Heat Transfer Behavior with Convex Target Surfaces

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The heat transfer resulting from the impingement of a heated circular jet on a cylinder was experimentally studied. Nusselt numbers at the stagnation point were found to be similar to values obtained with flat plate targets when impingement distances were less than 12 nozzle diameters. Nusselt number distributions along the longitudinal axis of the cylinder were found to behave in a manner similar to distributions on a flat plate target. Distributions around the circumference of the cylinder decayed at a very much faster rate. Local Nusselt number behavior in a region very near the stagnation point was independent of target geometry. Average Nusselt numbers were calculated and found to be strong functions of jet Reynolds number, axial impingement distance, and the averaging area.

Nomenclature

d	= exit diameter of impinging jet
D	= diameter of target cylinder
h	= local heat transfer coefficient
h_0	= heat transfer coefficient at the stagnation point
$\bar{h}$	= spaced averaged heat transfer coefficient
k	= thermal conductivity
M	= exit Mach number of jet
Nu	= local Nusselt number (hd/k)
Nu_0	= stagnation point Nusselt number (h_0d/k)
$\overline{Nu}$	= average Nusselt number (hd/k)
q''	= heat flux at cylinder surface
Re	= exit Reynolds number of jet (U_ed/ν)
S	= distance along cylinder surface from stagnation point
T_e	= jet temperature at nozzle exit
T_s	= temperature at cylinder surface
U_e	= velocity of jet at nozzle exit
X	= distance between nozzle exit and forward stagnation point on target cylinder
ν	= kinematic viscosity

Introduction

IMPINGING jets are widely used for heating and cooling solid surfaces because of their high convective heat transfer rates. Thus, many studies have been conducted to better understand impingement heat transfer. Some of the initial work in jet impingement and the resultant heat transfer can be traced back to Freidman and Mueller[1] and Vickers.[2] The utility of their results was limited, however, since the investigations were only for low-velocity laminar flow.

In the 1960s and early 1970s, a flurry of work was performed on impingement heat transfer involving flat plate targets. The main reason for this activity was the development of a miniature heat flux transducer by Gardon[3] in 1960. The flat plate geometry was selected primarily for its geometric simplicity. An excellent summary of these investigations has been prepared by Livingood and Hrycak.[4]

The problem of a jet impinging on a curved surface has not been studied as thoroughly. Recent advancements, particularly in the field of turbomachinery, have now made it necessary to investigate such geometries and the resultant heat transfer. The only recent investigations with curved target surfaces appear to be for circular jets impinging on hemispherical shells[5,6] or slot jets impinging on cylinders.[7,8] The geometries in these studies were chosen because of their two-dimensional nature.

Circular jets impinging on cylinders is another geometry of interest and, at the present time, little comprehensive flowfield or heat transfer data for this configuration exist. Also, impinging jets that are heated have not been studied as frequently. It was the purpose of this research program to experimentally investigate the three-dimensional configuration of a heated circular jet impinging on a cylinder and provide correlations for the resultant heat transfer.

Experimental Apparatus and Procedure

All experiments were performed in the impingement heat transfer facility at Texas A. & M. University; they are shown schematically in Fig. 1. The target was a polished aluminium cylinder having an outside diameter of 165.8 mm. The cylinder was hollow and water was circulated through it to keep the surface temperature constant. Cylinder wall thickness was 6.3 mm. Heated jets were obtained by passing compressed air through two 12.5 kW heaters, a 1.25 m section of straight insulated pipe for flow alignment, and exhausted through interchangeable ASME flow nozzles. Three nozzles were used having exit diameters of 12.7, 17.5, and 25.4 mm, respectively. Range of test parameters were

$$32{,}000 < Re < 160{,}000$$
$$7 < X/d < 25$$
$$80 < T_e < 190 \quad (^\circ\mathrm{C})$$
$$0.1 < M < 0.7$$

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The Reynolds numbers selected insured that the jets were turbulent and transition effects would not have to be dealt with. Axial measurement locations were selected to span the range studied by Potts[9] in his flowfield investigation of a jet impinging on a cylinder. As suggested by Obot et al.,[10] heat transfer measurements were not made with an impingement distance less than seven nozzle diameters to avoid nozzle geometry influences on the impinging flowfield. This would increase the utility of any results by allowing correlations to be independent of nozzle characteristics. The temperature range selected was primarily dictated by the maximum capacity of the heaters used.

