Heat transfer in a two-dimensional jet flow over an isothermal curved wall.

K. S. Rao
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UMI®
HEAT TRANSFER IN A TWO-DIMENSIONAL
JET FLOW OVER AN ISOTHERMAL CURVED WALL

A THESIS

Submitted to the Faculty of Graduate Studies through
the Department of Mechanical Engineering in partial
fulfillment of the requirements for the Degree
of Master of Applied Science at the
University of Windsor.

by

K. S. Rao

B. E. (Hons.), Andhra University, India, 1962.

Windsor, Ontario, Canada

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ABSTRACT

Experimental results were presented for velocity, skin friction, temperature and heat transfer measurements in a two dimensional turbulent air jet of limited height flowing around a semicircular cylindrical wall of copper held at isothermal conditions. The jet was initially at ambient temperature. The jet exit Reynolds numbers range from 4500 to 18000 and dimensionless downstream distances from 0 to 400.

The jet growth and the maximum velocity decay with downstream distance were graphically plotted. The non-dimensional velocity profiles agreed with Glauert’s theoretical velocity profile and approximate similarity was obtained over most of the flow. The wall shear stress and coefficients of skin friction were evaluated by the Preston method and checked by an alternate method.

The temperature profiles were plotted nondimensionally, using $y_m/\delta$ as the length scale for different combinations of nozzles and downstream locations and compared with the plane wall jet profiles. The temperature profiles followed one power law in the thin layer close to the wall and another in the rest of the jet. The profiles were more rounded than the plane wall jet and the flat plate (1/7th power) case profiles. A temperature distribution correlation was presented for the sensitive region next to the wall.

The local heat transfer coefficients were higher than
those for plane wall jet and for circular cylinders exposed to free jet flow. This was expected due to the adherence of the jet to the wall (Coanda effect) and the high intensity of turbulence and mixing. The Nusselt numbers based on the nozzle width were correlated to the jet exit Reynolds numbers and the dimensionless downstream distance. The correlation was compared with the results of the previous investigators.
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NOTATION

\( S = \) Jet nozzle width at exit, [L]
\( R = \) Radius of the curved wall, [L]
\( x = \) Distance measured along the wall from the origin of the curve in the jet flow direction, [L]
\( y = \) Distance from wall in the radial direction, [L]
\( z = \) Distance in the spanwise direction from the jet centreline, [L]
\( U_s = \) Jet velocity at nozzle exit, (ft./sec.)
\( u = \) Jet velocity, (ft./sec.)
\( u_m = \) Local maximum jet velocity, (ft./sec.)
\( P_a = \) Atmospheric pressure, (lbs./ft.²)
\( P_s = \) Static pressure on the wall, (lbs./ft.²)
\( P_t = \) Total pressure in the jet, (lbs./ft.²)
\( P = \) Stagnation pressure in the air supply duct, (lbs./ft.²)
\( \rho = \) Density of the fluid, (slugs/ft.³)
\( \nu = \) Kinematic viscosity of the fluid, (ft.²/sec.)
\( k = \) Thermal conductivity of the fluid, (BTU/hr./ft./°F)
\( \mu = \) Dynamic viscosity, (lb. sec./ft.²)
\( c_p = \) Specific heat of air, (BTU/lbm./°F)
\( h = \) Local heat transfer coefficient, (BTU/hr./ft.²/°F)
\( \theta = \) Angular position measured from the origin of the curve, degrees
\( x_1 = \) Distance measured along the wall from the nozzle exit in the jet flow direction = \( x + 3.625 \) in., [L]
\( y_{m/2} = \) Ordinate at half local maximum velocity, [L]
\[ r = \text{Temperature recovery factor} \]

\[ T_{a,w.} = \text{Adiabatic wall temperature, (°F)} \]

\[ T_w = \text{Isothermal mean wall temperature, (°F)} \]

\[ T_a = \text{Ambient temperature, (°F)} \]

\[ T_f = \text{Film temperature} = \frac{T_w + T_a}{2}, \text{(°F)} \]

\[ T = \text{Static temperature in the jet, (°F)} \]

\[ \tau_0 = \text{Shear stress, (lb./ft.}^2) \]

\[ c_f = \text{Coefficient of skin friction, (} \tau_0 \sqrt{\frac{1}{2}} \rho u_m^2 \text{)} \]

\[ Re = \text{Jet exit Reynolds number, (U_S/v)} \]

\[ (Nu)_s = \text{Nusselt number, (hS/k)} \]

\[ (Nu)_x = \text{Local Nusselt number, (hx/k)} \]

\[ Pr = \text{Prandtl number of the fluid, (c_P \mu g_c /k)} \]

\[ (St)_s = \text{Stanton number, (h/ρc_P U_s)} \]

\[ u_x = \text{Local shear velocity} = \sqrt{\frac{\tau_0}{\rho}} \text{ (ft./sec.)} \]

\[ q = \text{Heat flow rate, BTU/hr./ft.}^2 \]

\[ g_c = 32.2, \text{ (ft./sec.}^2) \]

\[ J = 778 \text{ ft. - lbs./BTU} \]

**SUBSCRIPTS:**

\[ s = \text{Nozzle exit} \]

\[ x = \text{Downstream distance} \]

\[ w = \text{Wall} \]

\[ a = \text{Ambient} \]

\[ m = \text{Maximum} \]
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CHAPTER I

INTRODUCTION
INTRODUCTION

The name 'wall jet' was first used by Glauert to refer to a jet flowing over a surface. When a jet emerges from a narrow slit tangential to a curved wall, it bends and flows along the wall. This phenomenon was well known as 'Coanda effect'.

Jets have an important place in heat transfer applications since they provide a practical way to obtain high heat and mass transfer coefficients over large areas of exposed surfaces. Applications of jets are found in paper drying, heating metal ingots, heat exchangers, and other process industries such as steel, glass, paper, etc., which involve heat transmission between a gaseous flow phase (flame or smoke) and a solid or liquid phase.

With the introduction of high altitude pressurized cabin bombers in World War II, ice formation on the inside surface of aircraft windshields had become a serious problem. One of the techniques available for eliminating this difficulty was that of blowing a jet of heated air over the inside of the windshield surface. This led to the first experimental study of heat transfer in wall jets. Since then this subject had been of interest to several investigators. A plane test surface was used in all these experiments, and heat transfer correlations
were obtained for isothermal or step wall temperature conditions. No work had been published so far on heat transfer in the jet flow over a curved wall.

The present investigation was an experimental study of heat transfer in a curved wall jet. The wall was a semi-circular cylinder of copper held at isothermal conditions. The jet nozzle width and the wall temperature were the main variables. The surrounding fluid was stationary and the flow was in the incompressible range.
CHAPTER II

LITERATURE SURVEY
LITERATURE SURVEY

Since the present investigation is primarily concerned with the heat transfer in wall jets, this literature survey deals essentially with the heat transfer aspects of the problem. This does not imply that a sound knowledge of the flow characteristics of wall jets is less important. Many excellent references are available for this purpose.

The symbols and the subscripts used in this section are explained in 'Notation' (p. viii).

2.1 AERODYNAMICS

2.1.1 PLANE WALL JET

The first theoretical analysis for both laminar and turbulent, radial and plane wall jets was given by Glauert (Ref. 1). The analytical solution for the laminar case was based on similarity considerations. For the turbulent case, Glauert demonstrated that complete similarity was not possible over the whole of a wall jet. An approximate similarity was obtained by making suitable assumptions for the eddy viscosity distribution. Glauert's predictions for a radial wall jet were confirmed experimentally by Bakke (Ref. 2). Sigalla (Ref. 3) presented the results of an experimental investigation of the distribution of skin friction along the wall of a plane turbulent wall jet, using Preston's method. Bradshaw and Gee (Ref. 4) made experimental investigations of the jet flow over both flat and curved surfaces, with and without an external stream. Their results agreed with Glauert's theoretical predictions and their skin friction
measurements were also in agreement with Sigalla's results. 

