the turbulence are reduced. Damping screens differ from turbulence producing screens by being of finer mesh and by being located in the large cross section ahead of the entrance cone at a distance from the working section rather than in the working section itself.

According to Dryden, turbulence levels as low as 0.02 to 0.05% are obtainable by the use of damping screens. The price paid for the reduction of turbulence by a damping screen is a loss of energy equal to the pressure drop $p_1$ minus $p_2$ times the total volume of air moved through the screen.

In order to avoid irregularities in the average speed over the cross section of the stream behind a turbulence grid and to insure isotropic turbulence, one should not place the grid nearer to the experimental tube than 16 mesh lengths. In the investigations reported in reference 9, it was found that the regular pattern of maxima and minima in the mean speed caused by the wake of the individual wires disappeared at distances of about 12 mesh lengths.
for the screens used. Experiments also show that the turbulence is very nearly constant across the cross section in the region where the mean speed is constant.
EXPERIMENTAL

Apparatus design, operating procedure and sample calculations are the three main divisions of the experimental section. The first division describes the wind tunnel, the experimental tube, the velocity measuring equipment, and the turbulence indicator and generators. The last two divisions describe the routine followed in making an experimental run and sample calculations from the data obtained.

APPARATUS DESIGN

Tunnel: The wind tunnel consists of eight functional parts: (1) A motor and fan, (2) honeycomb, (3) initial expansion section, (4) damping screens section, (5) accelerator, (6) working section, (7) velocity measuring section, and (8) the final expansion section. The dimensions of these sections are shown in figure 2.

An American 550E blower capable of producing 13,500 C.F.M. at a static pressure of 6 inches of water gauge when running at 1,255 RPM was used to force the air through the tunnel. The blower was driven by a 30 horsepower, 1,750 RPM U.S. Autostart electric motor. Power was transmitted by a sheave and V-belt arrangement such that the speed of the blower was 1,243 RPM when 60 cycle current was supplied to the motor. Air velocity was controlled with a sliding damper on the fan inlet. *

The eight sections of the tunnel proper were constructed of one half inch Douglas fir plywood. The ends of each section were

* Specifications of the tunnel are taken from Bollen's thesis (1)
framed with 1\(\frac{1}{2}\) by 3 inch saeh stock which served as flanges for bolting the sections together. The first section, containing a honeycomb, was an exception in that it was bolted directly to the blower through the sides, but it was flanged for connection to the next tunnel section.

After leaving the blower, the air enters a honeycomb made of mailing tubes one inch in diameter and eighteen inches long. This removes large scale eddies induced by the blower. The air then enters the initial expansion section where its velocity is decreased to about one-fifth the blower exit velocity. It then passes through the damping screens section, which consists of two 30 mesh, 0.015 inch diameter wire screens placed 15 inches apart in a square section. The purpose of these two screens was to insure a uniform, low turbulence in the working section as explained in the Theoretical Considerations. The damping screens, together with a streamlining of the joints between the accelerator and working sections decreased the intensity of turbulence in the tunnel to 0.9%.

After the damping screens the air passes through an accelerating section to a 6 by 18 inch working section. This section also helps to provide a uniform, low turbulence flow because of the large area reduction at the entrance to the working section. Next to the working section, which contains the experimental tube, is the velocity measuring section. Between the two sections is placed a 23 mesh, 0.015 inch diameter wire screen to damp out the wake caused by the experimental cylinder in order that velocity measurements
might be taken. To reduce exit losses a final expansion section is added.

Photographs of the tunnel and related equipment are shown on pages 31 and 32.

Experimental Tube: A description of the construction of the experimental tube and several supporting sketches are given in this section. Also included are the method of mounting the tube in the tunnel, supplying it with dry, constant pressure steam, and collecting the condensate formed.

The experimental tube was fabricated from standard two-inch copper pipe, 2.06 inches inside diameter with a wall thickness of 0.158 inches. The tube was cut into three sections and so threaded that the three could be screwed together, thus yielding a pipe 14 inches long which could be disassembled to facilitate working inside the middle section. The three pieces act as a top and bottom guard section and a working section, their lengths being respectively 1 5/8, 5 1/2 and 7 inches.

In order to measure local heat transfer coefficients, it was desired to isolate a small segment of the surface of the working section and make a modified calorimeter out of it. This was accomplished in the following manner.

A strip of the surface four inches long having an arc of about 0.38 inches was removed from the middle of the working section; the edges of the cut were then beveled inwardly. From another section of two inch copper pipe a strip of the surface was machined out to
fit the rectangle removed from the working section. A uniform cut of 0.03 inch was then made along all four edges of this strip so that insulation could be placed between it and the working section when the strip was placed in the rectangular hole cut in the latter. This resulted in a smooth pipe which had a segment of its surface four inches long and 0.344 inches wide thermally insulated from the rest of the pipe. This segment will be referred to as the experimental segment.

To serve as a steam chest and condensate trap for the experimental segment, a seven inch long piece of one-half inch copper pipe was used. A vertical four inch cut was made in this pipe, the edges of the cut were laid back and brazed to the outer edges of the back of the experimental segment. The top of the pipe was closed except for a bleed line. To the bottom of the pipe was brazed a piece of one-fourth inch copper tubing to serve as a condensate line. A three-eighths inch steam line entered this pipe directly opposite and one-half inch below the center line of the experimental segment. This whole unit, composed of the experimental segment, its steam chest and condensate, bleed and steam lines is the modified calorimeter referred to before.