Schwarz and Cosart (Ref. 5) obtained the skin friction coefficient from an experimentally measured velocity profile by solving the integral momentum equation. Their value of skin friction was about twice that of Sigalla. Seban and Back (Ref. 6) presented the experimental velocity profiles and skin friction in a plane wall jet in the presence of very low values of the free stream velocity. Myers, Schauer and Eustis (Ref. 7) made analytical predictions for the shearing stress, maximum velocity decay and jet thickness by momentum-integral methods. Corresponding experimental data was also presented over greater ranges than previously available. The shear stress was measured experimentally by a 'hot film technique' and compared with the wall shear stress computed from the measured velocity profiles.

2.1.2 CURVED WALL JET

Nakaguchi (Ref. 8) conducted experimental and theoretical studies on a jet along a curved wall and observed that the velocity profile in the jet was very similar to that of the free turbulent jet except in the minor part of the flow close to the surface. He also mentioned that turbulent mixing and centrifugal force acting on the jet influenced considerably its characteristics. Newman (Ref. 9) made a dimensional and theoretical analysis for an incompressible turbulent wall jet flow over a cylinder and compared them with his experimental measurements. Theoretically, he assumed that the velocity profiles were similar, the streamlines were circles about the cylinder centre, the skin
friction was negligible, and the momentum of the flow was nearly conserved. Fekete (Ref. 10) carried out an experimental investigation with certain improvements in Newman's equipment. The velocity profiles agreed with Glauert's theoretical predictions for a plane wall jet over a major portion of the flow. In the regions approaching separation, the jet thickness increased at a rate which was more than twice that of a plane wall jet.

2.2 HEAT TRANSFER

2.2.1 PLANE WALL JET

Zerbe and Selna (Ref. 11) obtained experimentally an empirical equation for the heat transferred to a flat surface from a plane, turbulent heated air jet directed tangentially to the surface. The equation for \((Nu)_s\), the Nusselt number based on nozzle width, could be written as:

\[
(Nu)_s = 0.071 (Re)_s^{0.8} (X/S)^{-0.6} \quad \text{--- (2-1)}
\]

valid over the ranges \(6000 < Re < 44000\)
\(15 < X/S < 196\)

Jacob, Rose and Spielman (Ref. 12) experimentally determined the coefficients of heat transfer from a two dimensional turbulent air jet to a flat plate of asbestos ebony, with entrainment of water vapor from the environment. The jet was not attached to the plane wall initially and the local heat transfer coefficient was calculated in terms of difference between the local wall temperature and maximum local jet temperature in the boundary layer. The results were correlated by the equation:

\[
(Nu)_s = 0.105 (Re)_s^{0.8}(X/S)^{-0.4} \quad \text{--- (2-2)}
\]

valid over the ranges \(1.34 \times 10^3 < Re < 2.94 \times 10^6\)
\(15 < X/S < 126\)
Seban and Back (Ref. 6) obtained, experimentally, the temperature profiles based on an adiabatic wall, and with some alteration of the theoretical eddy diffusivity distribution, obtained general agreement with the theoretical predictions of temperature profiles. A heat transfer correlation based on the Colburn analogy was given:

\[(St)_s = 0.25 (Re)_s^{0.25} (X/S)^{-0.60} \quad \text{(2-3)}\]

valid over the ranges \((1400 < Re < 7200)\)
\((18 < X/S < 258)\)

There was a small free stream velocity present in the experiment and the injection air was heated above the free stream temperature initially.

Mathieu (Ref. 13) studied, experimentally, the aerothermic characteristics of a jet evolving in the presence of a plane wall. The study included velocity and temperature profiles, intensities of turbulence, shear stress distribution and heat transfer between the wall and the jet. The jet was heated initially and the test surface was cooled by cold water circulation. The heat transfer correlation was of the form:

\[(Nu)_x = 2.70 (Re)_x^{0.65} \quad \text{(2-4)}\]

valid over the range \((50 < Re < 1000)\)
\((X/S < 450)\)

Myers, Schauer and Eustis (Ref. 14) made an analytical and experimental study of the heat transfer characteristics of a two-dimensional, incompressible, turbulent plane wall jet. The analytical prediction was made for the local Stanton number and the data were presented for a step-wall temperature distribution.
Experimental results of the temperature surveys in the wall jet, were shown for different downstream distances, wall temperatures, and Reynolds numbers. The temperature profiles were plotted non-dimensionally by referring the distances from the wall (y) to the local thermal boundary layer thickness, the latter being evaluated by an integration method. The influence of an unheated starting length on the heat transfer was predicted analytically, and compared with the experimental results. The comparison showed some difference in the exponential values and this was explained as due to the difference between the actual and assumed theoretical temperature profiles. It was found that the temperature profiles were not influenced by Reynolds number and became "fuller" as X/S increased. Also the wall jet temperature profiles were "fuller" than the 1/7 power profiles obtained in the flat plate work. It was further observed that the wall jet temperature profiles did not seem to fit any 1/n power profile very well and no attempt was made to correlate the data in any other manner. By considering the wall jet as a combination of the flat plate flow and the free jet, an average value of 1.6 was assumed for the ratio of eddy diffusivity of heat to that of momentum, in the theoretical analysis. The experimental correlation for the heat transfer was:

$$\left(St\right)_s = 0.118 \left(Re\right)_s^{-0.2} \left(X/S\right)^{-0.5625} \left[1 - \left(t/x\right)^{0.45}\right]^{-0.0625}$$

over the range (16600 < Re < 38100)

$$30 < X/S < 181$$
A method of extending the fundamental solution for the step wall temperature case to arbitrary heating conditions was indicated, and due to the linearity of the thermal energy equation, the solutions could be superimposed. The procedure was given by Tribus and Klein (Ref. 15).

Akfirat (Ref. 16) measured local heat transfer coefficients between an isothermal flat plate and a two-dimensional wall jet and the results were compared with the heat transfer data previously available. The heat transfer information for the laminar and transitional regions were also included. The correlation given for a fully developed turbulent flow was:

\[(\text{Nu})_s = 0.097 (\text{Re})_s^{0.8} (x/S)^{-0.6} \]  

over the ranges \((1385 < \text{Re} < 16620)\)  
\((30 < x/S < 80)\)

2.2.2 CURVED WALL JET

No work, either analytical or experimental, could be found during the author's literature survey to determine the effect of wall curvature on the heat transfer in turbulent wall jets.

Schuh and Persson (Ref. 17) investigated heat transfer in a two-dimensional free jet flow of limited height over a circular cylinder placed symmetrically in the flow. They found that mean heat transfer coefficients for thin jets adhering to the surface were 20 percent higher than the values for unlimited parallel flow case. This was attributed to the Coanda effect and the high intensity of turbulence.
CHAPTER III

TEST FACILITIES
TEST FACILITIES

The overall view of all the equipment used is shown in Fig. 1 (a). A schematic diagram is given in Fig. 2 showing the references to the letter code used.

The test facilities consisted of 1) air jet 2) test surface 3) hot water supply 4) traversing mechanism 5) probes and measuring instruments.

3.1 AIR JET

The air jet was provided by a type HS, size 200 American Standard centrifugal fan (A). This fan was driven by a 5 H.P., 1745 rpm General Electric induction motor. A 30 inch long wooden guiding duct (B) with a rectangular cross section was attached to the fan exit. A honeycomb flow straightener was placed in the guiding duct to ensure reduction of the turbulence level induced by the fan. A wooden contraction duct (C) was placed after the guiding duct. A converging nozzle (D) made of brass with a rectangular exit of 0.25 in. width and 9 in. span was attached to the contraction duct. The contraction ratios for the nozzle together with the duct were 62:1 in the direction of 0.25 in. width and 2.5:1 in the perpendicular direction. A Kiel probe with a 0.125 in. diameter shroud was placed in the contraction duct to measure air supply pressure. Two other nozzles of width 0.125 in. and 0.0625 in. were also available to vary the jet-exit Reynolds number.
3.2 TEST SURFACE

The test surface is shown in Fig. 1a and Fig. 1b. A layout of the test surface is shown in Fig. 3. The test surface was made of 0.0625 in. thick copper sheet, with an outside radius of 9 inches. The height of the test surface was 9 inches which was the same as the span of the jet nozzle. Since our interest was to study the heat transfer in the fully developed region of a curved wall jet, the curved surface was limited to an angle of 135° (see Fig. 3). The surface was extended tangentially by a length of 3.625 in. at both ends of the curve. The purpose was to avoid a developing flow and to reduce the possibility of jet separation occurring over the curved surface. Two semi-circular (radius 22 in.) end plates made of 0.3125 in. thick plexiglas were mounted on top and bottom of the test surface to obtain a quasi two-dimensional jet flow.

The inner side of the test wall was fitted with a water jacket, 1.5 in. wide, having the same shape as the wall. The frame of the water jacket was carefully machined from 0.625 in. thick phenolite, a non-conducting fibrous material, and the back wall of the water jacket was made of 0.0625 in. thick copper sheet. Both the outer and the inner walls of the jacket were screwed to the phenolite frame at 2 in. intervals. Non-conducting O-rings and water sealings were used to secure a leak-proof water jacket.

The test surface was fully instrumented. Locations of the wall pressure taps, thermocouples and heat flux sensor were
shown in Fig. 3. The wall static pressure taps consisted of 0.040 in. holes drilled into the test surface. The line of the taps was 0.5 in. above the centre line. The wire used was 30 gauge copper-constantan and had an accuracy of ±1.5° F up to 200° F. To measure the heat flow rate, five temperature-monitored heat flow sensors, type-20453 made by RDF Corporation were mounted flush on the surface at the locations shown. The sensors were small, very thin foil type devices with provision to read both the surface temperature and the heat flow rate. (See Appendix II).

The test surface was finished smoothly and mirror-polished to minimize radiation heat losses. The back wall of the water jacket and the edges of the test surface were carefully insulated with one inch thick fiberglass board and the resulting heat losses due to conduction were assumed to be negligible.

3.3 HOT WATER SUPPLY

The arrangement to feed the hot water at a required temperature to the jacket consisted of a three way mixing valve, a pressure regulator, a distributor and a draining device. The temperature of the hot water was varied by an appropriate mixture of hot and cold water. The mixing was achieved by a pneumatically operated Johnson Control valve. To obtain isothermal conditions for the test surface, the hot water was fed to the jacket from top through several equally-spaced inlets and drained at the bottom. The surface temperature was maintained constant and was low enough for heat losses by radiation to be
negligible.

3.4 TRAVERSING MECHANISM

The traversing mechanism is shown in Fig. 1b. It contained a size 12 in. dial-type vernier caliper. This arrangement provided for a three-dimensional traverse of the wall jet—the full length in the direction of the flow, 6 in. in a direction perpendicular to the surface (with an accuracy of 0.001 in.) and 6 in. in the spanwise direction (with an accuracy of 0.01 in.). There was a provision to lock the radial traverse to the top end plate at any desired position to make a traverse of the jet there. Two probes could be mounted on the traverse, one above the other.

3.5 PROBES AND MEASURING EQUIPMENT

The probes are shown in Fig. 1b and Fig. 4.

The velocity distribution close to the wall in the inner layer was measured by the flat-end boundary layer type total pressure probe. The probe tip dimensions were 0.015 in. x 0.045 in. outside, and 0.008 in. x 0.030 in. inside. The velocity profiles in the rest of the jet were determined by a pitot static probe. The temperature distribution across the jet was found by the total temperature probe, specially flattened at the tip for thermal boundary layer work. The flow impinged on the tip where a stagnation region was formed. Air from the stagnation region passed into the probe to the thermocouple junction behind the tip and then returned to the stream through the two small aspiration holes in the walls of the tube adjacent to the junction. The probe design was suggested
by Hottel and Kalitinsky (Ref. 18) and was successfully used to measure the true total temperature to within 0.75°F at velocities up to 1000 ft/sec. All the probes were made to specifications, and supplied by the United Sensor Corporation. An electric eye arrangement was used in the boundary layer measurements to indicate the contact of the probe with the test surface.

A multi-tube inclined manometer was utilized in making all wall static pressure measurements. The total pressures were measured with atmospheric pressure as the reference, by a Lambrecht inclined manometer filled with alcohol. The wall thermocouples and the boundary layer temperature probe readings were taken with a Leeds and Northrup temperature potentiometer capable of reading temperatures up to an accuracy of 0.25°F. All thermocouple readings were made with reference to a common ice junction, containing crushed ice pellets in a thermally insulated glass jar. The thermocouples were connected to the potentiometer through a ten point contact switch, which allows quicker reading of various temperatures. The temperature potentiometer was standardized initially by using the ice junction.

The rate of heat flow at each downstream location was measured by the heat flow sensors described in appendix II. The sensors were bonded to the test surface near the centre line with AP cement, which ensured a good bond without introducing any film resistance. The output of the sensor in microvolts was measured by a Honeywell Precision Potentiometer Model 2779 to an
accuracy of 0.01 μV. This ensured an accuracy of 0.2 BTU/ft²-hr
in terms of heat flow. A Honeywell Guarded Microvolt Null De-
tector Model-3990 with a resolution of 10 nanovolts to null
sensitivity, was used in conjunction with the microvolt potentiom-
ter to measure the heat flow. A sensor calibration curve sup-
plied by the manufacturer, gave a multiplication factor cor-
responding to each surface temperature from -300° F to +440° F.
The sensor output was to be multiplied by this factor to obtain
the actual heat flow at that particular location.

To set up a required temperature difference between
the surface and the ambient quickly, and to check the isothermal
conditions at the wall often, an Alnor Indicating Pyrometer with
a flexible arm was used in the experiment. The arm carried
thermocouple No. 4040 and read temperatures within an accuracy
of 1% of the full scale range.

The details of all the equipment used are listed in
Tab. 1.
4.1 CALIBRATION

The nozzle velocity profiles were measured with the flat tip boundary layer type pressure probe. The calculation of velocities was based on ambient pressure and temperature. The jet exit velocity was kept constant at a nominal value of 140 ft./sec. for all the three — 0.25, 0.125, 0.0625 in. nozzles. The jet exit velocity profile at the 0.125 in. nozzle exit is shown in Fig. 5. The profile remained effectively uniform and flat over 90 per cent of the nozzle width, varying only ±1 per cent from the central core velocity of 141 ft./sec. From the nature of the velocity distribution, it was concluded that the jet velocity at the nozzle exit was practically uniform.

A mean temperature recovery factor r, equal to 0.9, was adopted for calculating static temperature, since the variation of the recovery factor was small over the range of jet velocities and temperatures under investigation. The static temperatures in the jet were calculated from the temperature measurements \( T_p \) of the total temperature probe, by using the relation:

\[
T = T_p - r \frac{u^2}{2g_c c_p J} \tag{4-1}
\]

4.2 TWO-DIMENSIONALITY

The nozzle was tested for two-dimensionality by measuring the velocities at different locations along the vertical centerline in the spanwise direction. The velocity remained uniform over 90 per cent of the span, as shown in Fig. 6, thus confirming the two-dimensionality of the jet at the nozzle exit.
The jet was checked for two-dimensionality at an angular position of 67.5 degrees downstream, by measuring the velocity profiles at three different spanwise locations, 1.5 in. apart. The results are plotted as shown in Fig. 7 and 8. The two-dimensionality was good in the inner layer and there was some scatter in the outer layer. Since the extent of scatter was small, it was concluded that the jet was quasi-two-dimensional in the fully developed region.

4.3 VELOCITY PROFILES

The experiment was divided into two parts—(1) cold runs and (2) hot runs.

The cold runs were ideally suited to measure velocity profiles, assuming there was no substantial influence of heating on the velocity measurements at Prandtl number of 0.71.

The curved wall jet velocity distributions were determined at seven downstream locations. Velocity calculations near the wall were based on the wall static pressure at that location. A linear static pressure distribution from the wall, across the jet, to the ambient value was assumed (Ref.9). The flat tip total pressure probe was used in the inner layer.