With the exception of the experimental segment the above unit was insulated by painting with plastic Tygon. Each of the four sides of the segment were covered with a strip of asbestos impregnated hard rubber gasket for insulation. The whole unit, with its insulation, was placed in the working section of the two inch copper
pipe with the experimental segment fitting snugly into the beveled opening in the pipe. It is held firmly in place by two Micarta blocks which are forced against it by two screws from the opposite wall of the working section.

The top of the upper guard section was closed except for a small bleed line by brazing a copper sheet to the top of the two inch pipe. The bottom of the lower guard section was closed by a plate which was bolted to it, thus permitting easy removal of the plate when desired. The plate had five holes drilled in it and five short pieces of copper tubing brazed over them. This permitted the bleed, condensate and steam lines of the experimental unit to pass through; the other two tubes were a bleed and condensate drain for the main part of the pipe. The lower guard section also had a hole tapped in its side about two inches below the top to receive a horizontal manometer leg.

After the three sections were screwed together, with the experimental segment in place and the bottom plate bolted on, the whole experimental pipe was placed in a lathe. The working section was turned down so that the whole section, especially at the junction of the segment and the circumference of the working section, would be smooth and cylindrical. Detailed drawings are shown in figures 3 and 4.

The tube was mounted in the working section of the tunnel so it could be readily rotated. It was supported by a clamp brazed to a circular plate on which was inscribed a mark indicating the
position of the center line of the experimental segment. This plate rested on another plate bolted to the top of the tunnel which was marked off in degrees. The 4 inch experimental segment was centered in the 6 by 18 inch working section, ten inches downstream from its forward edge.

Steam was passed through a preliminary centrifugal water separator, reduced from 70 psig to about two inches of water gauge by a reducing valve, and passed into a large steam chest which also acted as a centrifugal water separator. The bleed valve on this chest provided a finer adjustment on the steam pressure. Two steam lines emerged from the chest, one supplied the main part of the pipe with steam, the other supplied the steam chest on the experimental segment. The latter also acted as a superheater in that it was covered with glass cloth, wrapped with resistance wire connected to a Variac, and insulated. The Variac, by regulating the voltage, maintained a constant degree of superheat on the steam. The temperature of the steam was measured by a Copper-Constantan thermocouple placed at the exit of the superheating section.

The condensate from the experimental segment was drained out of the condensate line attached to the experimental unit by a simple sump collector sketched in Figure 5 and collected in a 25cc burette.

Velocity Measurements: The velocity of air in the wind tunnel was determined in the velocity measuring section by means of a pitot tube constructed according to A.S.T.M. specifications. The
velocity head measurements were indicated by a calibrated inclined manometer and a vertical manometer. A differential manometer was also used at times as a check. The screen at the entrance to this section was sufficient to damp out the wake caused by the experimental tube at the medium and high turbulence levels; but at the low turbulence level there was sufficient variation in the velocity to warrant a velocity traverse over the cross section. This traverse was integrated using an outline given by Linford (13). One of the traverses, including calculations and explanations, is included in the appendix.

The resulting velocity calculated by these measurements is that of the free stream, that is, it is the velocity which would be measured in the working section if the tube were not present.

Turbulence Indicator and Generators: The turbulence indicator used is essentially a hot wire wake angle instrument. It was first developed by Schubauer (22). Figure 1 is a sketch of the instrument probe which was mounted in a ball bearing support for rotating in the tunnel. Page 29 shows a photograph of the assembled instrument. The theory behind this instrument has been discussed in the Theoretical Considerations section; the calculations involved are shown in the Sample Calculations section. Auxiliary equipment used is a portable potentiometer and a 120 to 3 volt transformer attached to the Variac used with the superheater.

The high and medium turbulence levels were obtained by adding a dowel grid or a screen to disturb the air flow at the entrance
TURBULENCE INDICATOR PROBE

Micarta Rod
Lucite Bushing
Steel Construction

FIGURE 1
TURBULENCE MEASURING INSTRUMENT
FIGURE 2

WIND TUNNEL
View of wind tunnel looking downstream from the accelerating section.

View of wind tunnel looking upstream from the accelerating section.
View of instrument boards. Shows condensate collection system, steam pressure manometers, Variac, potentiometer and inclined manometers.

Turbulence generators. Upper generator for 11.5% level, lower one for the 3.0% level.
CROSSECTION OF EXPERIMENTAL TUBE

FIGURE 3
SECTION A-A THROUGH TEST CYLINDER WITH LOWER GUARD SECTION REMOVED

**FIGURE 4**

Side View of Working Section. Shows experimental tube protruding from working section, steam chest, and steam lines.
FIGURE 5
CONDENSATE COLLECTION SYSTEM

Glass Tube

Collection Burette

To Condensate Line From Experimental Segment
to the working section. The low turbulence level was that normally existing in the tunnel in the absence of these disturbing members.

The 11.5% turbulence level was generated by a grid of vertical one-half inch dowels mounted on one inch centers, in a frame which was readily removable from the tunnel. The frame matched the end flanges of the working and accelerating sections so that it could be bolted between them. The 3% level was generated by two screens placed next to each other and mounted on the same kind of frame as the above generator. The diameter of the wire and mesh size of the two screens were 0.78, 0.5 and 0.625, 0.25 inches respectively. A photograph of these two turbulence generators appears on page 32.

OPERATING PROCEDURE

Three different determinations are involved in the operating procedure. These are measuring the rate of condensation of steam in the experimental segment, finding the intensity of turbulence, and determining the velocity of the free air stream. They are discussed in this order.

At the beginning of a run the tube was rotated to the desired position, the blower and steam turned on and the three bleed lines opened. The steam pressure was regulated to about two inches of water gage and the Variac adjusted to give the steam entering the experimental segment a temperature of about 216°F. The air velocity was measured by the pitot tube and adjusted to the desired value by means of a damper on the entrance to the fan. The appara-
tus was allowed to run for about one and a half hours until it reached equilibrium. It was found that by this time the pressure indicated by the two monometers, one connected to the tube, the other to the steam chest, became constant and the rate of condensation was uniform. The time, the burette reading, and the temperature of the air were then recorded at intervals varying from five to twenty-five minutes depending on the rate of condensation.

The steam pressure and air velocity were maintained constant throughout the run. It was almost always possible during any given run to maintain the temperature of the air constant to within two degrees or less; if during a run the temperature should vary as much as four degrees the run was started over. Readings were continued until at least 20cc had been collected with no individual reading deviating from the average by more than 1.5% of the total collected.

The experimental measurements were limited to one half of the cylinder’s circumference; but as a check on the symmetry, check measurements of the heat transfer were carried out over the other half of the circumference without manifesting any substantial differences between the two halves. The half nearest the instrument board was used.

Turbulence level measurements were made before and after each set of experimental runs at the three levels. They were taken in the middle of the tunnel on the same line as the center line of the experimental tube by removing the tube and inserting the wake angle
instrument in its place at the top of the tunnel. The heating
element was connected to the Variac through the transformer; the
thermocouple wires were attached directly to the potentiometer.
The voltage was adjusted until the thermocouple measuring the
temperature of the wake between the 0° and 45° positions of the
probe behind the heating wire registered a significant deflection
on the potentiometer's galvanometer. The probe was then rotated
by hand and the galvanometer deflections at various angles record-
ed. From these points the intensity of turbulence was obtained
as indicated in the Sample Calculations section.

A pitot tube traverse across the velocity measuring section
at the medium and high turbulence levels indicated no significant
variation of the velocity across the tunnel. However, at the low
turbulence level the screen at the entrance to this section failed
to completely damp out the wake caused by the experimental tube.
A traverse indicated that the velocity was constant over a short
distance 2 1/2 to 5 inches from the side wall of the tunnel. Hence
the tip of the pitot tube was always placed four inches from the
wall when checking the velocity during a run. When making a
velocity traverse at the low turbulence level, the velocity head
was noted at this position. During the runs at this level, the
velocity was always adjusted to this value by moving the damper on
the fan until this reading was indicated by the inclined manometer
connected to the pitot tube. The corresponding velocity was
obtained by the traverse and integration shown in the appendix.
The results of this experiment are presented in the form of a correlation between the dimensionless Nusselt and Reynolds numbers. The heat transfer coefficient may be obtained from the Nusselt number by multiplying it by 0.0643.

\[ \text{Nu} = \frac{hD}{k} \]

\[ h = \frac{q}{A \Delta t} \]

\[ q = (60)(2.1h)R_c \]

\( R_c \) is cc/min of condensate collected

2.\( h \) is the heat of condensation of steam at 212°F, BTU's/cc

A is 0.0094 ft²

D is 0.196 ft

k is 0.0165 BTU's/°F hr ft

combining all factors:

\[ \text{Nu} = \frac{qD}{kA \Delta t} = \frac{(2.1h)(60)(R_c)(0.196)}{(0.0165)(0.0094)(\Delta t)} = 162,500 \frac{R_c}{\Delta t} \]

The heat transfer coefficient is based upon the total driving force from the condensing steam to the ambient air. All units are on the FPS system. The area chosen was the outside area of the experimental segment; the diameter used was the outside diameter of the experimental tube. The conductivity of the air film, k, was evaluated at the arithmetical mean temperature between the tube and the ambient air stream.

To obtain the local Nusselt number, the value of the rate
of condensation in cc/min and the driving force, which equals $212 - t_{air}$, are put into the above equation. These values are included for all runs in Table 1 of the appendix.

It will be noticed that no correction for radiation has been made. The reason for this is that the surface of the experimental segment was kept well polished; the emissivity was then a small but unknown quantity. Table 13, McAdams (15, p 395), lists several values for the emissivity of polished copper varying from 0.02 to 0.07. Schmidt and Wenner (19, p 6) appraised the value of the emissivity of their copper tube as 0.1. Rather than apply an arbitrary correction to the heat transfer values it was decided not to make a radiation correction. It will be shown below that in any case the correction is small and it may be applied by the reader if so desired.

$$q_r = 0.173 A \epsilon \left[ \left( \frac{T_1}{100} \right)^{0.1} - \left( \frac{T_2}{100} \right)^{0.1} \right]$$

$$= (0.173)(0.0094)(0.1) \left[ \left( \frac{672}{100} \right)^{0.1} - \left( \frac{510}{100} \right)^{0.1} \right] = 0.2 \text{ BTU/hr}$$

at $0^\circ$, low turbulence, low Reynolds number, $q$ is 23 BTU/hr at $0^\circ$, high turbulence, high $Re$, $q$ is 56.5 BTU/hr

low variation is $\frac{0.2 \times 100}{23} = 0.87\%$

high variation is $\frac{0.2 \times 100}{56.5} = 0.35\%$
The Reynolds number was based on the outside diameter of the experimental tube, the velocity in the unobstructed working section and the density and viscosity of air at $80^\circ F$, which was the average temperature of the air in the laboratory.