The velocity distribution in the outer layer was measured by the pitot static probe. This probe was mounted under the total pressure probe with 1 in. distance between centres. The readings were taken at suitable intervals across the jet (perpendicular to the wall) until the manometer showed a zero reading, just outside the jet boundary.
4.4 HOT RUNS

The test surface was heated to a suitable temperature by circulating hot water through the jacket. The water temperature was controlled by a Johnson hot and cold water mixing control valve operated by compressed air at 20 psig. A required nominal temperature difference between the surface and the ambient was set up by trial and error. The rate of water flow was adjusted to ensure a steady and uniform surface temperature. The latter was measured by several wall thermocouples at various downstream locations. An indicating pyrometer was used for a quick reading of wall temperatures at points away from the thermocouples in the spanwise direction. The pyrometer was particularly helpful in quickly setting up the required temperature difference.

The maintenance of a constant surface temperature proved very difficult due to sudden fluctuations in the temperature of the building-hot water-supply line. As a result, most of the experiments were carried out during nights when the load and temperature fluctuations on the hot water supply were at a minimum.

The hot runs consisted of (1) velocity profile measurements, (2) temperature profile measurements and (3) heat transfer measurements.

The velocity profiles were measured (1) to assist in calculating static temperatures and (2) to check the cold run velocity profiles. There was no significant change in either the slope near the wall or the magnitudes of the velocity profile due to heating, thus bearing out the initial assumption that
heating the wall has little influence on the velocity profile in the range of temperatures used in the investigation.

The temperature profiles were measured at seven downstream locations for each slot Reynolds number. The flat-tipped total temperature probe, specially designed for boundary layer work was used in the radial traverse together with the flat pitot probe to determine the static temperatures in the jet. The static temperature evaluation was based on the assumption that the jet was incompressible, since the maximum Mach number was only 0.13.

The heat transfer measurements at each downstream location were made by the micro-foil heat flow sensor. The sensors were fixed on the test surface 0.5 in. below the centreline at each location. The surface temperature indicated by the sensor was measured by the temperature potentiometer. The sensor heat flow was measured by the precision potentiometer.

The heat flow measurements were repeated for three nominal temperature differences--40°, 25°, 15°F; three nozzle Reynolds numbers--1.8 x 10^4, 9 x 10^3, 4.5 x 10^3, and seven downstream locations--θ = 0, 20, 40, 67.5, 95, 115, and 135 degrees.
CHAPTER V

EXPERIMENTAL RESULTS
5.1 DATA REDUCTION

The fluid was treated as incompressible because the nominal maximum jet velocity was only 140 ft./sec. From barometric pressure and room temperature, the density of air was determined. All other fluid properties like specific heat, thermal conductivity and viscosity were evaluated at the film temperature, $T_f$. Since the experimental data was obtained with small temperature differences, it was found very convenient to treat the fluid as a constant-density medium and this considerably simplified the calculations. Since the Prandtl number of air also did not vary significantly with the temperatures used, a constant value of 0.71 was assumed.

The experimental data was reduced with the help of an IBM 1620 computer. The input data were the radial distance from the wall ($y$), temperature probe reading, pressure reading, ambient conditions, wall static pressure and the fluid properties. The values of $u$, $u/u_m$, $y/y_m$, $T$, $T_f$, etc., were obtained as output information. The computer was also used for making correlations.

5.2 PRESENTATION AND DISCUSSION OF RESULTS

5.2.1. VELOCITY PROFILES

Neglecting fluid compressibility, velocity was determined from the relation $P_t - P_s = \frac{1}{2} \rho u^2$, with $\rho = \frac{P_a}{R_c T_a}$, where $R_c$ is the gas constant. The error involved in taking $\rho$ as constant was assumed to be negligible. The static pressure $P_s$ at each point in the traverse was obtained by assuming a linear variation across the jet from the value of the static pressure at the wall to the ambient pressure just outside the jet (Ref. 9). The velocity($u$)
and the distance from the wall \((y)\) were made non-dimensional by referring to the local maximum velocity \((u_m)\) and the ordinate at half maximum velocity \((y_m)\) respectively.

The non-dimensional velocity profiles at different downstream locations along the curved wall are shown in Figs. 9 to 11. Three different Reynolds numbers—\(1.8 \times 10^4\), \(9 \times 10^3\) and \(4.5 \times 10^3\), based on the nozzle width were used in the investigation.

It is seen, from Fig. 9, that the experimental results closely follow the theoretical velocity profile of a plane wall jet, given by Glauert (Ref. 1). This agrees with the observations of Newman and Fekete (Refs. 9 and 10).

The non-dimensional velocity profiles for \(\theta = 20, 40, 67.5, 95\) and \(115\) degrees are plotted together to check for similarity, as shown in Figs. 9, 10, and 11. With the exception of \(\theta = 0^\circ\) due to the effects of transition from plane to curved wall and \(\theta = 135^\circ\) due to its nearness to the region of separation, the non-dimensional velocity profiles collapsed on a single curve within limits of allowable scatter, thus showing approximate similarity of velocity profiles over most of the flow.

The shape of velocity profiles changed very slowly with distance downstream. The experimental data and theory were in good agreement except in the outer part (Fig. 9). Complete similarity of velocity profiles could not be obtained over the whole of the jet. This agreed with Glauert's conclusions for a plane wall jet (Ref. 1). Sawyer (Ref. 21) also showed that to
obtain complete similarity the ratio of jet thickness to wall radius of curvature should be constant along the jet and this condition was satisfied only for a jet blowing over a surface of logarithmic spiral profile.

5.2.2 JET GROWTH

The growth of the wall jet is shown in Fig. 12 superimposed over a schematic plan of the curved test surface to indicate the proportion of growth at various downstream locations. The same data is replotted in Fig. 13 and the curves of best fit are drawn. It was seen from the figures that the jet had a higher growth rate at lower Reynolds numbers, particularly at downstream locations far away from the nozzle. This could be explained as follows: in two-dimensional jets blowing around circular cylinders, considerable entrainment of fluid from the surroundings takes place, into the outer part of the jet. This entrainment causes a growth of the jet thickness and a decay of its maximum axial velocity. Newman (Ref. 9) postulated that the jet momentum was very nearly conserved along the downstream distance. The jet at lower Reynolds numbers starts initially with a lesser momentum and loses its velocity rapidly. So, in order to conserve the momentum, there should be a higher entrainment and this explains the higher values for jet thickness as the Reynolds number decreases. Fekete's measurements (Ref. 10) showed that in regions approaching separation, the jet thickness increased
at a rate which was more than twice that of the plane wall jet.

The jet growth data is replotted in Fig. 14 as \( \frac{y_m}{2}/R^3 \) vs \( \frac{y_m}{2}/R \). Corresponding plot of Newman (Ref. 9) is also shown. A straight line plot could not be obtained for the experimental data. The plot showed curvature initially. This could be explained that up to \( \theta = 40^\circ \), the jet exhibited the effects of transition from the plane entry length to the curved test surface. No attempt was made to fit any correlation for the growth of the jet.

The data is replotted in Fig. 15 as \( \frac{y_m}{2}/R \) vs \( x/S \) and compared with the results of a curved wall jet without the initial plane entry length and a plane wall jet of similar Reynolds number, nozzle width etc. The data plot was in between the two and this indicated the transition effects mentioned above. The jet flow parameters are shown in Tab. III.

5.2.3 MAXIMUM VELOCITY DECAY

Plots of \( \frac{\rho U_m^2 R \theta}{(P-P_a)S} \) vs \( \theta \) are shown in Figs. 16 to 18.

The wall jet continuously entrains fluid from the surroundings: thus the jet width increases and the fluid velocity decreases with increasing \( \theta \). The rate of decrease of the local maximum velocity \( (U_m) \), is larger in any jet along a wall than in a free turbulent jet because of the skin friction of the wall.

At sufficiently high Reynolds numbers, it can be assumed that the effect of skin friction on the jet momentum is small and negligible (Ref. 9). Thus the sum of the moment
of momentum and the momentum of the pressure forces about the centre of the circular cylinder is constant. With this assumption, the non-dimensional maximum velocity coefficient based on actual jet momentum can be written:

\[
\frac{\rho U_m^2 R \theta}{(P-P_a)S} = f(\theta)
\]

This equation can be used to predict the rate of decay of maximum velocity of the jet and is valid only outside the potential core, where the velocity essentially remains constant.