$$N_{Re} = \frac{DV\rho}{\mu}$$

$$V = \sqrt{2gH} = \sqrt{\frac{\Delta h}{12}} \frac{6h}{S.G._{air}}$$

$\Delta h$ = velocity head in inches of water

$D = 0.196$ ft

$\mu = 1.22 \cdot 10^5$#/ft sec

$\rho = 7.31 \cdot 10^2$#/ft$^3$

$S.G. = 1.178 \cdot 10^4$

$$N_{Re} = \sqrt{\frac{\Delta h}{12}} \frac{6h}{S.G._{air}} \left[ \frac{(0.196)(7.31 \cdot 10^2)}{1.22 \cdot 10^5} \right] = 79,500\sqrt{\Delta h}$$

The intensity of turbulence is obtained from the angular width of a thermal wake behind the hot wire of the wake angle instrument as discussed in the Theoretical Considerations and Procedure sections. The galvanometer deflections are plotted against the angle of the probe behind the wire and the angular width of the wake at one-half the maximum galvanometer deflection is determined. This angle is converted into percent turbulence by the following expression:

$$\% = \frac{\alpha^2 - \alpha_o^2}{1.53} \quad \text{where} \quad \alpha_o = 190.8 \sqrt{\frac{k}{c\gamma x \rho}}$$
Numerous determinations were made, especially at the lower turbulence levels where $\alpha$ is an important part of the expression; the results were then averaged. Table 2 in the appendix lists the values obtained; also included are several plots illustrating how to find $\alpha$. A sample calculation is shown below.

\[ k = 0.015 \text{ BTU/hr } ^\circ F \text{ ft} \]
\[ \alpha = 5.3 \text{ (from figure 15)} \]
\[ \rho = 7.34 \times 10^2 \text{ #/ft}^3 \]
\[ c = 0.24 \text{ BTU/# } ^\circ F \]
\[ x = 0.0261 \text{ ft} \]
\[ \Delta h = 0.87 \text{ in} \]

simplifying the expression for $\alpha$;

\[ \alpha_o = 190.8 \sqrt[3]{\frac{0.015}{0.0734(0.24)(3600)(0.0261)}} \frac{V}{V} = 17.8 \sqrt{\frac{1}{V}} \]

since $V = \frac{(6.43)(\Delta h)}{(11.78 \times 10^3)(12)}$

\[ \alpha_o = \frac{2.16}{\sqrt{0.87}} = 2.24 \]

\[ \% = \frac{(5.30)^2 - (2.24)^2}{1.53} = 3.1\% \]
RESULTS AND DISCUSSION

The results of this research are presented in figures 6 to 14 and in the tables in the appendix. The first three figures show how the local heat transfer is affected by velocity at three different turbulence levels. Figures 12, 13 and 14 show the effect of turbulence on the heat transfer at three different Reynolds numbers. The first three are discussed in the order of increasing turbulence. The low turbulence section contains a description of the curves and a comparison of these results with those of other investigators. The next two sections contain a description of the medium and high turbulence curves; the last three figures are then briefly discussed together. Following this, a discussion of the shape of these heat transfer curves based on the boundary layer theory presented under Theoretical Considerations is given.

RESULTS

Low Turbulence: A plot of the dimensionless Nusselt number against the angle θ from the forward stagnation point for the low turbulence level at three Reynolds numbers is presented in figure 6. A marked variation in local heat transfer coefficients around the cylinder is evident. Starting at the forward stagnation point, the local Nusselt number first decreases gradually and then more rapidly to a distinct minimum, at about 83°, which is less than half the value at 0°. It then increases rapidly and tapers off to a value at 180° that is about 25% higher than at the forward stagnation
point of $0^\circ$. The separate curves at the three different velocities of 33.1, 60.7 and 93.5 feet per second corresponding to Reynolds numbers of 39,000, 71,500 and 110,000 are seen to be practically identical in shape but are displaced toward higher Nusselt numbers with increasing velocities. The only pronounced difference is that the minimum becomes progressively sharper at higher Reynolds numbers.

It is to be noted that the data points give the average value of the heat transferred over the width of the experimental segment, which subtends an angle of $16.8^\circ$. Hence at those positions where the Nusselt number passes through a minimum, the true value must be lower than that measured. The actual maximum values, accordingly, are higher than the measured values. The degree to which the maxima and minima are in error depends upon the general slope of the curve on either side of these points. If the slope is very sharp, the values may be significantly in error; if the slope is small, the values are almost correct.

A number of investigators have worked on the local heat transfer coefficient distribution about a cylinder at the low turbulence prevailing in their particular wind tunnels. As far as is known, however, no data has been published showing how turbulence affects this distribution; although some work has been done on the relationship between overall coefficients and turbulence.

Comparison of the experimental data obtained at low turbulence in this research with those of other investigators is shown in
figures 9, 10 and 11. The agreement is quite good, although our data is a little higher than those of Schmidt and Wenner (19), whose experiments appeared to be the most carefully executed of any of the references studied.

Medium Turbulence: The Nusselt number distribution at 3% turbulence for three different Reynolds numbers is shown in figure 7. The main differences between these curves and the low turbulence ones are: (1) in general, the Nusselt numbers have increased considerably, (2) the first minimum is sharper and has moved back about 4°, and (3) a new maximum at about 110° followed by a minimum at 140° appears in the curves and becomes more prominent with increasing velocity. This maximum causes the values to rise much more rapidly from the first minimum and introduces a second minimum which isn't nearly as sharp as the first. This hook in the curves is only an inflection at the lowest velocity, but gradually develops into a distinct maximum of approximately the same magnitude as occurs at 0° by the time the highest velocity is reached. Increasing the velocity also causes the curves to be displaced evenly toward higher Nusselt numbers. The general shape of the curves preceding the first minimum and following the second one is the same at both the low and medium turbulence levels.

High Turbulence: Figure 8 shows how the local Nusselt numbers are affected by velocity at the 11.5% turbulence level. These curves are similar in shape to the high velocity curve at the
3% turbulence level.

The local Nusselt number is high at the front of the tube and decreases rapidly to a sharp minimum at 90°, which is less than half of that at 0°. The first minimum moves toward the rear of the cylinder about 1° with each increase in the turbulence level used. From this point the local Nusselt number rises sharply to a maximum around 113° which is just a little smaller than the maximum at 0°. The local Nusselt number then drops to a second minimum at 140°, which is almost the same magnitude as the first one, and finally rises gradually to a value at 180° which is about two-thirds of that at 0°. The shape of the initial drop of the heat transfer curve to the first minimum and the rise following the last one is about the same for all turbulence levels.

The curves at the three different Reynolds numbers have the same general shape, the differences being the increased sharpness of the new maximum and the increase in Nusselt numbers with increasing velocities. It should be pointed out that the first and second minima and the maximum between them occur at approximately the same position on the surface of the cylinder regardless of the velocity used.

Effect of Turbulence on Heat Transfer at a Given Reynolds Number: It is a well established fact that an increase in the intensity of turbulence has many of the same effects as increasing the Reynolds number. One of the effects of increasing the intensity of turbulence is lowering of the critical Reynolds number, thus
permitting the abnormal behavior occurring in this critical range to take place at a much lower Reynolds number. The critical range for cylinders has been placed between the Reynolds numbers of 150,000 and 300,000.

Upon inspecting figure 11 one will notice that the curve by Schmidt and Wenner at a Reynolds number of 170,000 is in this critical range, while the one obtained in this research is evidently not. The new maximum which suddenly appeared in their curve in this critical range becomes even more pronounced with increasing velocities and, although their other curves are not presented here, is quite prominent by the time their post critical curve at a Reynolds number of 420,000 is reached. In fact, the maximum value of the Nusselt number at this point is almost twice that at the 0° position.

In view of this fact it is not surprising to find a new maximum gradually developing in the heat transfer curves with increased turbulence in figures 12, 13 and 14. These figures show how the level of turbulence affects the heat transfer distribution at a given Reynolds number. It will be noticed that at none of the three velocities does a new maximum develop in the curves at the low turbulence level, thus indicating that the critical Reynolds number has not been reached. This new maximum then gradually develops with increasing turbulence at the low Reynolds number in figure 12, and becomes more pronounced with increasing turbulence at the higher Reynolds numbers in figures 13 and 14. The critical range has thus
been reached at these velocities because the turbulence has decreased the critical Reynolds number. Incidentally, this new maximum in the heat transfer curves appears at almost the same angle from the front of the cylinder as does that of Schmidt and Wenner in their critical and post critical regions.

DISCUSSION

Low Turbulence: The shape of the low turbulence curves is readily explainable by the boundary layer theory. The laminar layer of air attached to the cylinder at the stagnation point builds up gradually at first and then more rapidly as the point of separation is approached. Since heat transfer in a laminar layer is through molecular exchange, the process is slow and inversely proportional to the thickness of the layer. For this reason the heat transfer coefficients decrease steadily from the stagnation point to a minimum value at about 83°C. Fage and Falkner's experiments (10,11) indicated that at this point separation of the boundary layer occurred. The heat transfer data in this research also support this observation.

Even though there is no longer a laminar layer retarding heat transfer after the point of separation, there is no abrupt jump in the Nusselt number, but only a steady rise. At the point of separation there is a reversal of flow that extends a certain distance toward the rear of the pipe. This means that the random, turbulent flow becomes warmer as it approaches the point of separation from the rear. Since this air is the effective sink for the
heat that is transferred from the experimental cylinder, the
real driving force, $\Delta t$, will become smaller and the Nusselt num-
ber based on the apparent driving force will be underrated. This
effect is counteracted by the much greater heat flux in this
region as compared with the laminar flow region. The smooth rise
in Nusselt numbers following the separation point is then a combi-
nation of these two opposing forces.

The heat that is transferred at the rear of the cylinder is
higher than at the front because the statistical boundary layer here
is thinner than at the front and is also in turbulent motion.

High Turbulence: By keeping in mind the work done by Schubauer
and Linke on boundary layer phenomena discussed under Theoretical
Considerations, one can explain the peculiarity observed in the
Nusselt number distribution in the critical range of Reynolds num-
bbers. This can be done regardless of whether this range is reached
by increasing the velocity or increasing the turbulence level, thus
decreasing the critical Reynolds number.

The decrease in Nusselt numbers from the $0^\circ$ position to the
first minimum at the higher turbulence levels is due to the same
factors involved at low turbulence. That is, the building up of
the boundary layer and the decrease in the actual driving force due
to an increase in temperature of the boundary layer both contribute
to a lowering of the rate of heat transfer. Separation of the
laminar boundary layer then occurs at the first minimum in the heat
transfer curves.
Following this separation, the free boundary layer goes through a transition, the resulting fanning out accompanying the transition causes the layer to re-attach itself as a turbulent layer. This layer soon is unable to proceed further into an unfavorable pressure gradient and a second separation occurs. The re-attachment of the layer occurs at the peak of the new maximum in the heat transfer curves, while the subsequent separation is indicated by the second minimum of the heat transfer curves. This is substantiated further by the fact that at the high turbulence level the Nusselt number at the second minimum is approximately the same magnitude as the first minimum, thus indicating that the same phenomenon is occurring at both places, namely separation.