5.2.4 SURFACE SHEAR STRESS

Preston's method was used to calculate the surface shear stress with a flat pitot probe. The method is described in Appendix I (a).

As a check on Preston's method, the surface shear stress values were evaluated by Rajaratnam and Froelich's method described in Appendix I (b). This method did not include corrections for turbulence and displacement effects.

The mean wall shear stress values obtained from both of the above methods are tabulated in Table IV. It can be seen that the agreement is good.

The coefficients of skin friction \( (c_f) \) are calculated and plotted as shown in Fig. 19. The region of local dynamic similarity where the 'law of the wall' (see App. Ia) may be expected to hold varies from about 1/5th to 1/20th of the boundary-layer thickness for conditions remote from, and close to, separation.
respectively (Ref. 19). It was assumed that the uncertainty in the determination of the extent of this region resulted in the observed scatter of about 5% for values of the skin friction coefficient (Fig. 19).

The skin friction data is replotted as shown in Fig. 19a and the line of best fit is drawn. The correlation obtained for the data was of the form:

$$c_f/\nu (U_m y_m/\nu)^{0.25} = 0.063(x/S)^{-0.12} \quad \text{valid over range:} \quad 30 < x/S < 350 \quad 4500 < Re < 18000$$

5.2.5 TEMPERATURE PROFILES

The temperature traverses were made at seven downstream locations in the jet for each nozzle Reynolds number. Five of the experimental temperature profiles are shown in Figs. 20 to 22. The profiles were normalised by plotting the ordinate $(T_w - T)/(T_w - T_a)$ vs $y/y_{m/2}$.

It can be seen that $(T_w - T)/(T_w - T_a) = 1 - (T - T_a)/(T_w - T_a)$. Hence, the normalised temperature in the jet referred to the ambient, $(T - T_a)/(T_w - T_a)$ can also be plotted as shown on the right in the figures. In all the plots, $y_{m/2}$ was used as the length scale.

A comparison is made with temperature profile of a plane wall jet (Ref. 7) of corresponding parameters in Fig. 20. It can be seen that the temperature profile is more rounded than the plane wall jet profile. The slope of the temperature profile was larger near the wall. The rate of decay of temperature

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was approximately the same as that of a plane wall jet in the outer part.

Approximately 90 percent of the temperature differential occurs within $y/y_m/2 = 0.1$ in a thin thermal boundary layer very close to the wall. The profiles are not significantly influenced by Reynolds number and become "fuller" in this region as the distance downstream of the start of heating increased. This is to be expected since the jet gets hotter as it moves downstream along the heated test surface. For $y/y_m/2 > 0.20$ in the outer layer of the jet, the temperature profiles indicated approximate similarity at positions downstream of $\theta \geq 40$ degrees. The profiles showed a very weak dependence on the downstream distance in this region.

The temperature data $(T_w - T)/(T_w - T_a) vs y/y_m/2$ is replotted on a log-log paper as shown in Figs. 23 to 25. The plots clearly demonstrated that the temperature profile followed two different $1/n$ power laws, one in the temperature-sensitive region very close to the wall $(y/y_m/2 < 0.1)$ where 90 percent of the temperature differential occurs and another in the rest of the jet. The plots also indicated the dependency on downstream distance, as observed above. The thickness of the temperature-sensitive region decreased as $x/S$ increased, as could be seen from the plots.

The temperature distribution data in the sensitive region adjacent to the wall for the three Reynolds numbers was correlated to the normalised downstream distance $(x/S)$ and distance from the wall $(y/y_m/2)$ as shown in the log-log plot of Fig. 26.
The correlation obtained was:

\[
\frac{(T_w-T)}{(T_w-T_a)} = 0.85 \left(\frac{y}{y_{m/2}}\right)^{0.07143} \left(\frac{x}{S}\right)^{0.0561}, \quad (5.2)
\]

Range: \(10 < x/S < 400\), \(y/y_{m/2} < 0.1\)
\(4500 < Re < 18000\)

5.2.6 HEAT TRANSFER

The heat flow rate \(q\) was measured at seven downstream locations for each nozzle Reynolds number.

The coefficient of heat transfer was given by \(h = q/\Delta t\), where \(q\) was the heat flow rate measured by the microfoil heat flow sensor at each location and \(\Delta t\) = mean temperature differential between the isothermal test surface and the ambient temperatures.

Three temperature differentials, \(40^\circ\), \(25^\circ\) and \(15^\circ\) F were used and the measured values of \(h\) did not show any significant difference. The mean value of \(h\) and the corresponding Nusselt number were calculated at each downstream location and tabulated in Tab. V.

Plots of \(h\) vs \(u_m\) and \(h\) vs \(x_1/S\) are shown in Figs. 27 and 28 respectively. It can be seen that the coefficient of heat transfer increases with \(Re\) and \(u_m\) and decreases with \(x_1/S\). This can be explained as follows: at distances closer to the nozzle, the jet was almost at ambient temperature leading to higher rates of heat flow. As the jet moves further downstream, it gets relatively hotter, thus lending to lower values for \(q\). Another factor contributing to the decrease in \(h\) was the increasing mass entrainment into the jet from the ambient as the downstream distance increases.

The heat transfer data for the three Reynolds numbers is plotted as \(Nu/(Re)^{0.8}\) vs \(x_1/S\) on a log-log paper in Fig. 29.
The dimensionless parameters that enter into the correlation of the heat transfer data are the Nusselt number, \((\text{Nu})_s\), the Reynolds number, \((\text{Re})_s\), and the dimensionless distance from the nozzle exit, \(x_1/s\). The correlation expected was of the form:

\[
(\text{Nu})_s/(\text{Re})_s^M = C(x_1/s)^N
\]

The correlation shown in Fig. 29 was a good fit to the data, approximating a power law with a scatter of points in a band of about 11 per cent width. The scatter was assumed to be due to transition and separation effects. The correlation obtained was:

\[
\text{Nu} = 0.71(\text{Re})^{0.8}(x_1/s)^{-0.61}
\]
valid over the range: \(30 < x_1/s < 400\)
\(4500 < \text{Re} < 18000\)

The heat transfer correlation is compared with the corresponding correlations for a plane wall jet (Ref. 16) and for circular cylinders exposed to free jet flow (Ref. 17), as shown in Fig. 30. The test surfaces were held at a constant temperature in all cases and the other flow parameters were closely identical. It can be seen that the Nusselt numbers are much higher for the curved wall jet than in the other two cases. This can be explained by the ability of the jet to adhere to the heated surface, and the high intensity of turbulence and mixing in the curved wall jet.

A full comparison with the heat transfer results of various investigators for wall jets is shown in Tab. VI, which also includes the range of variables. Each of them had given a different form of correlation; for comparison the data was rearranged in the form:

\[
\text{Nu} = C (\text{Re})^M (X/S)^{-N}
\]
CHAPTER VI

CONCLUSIONS
1. Heating the jet does not significantly affect the magnitude of the velocity profile or its slope near the wall.

2. The temperature profile is more rounded than the plane wall jet and the flat plate (1/7 power law) profiles, and its slope near the wall is larger. The temperature distribution in the outer part is approximately the same in both cases.

3. The temperature profile is not significantly influenced by Reynolds number and becomes "fuller" with increasing downstream distance.

4. The temperature profile follows two different $\frac{1}{n}$ power laws, one in the temperature-sensitive region very close to the wall ($y/ym/ < 0.1$) where approximately 90 percent of the temperature drop occurs and another in the rest of the jet. The thickness of this sensitive region decreases as $x/S$ increases. In the outer layer of the jet ($y/ym/ > 0.20$), the temperature profiles show approximate similarity at positions downstream of $\theta \geq 40$ degrees.