The cause for the sudden rise in heat transfer following the first minimum in the distribution curves is that in the short interval between separation and re-attachment there is no boundary layer as such. There is, however, rapid circulation of eddies in this region, thus causing a relatively high heat transfer to take place. The second maximum in the heat transfer curves occurs at the point where the turbulent boundary layer re-attaches itself to the cylinder. Due to the fact that a boundary layer, which is warmer than the main air stream, is now attached to the cylinder, the heat transfer decreases until the second minimum is reached. At this point separation of the boundary layer again occurs. The subsequent rise in heat flux occurs at about the same rate for all higher turbulence runs due to the similarity in flow conditions at the rear of the cylinder.
CONCLUSIONS

1. Within the range of Reynolds numbers covered in this experiment, the heat transfer coefficient distribution about a cylinder normal to an air stream of 0.9% turbulence has a maximum at the front of the cylinder and then decreases to a minimum at about 83°. The latter is less than half of the value at 0°. The heat transfer then increases to a second maximum at 180°, which is about 25% higher than at 0°. Increasing the Reynolds number increases the heat transfer considerably, but does not appreciably change the shape of the distribution curve.

2. Increasing the intensity of turbulence to 3% modifies the low turbulence distribution curves in that a new maximum appears at about 110°. This maximum grows from a marked inflection, at a Reynolds number of 39,000, to a value, at a Reynolds number of 110,000, which is a little larger than at either 0° or 180°. At each Reynolds number covered in this experiment, the increase in turbulence considerably increased the heat transfer.

3. At the 11.5% turbulence level, the distribution of coefficients about a cylinder is similar to that occurring at the highest Reynolds number at the 3% level. The heat transfer is high at the front, decreases to a minimum at about 91° and then increases sharply to a new maximum at about 112°, which is just a little smaller than that at 0°. This is followed by
a sharp drop to a second minimum, at 140°, and a gentle rise to values at 180° which are about two-thirds of those at 0°. The second minimum has about the same value as the first, thus indicating that the same basic flow phenomenon probably is occurring at both places. Increasing the Reynolds number shifts the curves toward higher Nusselt numbers, without changing their general shape, except for intensifying the new maximum.

4. At the 0.9% turbulence level, the critical range of Reynolds numbers was not reached at any velocity used; however when the turbulence was increased to the 3.0% and 11.5% levels, the critical number was lowered so that the critical range was reached even at the lowest Reynolds number used. This was indicated by the fact that the new maximum which gradually developed in the distribution curves with increasing turbulence is the same as that appearing in Schmidt and Wenner's curves (18, p 17) in the critical and post critical range of Reynolds numbers.

5. The shape of the heat transfer coefficient distribution curves in the critical range helps to substantiate Schubauer's (19) contention of a double separation of the boundary layer in this range. That is, the first minimum, the new maximum, and the second minimum in the heat transfer curves could quite conceivably be caused by the separation of a laminar boundary layer, the reattachment of the layer, and its
subsequent separation, respectively.

6. The position of the first sharp minimum at about $83^\circ$ in the heat transfer distribution curves at low turbulence helps to substantiate Page and Falkners (10,11) contention that at velocities below the critical Reynolds number separation of a laminar boundary layer occurs between $80^\circ$ and $85^\circ$.

Note: These conclusions hold only for the range of Reynolds numbers covered in this experiment, that is from 30,000 to 110,000.
Heat Transfer Coefficient Distribution At 3% Turbulence

$N_{Re} \approx 110,000$

$N_{Re} \approx 71,500$

$N_{Re} \approx 39,000$

Figure 7
Heat Transfer
Coefficient Distribution
at 11.5% Turbulence

Figure B
Comparison of Data by Several Investigators

1. Kruzhilin $N_{Re} 39,500$
2. Winding, Cheney $32,800$
3. Zapp $39,000$
4. Lohrisch $39,600$
5. Schmidt, Wenner $39,800$
6. Drew, Ryan $39,600$

Figure 9
Comparison of Data by Several Investigators

$\frac{N}{h_D} = \frac{h_D}{h_R}$

- Small $\text{Re} = 61,600$
- Zapp $\text{Re} = 71,500$
- Schmidt Werner $\text{Re} = 64,500$

Figure 10
Variation of Local Transfer Coefficients with Turbulence at $N_{Re} = 39,000$

Figure 12
Variation of Local Transfer Coefficients with Turbulence at $N_{Re} = 71,500$
Variation of Local Transfer Coefficients with Turbulence at $N_{Re} = 110,000$

**Figure 14**


NOMENCLATURE

A = heat transfer area, sq. ft.
B = eddy conductivity
C = specific heat of air, BTU/#°F
D = outside diameter of experimental tube, ft.
g = gravitational constant, ft/sec²
h = local heat transfer coefficient, BTU/hr*ft.²°C.
    h₃ = steam side coefficient
    h₄ = boundary layer coefficient
Δh = velocity head, inches of water
H = head of fluid flowing, ft.
k = thermal conductivity of air, BTU/hr°C ft.
kₚ = pressure drop coefficient
L = scale of turbulence
Nₐ = Nusselt number, \( \frac{hD}{k} \)
q = rate of heat transfer, BTUs/hr
Rₑ = rate of condensation in cc/mm
Rₑ = Reynolds number, \( \frac{DVP}{µ} \)
Rₑ = coefficient of correlation between instantaneous values of two velocity fluctuations separated by the cross stream distance z.
S.G. = specific gravity
t = temperature, °F
Δt = temperature difference between condensing steam and air temperature.
T = temperature, °R
\[ U = \text{overall heat transfer coefficient, BTUs/hr ft}^2 \, ^\circ F \]
\[ V = \text{velocity of the main air stream fps} \]
\[ X = \text{distance between thermocouple and hot wire, ft.} \]
\[ y = \text{distance across wake measured from the center} \]
\[ z = \text{cross stream distance} \]
\[ Z = \text{intensity of turbulence, \%} \]

Greek

\[ \alpha = \text{subtended angle} \]
\[ \epsilon = \text{emissivity} \]
\[ \rho = \text{density of air, \#/ft}^3 \]
\[ \mu = \text{viscosity of air, \#/ft sec} \]
\[ \theta = \text{number of degrees from forward stagnation point on experimental tube to the middle of the experimental segment} \]
Table I
Experimental Data and Calculated Results

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<th>% turbulence</th>
<th>V</th>
<th>Re</th>
<th>θ</th>
<th>tair</th>
<th>R_O x 10</th>
<th>h</th>
<th>Nu</th>
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Velocity Traverse and Integration

A velocity traverse was taken along four different lines on the same plane across the velocity measuring section of the tunnel, 16 inches behind the damping screen. The first was a horizontal traverse across the tunnel, halfway between the top and bottom of the section. The next three were vertical traverses from the top to the bottom of the section at $\frac{1}{2}$, $1\frac{1}{2}$ and 9 inches respectively from one side wall. After the traverses were made, it was noted that the readings were almost symmetrical about the vertical center line so that only half the cross section actually had to be integrated. On paper, the tunnel's cross section was marked off into 18 divisions vertically and horizontally, thus making $13^2$ rectangles geometrically similar to the main cross section. Velocity head readings in inches of water were taken at the center of each of these $\frac{1}{3} \times 1$ inch rectangles and their square roots were recorded in the appropriate position on the velocity traverse chart. Since only three vertical and one horizontal rows were thus filled in on the chart, the remaining six vertical rows were approximated in the following manner:

At each horizontal level, the difference between the vertical $1\frac{1}{2}$ and 9" traverses was prorated along that horizontal row by comparison to the percent change that the velocity underwent at each position along the horizontal traverse at the mid-section.

The integration of the flow across the cross section is accomplished by taking the sum of the values at each position and dividing by the total number of values. This is essentially the same
thing as finding the volume rate of flow through the section and dividing by the cross sectional area; hence the resulting velocity is the free stream velocity that would exist in the working section if the experimental tube were not present.

Velocity traverses were made only at the 0.9% turbulence level. See pages 27 and 38 for further details.

The experimental traverse data is presented in the following three tables. The resulting velocity traverse chart used for integrating the flow at the medium Reynolds number is also included. The figures within the heavier lines are actual traverse data, while the others are prorated values. The values are practically symmetrical about the vertical center line. It should be noted that the maximum variation in velocity as indicated by the chart is around 6%; hence the average velocity resulting from this integration shouldn't be significantly in error.
### Table II