5. The temperature distribution data in the sensitive region can be correlated as:

$$
\frac{TW - T}{TW - Ta} = 0.85 (y/ym/)^{0.07143}(x/S)^{0.0561}
$$

within ranges: $10 < x/S < 400$, $y/ym/ < 0.1$

$4500 < Re < 18000$
6. The coefficient of heat transfer increases with \( Re \) and \( u_m \) and decreases with \( x_1/S \).

The heat transfer data can be correlated in the form:

\[
(Nu)_s = 0.71(Re)_s^{0.8} (x_1/S)^{-0.61},
\]

with a maximum deviation of \( \pm 11 \) per cent within ranges:

\[
30 < x_1/S < 400 \quad \quad 4500 < Re < 18000
\]

7. The Nusselt numbers for the curved wall jet are much higher than for plane wall jet and for circular cylinders exposed to free jet flow.

8. The coefficient of skin friction has higher values at lower Reynolds numbers and the skin friction decreases as the jet moves downstream. The data correlation is of the form:

\[
\frac{c_f}{2} \left( \frac{u_m y_{m/s}}{v} \right)^{0.25} = 0.063 \left( x/S \right)^{-0.12}
\]

valid over the range:

\[
30 < x/S < 350 \quad \quad 4500 < Re < 18000
\]
APPENDIX I
SURFACE SHEAR STRESS CALCULATION

A. Preston's Method

Many cases exist, where the various methods of measuring skin friction, such as deduction from the pressure drop, towing tests, and direct measurement of the force on a small element of a flat surface, are not very suitable for application. Preston (Ref. 19) recognized this and developed a simple method of determining local turbulent skin friction on a smooth surface which utilizes a round pitot tube resting on the surface.

Preston's method is based on the evidence of the existence of a region of local dynamic similarity for a limited distance from the surface. In this region the conditions of the flow are functions only of \( \tau_0 \), \( \rho \), \( \nu \), and some suitable length. A universal non-dimensional relation is obtained in this region such that

\[
\frac{u}{U_*} = f\left(\frac{\nu L}{V}\right)
\]  \hspace{1cm} \text{(A-1)}

Due to the assumption of local dynamic similarity, this relation is independent of the pressure gradient. The dynamic head \((P_t - P_s)\) measured by one of a set of geometrically circular pitot probes resting against the wall is related to \( \tau_0 \) as

\[
\frac{(P_t - P_s) d^2}{\rho \nu^2} = F(\tau_0 d^2) \frac{1}{(\rho \nu^2)}
\]  \hspace{1cm} \text{(A-2)}

where \( d \) is the outside tube diameter. The displacement of the effective center and scale effects are included in the relation and a single calibration can be used for all geometrically similar
circular pitot tubes resting against a wall.

Preston also experimented with a flattened boundary layer pitot probe similar to the one used in the present investigation. For work with flat pitot probes, the relation given was
\[
\log_{10} \left( \frac{\tau_0 y^2}{\rho v^2} \right) = G \left( \frac{P_t - P_s y^2}{\rho v^2} \right) \quad (A-3)
\]

The calibration which Preston gives is represented by a straight logarithmic line over the range
\[
(5.0 \leq \log_{10} \left( \frac{(P_t - P_s) y^2}{\rho v^2} \right) \leq 7.5)
\]
and shows considerable scatter outside the range. The calibration line over this range is given by
\[
\log_{10} \left( \frac{\tau_0 y^2}{\rho v^2} \right) = -1.372 + \frac{7}{8} \log_{10} \left( \frac{(P_t - P_s) y^2}{\rho v^2} \right) \quad (A-4)
\]

In the present investigation the results lie within the range given by Preston and the wall shear stress was calculated directly from the above equation.

**B. RAJARATNAM and FROELICH'S METHOD**

Surface shear stress was also evaluated by a relation given by Rajaratnam and Froelich (Ref. 20). The relation, in its simple form, can be written as
\[
1.34 Y = X \log X \quad (A-5)
\]
where \( X = 7.5 \frac{u_m y}{\nu} \sqrt{C_f/2} \) and \( Y = \frac{uy}{\nu} \). This law of the wall is valid over the range:
\[
225 < X < 7500
\]
\[
394 < Y < 21700
\]
When this relation was plotted on a double log paper, it was found that the relation between \( X \) and \( Y \) is essentially a...
linear one and could be given as

\[ X = 1.19 Y^{0.875} \quad \text{------------------------} \quad (A-6) \]

At any section of the boundary layer, taking a point in the wall law region, such that the value of \( Y \) is in the range of 394 to 21700, \( X \) could be obtained from the graph. Then \( \alpha_f \) and hence \( \tau_o \) could be easily obtained.

The above equation could be further simplified and written as

\[ \frac{u^*}{u} = 0.158 \left( \frac{Y}{uy} \right)^{0.125} \quad \text{------------------------} \quad (A-7) \]

This equation could be directly used to evaluate \( \tau_o \).

In the present investigation, surface shear stress values were calculated from the above equation. The values were tabulated in Tab.IV, and compared with the results obtained by Preston method. The agreement was very good.
APPENDIX II

MICRO-FOIL HEAT FLOW SENSORS

To evaluate the heat transfer coefficient $h$ in the experiment, the micro-foil heat flow sensors supplied by the RdF Corporation, Hudson, New Hampshire were used.

The heat meter consists of a thin wafer of low density, compression resistant insulation enclosed between two 0.00025 in. butt-bonded foil thermocouples. The foils form a low impedance differential thermocouple system which produces an electromotive force proportional to the temperature difference between wafer faces and measures the heat flow across the very thin wafer.

Because of the heat sensor's fast response, it is suitable for applications involving direct measurement of transient heat fluxes. Its low thermal resistance—believed to be lowest among all heat flow transducers—makes it especially effective in measuring heat exchanges between exposed surfaces and their surroundings.

The application of the sensor should be by bonding to the surface with any of several cements, except those that dry by solvent evaporation. Epoxy cement is best for ease of application, the choice depends on temperature. Double sided pressure sensitive tape can be used also where thickness and loss of response time is not critical. AP cement was used in this experiment for bonding.

According to the manufacturer, the sensor has an output of the order of 0.05 microvolts per BTU/HR-Ft². The output varies
with the surface temperature. A calibration curve over a range of temperature from -300°F to +400°F was supplied by the manufacturers. A specimen of the technical data and calibration curve were shown in the following pages. The change in calibration factors due to change in base temperature is approximately 0.1% per °F. The surface temperature could be measured by either the wall thermocouples, or the thermocouples incorporated in the heat-flow sensor. In the experiment, it was found that the latter usually give a slightly lower value.
CALIBRATION

MICRO-FOIL HEAT FLOW SENSOR

RoF PART NO. 20453
SERIAL NO. 107

HEAT FLOW SENSOR

Output at 70°F 0.053 microvolts per $\frac{\text{Btu}}{\text{HR-FT}^2}$

Polarity: (For heat flow into surface)
  White - Positive (+)
  Red - Negative (-)

Temperature Multiplication Factor: See Attached Graph

*Thermal Resistance: 0.005 $\frac{\text{F}}{\text{per Btu}}$ per $\frac{\text{HR-FT}^2}{\text{F}}$ (TYP)

*Heat Capacity: $0.02 \frac{\text{Btu}}{\text{Ft}^2}$ per $\text{F}$ (TYP)

Response Time: 0.06 sec. (62% response to step function) (TYP)

THERMOCOUPLE

<table>
<thead>
<tr>
<th>ASA Type</th>
<th>Material</th>
<th>Polarity</th>
<th>Color</th>
</tr>
</thead>
<tbody>
<tr>
<td>T</td>
<td>Copper</td>
<td>Pos (+)</td>
<td>Blue</td>
</tr>
<tr>
<td></td>
<td>CONSTANTAN</td>
<td>Neg (-)</td>
<td>Red</td>
</tr>
</tbody>
</table>

Output per ASA C96.1-1964 and NBS Circular 561.

*Thermal resistance is the temperature difference between the front surface and rear mounting surface of the sensor per unit of heat flow through the sensor. Heat capacity is the amount of heat required to raise the mean temperature of the sensor 1°F. Typical values of these two properties are given primarily to indicate sensor capabilities and are required for heat flow calculations only in very rare instances.

DATE: Aug 24 67
BY: T. Harris

RoF CORPORATION
Hudson, New Hampshire

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SURFACE TEMPERATURE, DEGREES FAHRENHEIT

MICRO-FOIL HEAT FLOW SENSOR

OUTPUT MULTIPLICATION FACTOR VS RECEIVING SURFACE TEMPERATURE
(70°F BASE)

RoF CORPORATION
HUDSON, NEW HAMPSHIRE
REFERENCES


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13. Mathieu, J.


15. Tribus, M. & Klein, J.


16. Akfirat, J. C.


17. Schuh, H. & Persson, B.


18. Hottel, H. C. & Kalitinsky, A.


19. Preston, J. H.


20. Rajaratnam, N. & Froelich, C. R.


<table>
<thead>
<tr>
<th>Ser.No.</th>
<th>Equipment</th>
<th>Model/Type</th>
<th>Range</th>
<th>Accuracy</th>
<th>Measurement</th>
<th>Manufacturer</th>
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<tbody>
<tr>
<td>a)</td>
<td>Flat pitot probe</td>
<td>BA-031-20-C-19-1 Boundary layer type</td>
<td>Total pressure</td>
<td>±0.01 in.</td>
<td>Wall static</td>
<td>United Sensor &amp; Control Corp., Water-town, Mass., U.S.A.</td>
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<tr>
<td>b)</td>
<td>Flat temperature probe</td>
<td>USC-5053 Boundary layer type</td>
<td>Total temperature</td>
<td>±0.01 in.</td>
<td>Wall static</td>
<td>&quot;</td>
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<td>c)</td>
<td>Pitot static probe</td>
<td>PDA-24-F-22-KL Reinforced tubing</td>
<td>Dynamic pressure</td>
<td>±0.01 in.</td>
<td>Wall static</td>
<td>&quot;</td>
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<tr>
<td></td>
<td>Inclined Manometer bank</td>
<td>Water-filled 0-25 in.</td>
<td>Wall static</td>
<td>±0.01 in.</td>
<td>&quot;</td>
<td>T.E.M. Instrument Ltd., Crawley</td>
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<td>Inclined Manometer</td>
<td>Alcohol-filled slopes 1:1, 1:2, 1:5, 1:10, 1:25</td>
<td>Total pressure</td>
<td>±0.01 in.</td>
<td>&quot;</td>
<td>Wilh Lambrecht KG., West Germany</td>
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<td>Temperature potentiometer</td>
<td>Cat. No. 8693 -100 to +400°F</td>
<td>Wall temperature</td>
<td>±0.01 in.</td>
<td>&quot;</td>
<td>Leeds &amp; Northrup Co., Philadelphia, U.S.A.</td>
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<td>Temperature indicating pyrometer</td>
<td>Thermocouple 4040</td>
<td>Wall temperature</td>
<td>±0.01 in.</td>
<td>&quot;</td>
<td>Alnor Instrument Co., Chicago, U.S.A.</td>
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<td></td>
<td>Hot water mixing</td>
<td>Johnson control -40 to +160°F</td>
<td>Wall temperature</td>
<td>±0.01 in.</td>
<td>&quot;</td>
<td>Johnson Service Co., Milwaukee, Wis., U.S.A.</td>
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<td>Precision Potentiometer</td>
<td>Model 2779 microvoltmeter</td>
<td>±(0.02% of reading +0.007 μv)</td>
<td>±0.01 in.</td>
<td>Heat flow rate</td>
<td>Honeywell Denver, Colo., U.S.A.</td>
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<td>Guarded Null detector</td>
<td>Model 3990 microvolt type</td>
<td>±0.01 in.</td>
<td>±(0.02% of reading +0.007 μv)</td>
<td>&quot;</td>
<td>&quot;</td>
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<tr>
<td></td>
<td>Heat flow sensors</td>
<td>Part No. 20453 Micro-foil type</td>
<td>±0.01 in.</td>
<td>±(0.02% of reading +0.007 μv)</td>
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<td>RDF Corporation Hudson, N. H., U.S.A.</td>
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<tr>
<td>Location</td>
<td>θ degrees</td>
<td></td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>----------</td>
<td>-----------</td>
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<td>8</td>
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<th>Downstream distances</th>
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<td>x in.</td>
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<td>3.14</td>
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<td>6.28</td>
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<tr>
<td>10.60</td>
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<tr>
<td>14.92</td>
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<td>18.06</td>
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<tr>
<td>21.20</td>
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(Ref. Fig. 3)
<table>
<thead>
<tr>
<th>Location</th>
<th>θ, degrees</th>
<th>Nozzle width $S$, inches</th>
<th>Local maximum velocity $U_m$, ft./sec.</th>
<th>Distance $y_m/2'$, inches</th>
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<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>0.250</td>
<td>0.125</td>
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<td>0.250</td>
<td>120.849</td>
<td>86.958</td>
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<td>0.125</td>
<td>91.323</td>
<td>65.814</td>
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<td>4</td>
<td>40</td>
<td>0.063</td>
<td>69.244</td>
<td>47.367</td>
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<td>44.559</td>
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<td>0.125</td>
<td>30.546</td>
<td>21.308</td>
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<td>115</td>
<td>0.063</td>
<td>28.548</td>
<td>3.646</td>
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<td>8</td>
<td>135</td>
<td>0.250</td>
<td>18.004</td>
<td>4.269</td>
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<tr>
<td>Location</td>
<td>$\theta$, degrees</td>
<td>PRESTON</td>
<td>RAJARATHNAM &amp; FROELICH</td>
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<td>8</td>
<td>135</td>
<td>1.600</td>
<td>1.725</td>
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MEAN WALL SHEAR STRESS $\tau_o$, lb/ft.$^2 \times 10^3$
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<th>Location</th>
<th>$\theta$, degrees</th>
<th>Nusselt Numbers</th>
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<td>2</td>
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<td>113.82</td>
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<td>67.88</td>
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<td>50.13</td>
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## TABLE VI
### COMPARISON OF HEAT TRANSFER CORRELATIONS

<table>
<thead>
<tr>
<th>Se. No.</th>
<th>Authors</th>
<th>Test Wall/Jet</th>
<th>Coefficient $C$</th>
<th>Exponent $M$</th>
<th>Exponent $N$</th>
<th>Range $X/S$</th>
<th>Range $Re$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Zerbe and Selna (Ref. 11)</td>
<td>plane, heated jet</td>
<td>0.071</td>
<td>0.80</td>
<td>0.60</td>
<td>15-196</td>
<td>6000-44000</td>
</tr>
<tr>
<td>2</td>
<td>Jacob, Rose and Spielman (Ref. 12)</td>
<td>plane, asbestos, constant temp.</td>
<td>0.105</td>
<td>0.80</td>
<td>0.40</td>
<td>15-126</td>
<td>1340-2.94 x 10^6</td>
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<td>3</td>
<td>Seban and Back (Ref. 6)</td>
<td>plane, bakelite, adiabatic</td>
<td>0.178</td>
<td>0.75</td>
<td>0.60</td>
<td>18-288</td>
<td>1400-7200</td>
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<td>4</td>
<td>Myers, Schauer and Eustis (Ref. 14)</td>
<td>plane, copper, step temp.</td>
<td>0.083</td>
<td>0.80</td>
<td>0.56</td>
<td>30-181</td>
<td>16600-38100</td>
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<td>5</td>
<td>Akfirat (Ref. 16)</td>
<td>plane, aluminum isothermal</td>
<td>0.097</td>
<td>0.80</td>
<td>0.60</td>
<td>30-80</td>
<td>1385-16620</td>
</tr>
<tr>
<td>6</td>
<td>Present investigation</td>
<td>curved, copper isothermal</td>
<td>0.71</td>
<td>0.80</td>
<td>0.61</td>
<td>30-400</td>
<td>4500-18000</td>
</tr>
</tbody>
</table>

$$Nu = C (Re)^M (X/S)^{-N}$$
Fig. 1a  TEST FACILITIES
Fig. 1b  TEST SURFACE AND TRAVERSE
Fig. 2  SCHEMATIC DIAGRAM OF TEST FACILITIES

A  FAN
B  GUIDING DUCT
C  CONTRACTION DUCT
D  NOZZLE
E  KIEL PROBE
F  TEST SURFACE
G  HOT WATER JACKET
H  END PLATES
K  RADIAL TRAVERSE
L  PROBES
M  TABLE
Fig. 3 LAYOUT OF THE CURVED TEST WALL AND INSTRUMENTATION
Fig. 4  TOTAL TEMPERATURE PROBE

NOTE: ALL DIMENSIONS APPROX.