**Velocity Traverse at Low Reynolds Number**

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### VELOCITY TRAVERSE CHART FOR MEDIUM REYNOLDS NUMBER

Inches from Side | Values are $\sqrt{\Delta h} \cdot 10^3$
---|---
0 | 758 \(\uparrow\) 720 \(\uparrow\) 785 \(\uparrow\) 785 \(\uparrow\) 785 \(\uparrow\) 785 \(\uparrow\) 781 \(\uparrow\) 772 \(\uparrow\) 759 \(\uparrow\) 759 \(\uparrow\) 772 \(\uparrow\) 781 \(\uparrow\) 787 \(\uparrow\) 787 \(\uparrow\) 790 \(\uparrow\) 790 \(\uparrow\) 758
2 | 726 \(\uparrow\) 716 \(\uparrow\) 701 \(\uparrow\) 701 \(\uparrow\) 701 \(\uparrow\) 701 \(\uparrow\) 701 \(\uparrow\) 727 \(\uparrow\) 727 \(\uparrow\) 763 \(\uparrow\) 789 \(\uparrow\) 907 \(\uparrow\) 916 \(\uparrow\) 916 \(\uparrow\) 916 \(\uparrow\) 916
4 | 940 \(\uparrow\) 916 \(\uparrow\) 901 \(\uparrow\) 901 \(\uparrow\) 901 \(\uparrow\) 901 \(\uparrow\) 901 \(\uparrow\) 867 \(\uparrow\) 834 \(\uparrow\) 834 \(\uparrow\) 866 \(\uparrow\) 890 \(\uparrow\) 906 \(\uparrow\) 914 \(\uparrow\) 914 \(\uparrow\) 914
6 | 943 \(\uparrow\) 905 \(\uparrow\) 896 \(\uparrow\) 896 \(\uparrow\) 896 \(\uparrow\) 896 \(\uparrow\) 896 \(\uparrow\) 873 \(\uparrow\) 852 \(\uparrow\) 852 \(\uparrow\) 874 \(\uparrow\) 890 \(\uparrow\) 900 \(\uparrow\) 905 \(\uparrow\) 905 \(\uparrow\) 905
8 | 750 \(\uparrow\) 726 \(\uparrow\) 718 \(\uparrow\) 718 \(\uparrow\) 718 \(\uparrow\) 718 \(\uparrow\) 711 \(\uparrow\) 872 \(\uparrow\) 872 \(\uparrow\) 895 \(\uparrow\) 911 \(\uparrow\) 921 \(\uparrow\) 926 \(\uparrow\) 926 \(\uparrow\) 926
10 | 767 \(\uparrow\) 734 \(\uparrow\) 726 \(\uparrow\) 726 \(\uparrow\) 726 \(\uparrow\) 726 \(\uparrow\) 719 \(\uparrow\) 902 \(\uparrow\) 878 \(\uparrow\) 878 \(\uparrow\) 902 \(\uparrow\) 919 \(\uparrow\) 929 \(\uparrow\) 934 \(\uparrow\) 934 \(\uparrow\) 934
12 | 767 \(\uparrow\) 740 \(\uparrow\) 730 \(\uparrow\) 730 \(\uparrow\) 730 \(\uparrow\) 722 \(\uparrow\) 722 \(\uparrow\) 904 \(\uparrow\) 904 \(\uparrow\) 878 \(\uparrow\) 878 \(\uparrow\) 904 \(\uparrow\) 922 \(\uparrow\) 934 \(\uparrow\) 940 \(\uparrow\) 940
14 | 755 \(\uparrow\) 750 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\) 740 \(\uparrow\)
16 | 767 \(\uparrow\) 750 \(\uparrow\) 737 \(\uparrow\) 737 \(\uparrow\) 737 \(\uparrow\) 737 \(\uparrow\) 727 \(\uparrow\) 903 \(\uparrow\) 870 \(\uparrow\) 870 \(\uparrow\) 902 \(\uparrow\) 926 \(\uparrow\) 942 \(\uparrow\) 950 \(\uparrow\) 950 \(\uparrow\) 950
18 | 767 \(\uparrow\) 918 \(\uparrow\) 918 \(\uparrow\) 918 \(\uparrow\) 918 \(\uparrow\) 918 \(\uparrow\) 910 \(\uparrow\) 93 \(\uparrow\) 870 \(\uparrow\) 870 \(\uparrow\) 902 \(\uparrow\) 926 \(\uparrow\) 942 \(\uparrow\) 950 \(\uparrow\) 950 \(\uparrow\) 950
20 | 945 \(\uparrow\) 945 \(\uparrow\) 945 \(\uparrow\) 945 \(\uparrow\) 945 \(\uparrow\) 945 \(\uparrow\) 898 \(\uparrow\) 898 \(\uparrow\) 898 \(\uparrow\) 898 \(\uparrow\) 898 \(\uparrow\) 898 \(\uparrow\) 898 \(\uparrow\) 898
22 | 965 \(\uparrow\) 927 \(\uparrow\) 918 \(\uparrow\) 918 \(\uparrow\) 918 \(\uparrow\) 918 \(\uparrow\) 910 \(\uparrow\) 93 \(\uparrow\) 870 \(\uparrow\) 870 \(\uparrow\) 902 \(\uparrow\) 921 \(\uparrow\) 927 \(\uparrow\) 927 \(\uparrow\) 927 \(\uparrow\) 927
24 | 961 \(\uparrow\) 916 \(\uparrow\) 908 \(\uparrow\) 908 \(\uparrow\) 908 \(\uparrow\) 908 \(\uparrow\) 889 \(\uparrow\) 889 \(\uparrow\) 889 \(\uparrow\) 889 \(\uparrow\) 889 \(\uparrow\) 889 \(\uparrow\) 889 \(\uparrow\) 889
26 | 961 \(\uparrow\) 906 \(\uparrow\) 900 \(\uparrow\) 900 \(\uparrow\) 900 \(\uparrow\) 900 \(\uparrow\) 895 \(\uparrow\) 895 \(\uparrow\) 895 \(\uparrow\) 895 \(\uparrow\) 895 \(\uparrow\) 895 \(\uparrow\) 895 \(\uparrow\) 895
28 | 916 \(\uparrow\) 916 \(\uparrow\) 908 \(\uparrow\) 908 \(\uparrow\) 908 \(\uparrow\) 908 \(\uparrow\) 886 \(\uparrow\) 886 \(\uparrow\) 886 \(\uparrow\) 886 \(\uparrow\) 886 \(\uparrow\) 886 \(\uparrow\) 886 \(\uparrow\) 886
30 | 758 \(\uparrow\) 790 \(\uparrow\) 785 \(\uparrow\) 785 \(\uparrow\) 785 \(\uparrow\) 785 \(\uparrow\) 785 \(\uparrow\) 781 \(\uparrow\) 772 \(\uparrow\) 759 \(\uparrow\) 759 \(\uparrow\) 772 \(\uparrow\) 781 \(\uparrow\) 787 \(\uparrow\) 790

Average $\sqrt{\Delta h} = 0.901$
Table V

Turbulence Data

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1 shown in figure 15
2 shown in figure 15
3 shown in figure 16
max. deflection 29
$\alpha$ at 1/2 max. 2.8°
turbulence 0.87%

max. deflection 22.5
$\alpha$ at 1/2 max. 5.3°
turbulence 3.1%
max. deflection 7.4
\( \alpha \) at 1/2 max. 18°
turbulence 11.7%