UNITED SENSOR & CONTROL CORP.
85 School St., Watertown, Massachusetts 02172

Material: 18-8

Drawn By

Date 7/18/67

BOUNDARY LAYER T/C USC-5053
Fig. 6  TWO-DIMENSIONALITY CHECK AT NOZZLE EXIT
$\theta = 67.5^\circ$, location $-5$

$S = 0.25$ in., $Re = 18000$

- $z = +1.5$ in.
- $z = 0$
- $z = -1.5$ in.

Fig. 7 DOWNSTREAM VELOCITY DISTRIBUTION

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Maximum spanwise non-dimensional velocity gradient in the jet outer layer

\[
= \frac{d}{dz} \left( \frac{u}{u_m} \right) = 0.03 \text{ in./inch}
\]

Fig. 8 DOWNSTREAM TWO-DIMENSIONALITY CHECK
$S = 0.25\text{ in.}$
$Re = 16000$

- $\theta = 25^\circ$
- $\theta = 40^\circ$
- $\theta = 57.5^\circ$
- $\theta = 95^\circ$
- $\theta = 115^\circ$

Glaucert's theoretical velocity profile

Fig. 9 VELOCITY PROFILES

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$S = 0.125$ in.
$Re_* = 9000$
- $e = 25^\circ$
- $e = 40^\circ$
- $e = 67.5^\circ$
- $e = 90^\circ$
- $e = 115^\circ$

Fig. 10 VELOCITY WPROFILES
$S = 0.0625 \text{ in.}$

$Re_0 = 4500$

- $\phi = 20^\circ$
- $\phi = 40^\circ$
- $\phi = 67.5^\circ$
- $\phi = 90^\circ$

Fig. 11  VELOCITY PROFILES

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Fig. 13  GROWTH OF CEF
$R = 9 \text{ in.}$

- $S = 0.250 \text{ in.}$
- $S = 0.125 \text{ in.}$
- $S = 0.063 \text{ in.}$

Fig. 14 GROWTH OF CURVED WALL JET

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Fig. 15 COMPARISON OF THE GROWTH OF JET

- Curved wall jet (Peter Tu)
- Plane wall jet (Hu)

R = 9 in., S = 0.25 in.

$u_g = 141 \text{ ft./sec.}$

$Re = 18000$
Fig. 16 MAXIMUM VELOCITY DECAY

\[ \frac{\rho u^2 R \theta}{(p - p_a)s} \]

R = 9 in., S = 0.25 in.
Reₙ = 18000
Fig. 17

$R = 9\text{ in.}, S = 0.125\text{ in.}$

$Re = 9000$

$\frac{\rho u^2 R e}{(R - P_a) S}$

$\theta$ (degrees)

Max. Velocity Decay

R = 9 in., S = 0.125 in.

Re = 9000

$\frac{\rho u^2 R e}{(R - P_a) S}$

$\theta$ (degrees)

Max. Velocity Decay
Fig. 18  MAXIMUM VELOCITY DECAY

\[ \frac{\rho u_m^2 R \theta}{(P - P_a)S} \]

\( R = 9 \text{ in.}, S = 0.0625 \text{ in.} \)
\( \text{Re}_* = 4500 \)
Fig. 19  SKIN FRICTION

\[ \frac{c_f \times 10^3}{f_{\text{ref}}} \]

- \( \triangle \) \( S = 0.25 \text{ in.}, \ Re = 18000 \)
- \( \circ \) \( S = 0.125 \text{ in.}, \ Re = 9000 \)
- \( \square \) \( S = 0.063 \text{ in.}, \ Re = 4500 \)
\( \frac{c_f}{2} \left( \frac{u_m}{v} \frac{y_m}{2} \right)^{1/4} = 0.063 \left( \frac{x}{S} \right)^{-0.12} \), Range: \( 4500 < Re < 18000 \)

Fig. 19a  \( \frac{X}{S} \rightarrow \)

SKIN FRICTION CORRELATION

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Fig. 20  TEMPERATURE PROFILES

- PLANE WALL JET
  (Myers, Schauer, and Bustis)

\[
\frac{(T_c - T)}{(T_w - T_a)}
\]

- \( S = 0.250 \) in.
- \( Re = 18000 \)
- \( \theta = 20^\circ \)
- \( \theta = 40^\circ \)
- \( \theta = 70^\circ \)
- \( \theta = 90^\circ \)
- \( \theta = 135^\circ \)

\[
y/y_2
\]
$S = 0.0625 \text{ in.}, \quad Re = 4500$

\[ \frac{(T_{hi} - T)}{(T - T_d)} \]

- $\circ \theta = 67.5^\circ$
- $\triangle \theta = 95^\circ$

- $\circ \theta = 20^\circ$
- $\triangle \theta = 40^\circ$

Fig. 25 LOGARITHMIC TEMPERATURE PROFILES
\[(T_w - T)/(T_w - T_a) = 0.05 \left( \frac{y}{h} \right)^{0.07143} (x/s)^{0.0561}, \text{ Range: } 10 < x/s < 400 \]

\[4500 < Re < 18000\]

- \(\bullet\) \(S = 0.250\) in., \(Re = 18000\)
- \(\square\) \(S = 0.125\) in., \(Re = 9000\)
- \(\triangle\) \(S = 0.063\) in., \(Re = 4500\)

\(y = 0.3\) in.

\[\left[ (T_w - T)/(T_w - T_a) \right] \times \left( \frac{y}{h} \right)^{0.07143}\]

\(y = 0.2\) in.

\(y = 0.1\) in.
Fig. 27 COEFFICIENT OF HEAT TRANSFER vs MAXIMUM VELOCITY
Fig. 28 COEFFICIENT OF TEAT THERMAL VS. DOING HEAT DISCHARGE

$\Delta S = 0.250$ in., $Re = 16000$
$\square S = 0.125$ in., $Re = 9000$
$\triangle S = 0.063$ in., $Re = 4500$

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\[ \text{Nu} = 0.71 \left( \frac{\text{Re}}{\text{X}_{1/S}} \right)^{0.8} \left( \frac{\text{X}_{1/S}}{\text{S}} \right)^{-0.61}, \quad \text{Range:} \quad \frac{30}{4500} < \frac{\text{X}_{1/S}}{\text{S}} < 400 \leq 18000 \]

- \( \text{S} = 0.250 \text{ in.}, \text{Re} = 18000 \)
- \( \text{S} = 0.125 \text{ in.}, \text{Re} = 9000 \)
- \( \text{S} = 0.063 \text{ in.}, \text{Re} = 4500 \)

**Fig. 29** HEAT TRANSFER CORRELATION

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Fig. 30 COMPARISON OF HEAT TRANSFER CORRELATIONS

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<tr>
<th>Year</th>
<th>Event</th>
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<tr>
<td>1941</td>
<td>Born in A. P., India on July 13</td>
</tr>
<tr>
<td>1956</td>
<td>Completed high school education at the United Lutheran Church Mission high school, A. P., India</td>
</tr>
<tr>
<td>1958</td>
<td>Completed Intermediate Science Course at the A.M.A.L. college, A. P., India</td>
</tr>
<tr>
<td>1962</td>
<td>Received the degree of Bachelor of Engineering (with Honours) in Mechanical Engineering from Andhra University, Waltair, India. Employed in the design organisation of Bhilai Steel Plant, India</td>
</tr>
<tr>
<td>1968</td>
<td>Currently a candidate for the degree of Master of Applied Science in Mechanical Engineering at the University of Windsor, Canada</td>
</tr>
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