Total temperature profiles within the impinging jet were measured using a TSI-model 1040 temperature and switching module along with a TSI-model 1220-20 hot-film probe configured as a resistance thermometer. A standard round-nose pitot probe in conjunction with water and mercury manometers was used to obtain total pressure profiles. Mean velocity profiles within the jet were obtained by combining data from the pressure and temperature profiles.

Local values of heat flux on the target cylinder were measured using four RdF-model 27036-3 heat flux gages mounted on the cylinder surface. The gages were nominally 0.63×10 5×0.07 cm in size. This particular gage also incorporated a T-type thermocouple to enable the measurement of surface temperatures. The effective probe volume for this sensor is $0.4 \times 0.2 \times 0.07$ cm. These sensors are self-generating transducers that require no special wiring, reference junctions, or signal conditioning. Readout is accomplished by connecting the sensors to any direct reading microvoltmeter. Each sensor is carefully calibrated after manufacturing and is supplied with its own calibration factor.

The four heat flux gages used were mounted 5.1 cm apart along the longitudinal axis of the cylinder. To obtain data points vertically between the gages, the entire cylinder could be moved relative to the jet centerline by placing shims underneath it. Four different shim heights were used, and this resulted in vertical distances between heat flux data points of 1.27 cm. For each shim height, the cylinder was rotated about its axis in increments of 10 deg from 0 to 170 deg, as measured from the stagnation point. Data were recorded when the cylinder surface temperatures measured by the heat flux gage thermocouples stabilized. Corrections for the gage thermal resistance were applied to all measured flux data-utilizing factory-supplied calibration data.

The heat transfer mechanism for air jets impinging on solid surfaces is clearly a combination of free convection, forced convection, and radiation. However, for the jet velocities and temperatures used in this study, it was assumed that the dominant heat transfer mechanism was forced convection. The governing equation would then be

$$q'' = h(T_{\text{ref}} - T_s) \tag{1}$$

Output from the heat flux gages resulted in surface heat flux and surface temperature being known at specific locations on the target cylinder. To determine local heat transfer coefficients, which was one objective of this study, a reference temperature (T_{ref}) must also be known. The selection of a reference temperature in convective heat transfer problems is somewhat arbitrary and typically depends on operating conditions and geometry. When heated impinging jets are used, it has been shown[11,12] that the jet recovery temperature is the best parameter to use as the reference temperature in Eq. (1). Although there appears to be no rigorous basis for this selection, it has given satisfactory results in studies involving jets impinging on flat plates.

In this study, it was assumed that the jet recovery temperature and centerline stagnation temperature were equal at a given downstream location (i.e., target location). This assumption was made based on the range of Mach numbers studied. Justification for using the jet centerline temperature as the fluid temperature over the entire impingement region is given by Striegl and Diller[13] and Hollworth and Gero.[11] Both of these investigations showed that a target "sees" a fairly extensive region ($S/d < 8$) where the effective fluid temperature is uniform.

Previous work by other investigators revealed that impingement heat transfer is effected by the exit Reynolds number of the impinging jet, along with the distance between the jet exit and point of impingement. If a heated jet is used, there is additional interaction between the jet and surrounding fluid due to temperature gradients that can significantly affect the amount of heat transferred. Although these findings were primarily based on results from jets impinging on flat plates, it was assumed they would continue to be important for jets impinging on other target geometries. Dimensional analysis also revealed the ratio of the jet nozzle diameter to the cylinder diameter as being significant for jets impinging on circular cylinders. The heat transfer resulting from a circular jet impinging on a cylinder was studied in this investigation by experimentally observing the effect the jet Reynolds number, jet exit temperature, impingement distance, and diameter ratio all had on heat transfer.

Experimental Results and Analysis

Stagnation point values of the Nusselt number as a function of impingement distance were obtained. It was found that the Nusselt number at the stagnation point increased with increasing Reynolds number and decreased as the jet was moved further away from the target cylinder. Both of these findings were as expected after reviewing the literature.

It was also found that the stagnation point Nusselt number did not vary with a change in jet exit temperature. The maximum temperature difference between the impinging jet and ambient (150°C) was apparently too small to create density gradients sufficient to bring out variations in the momentum transfer coefficients. This resulted in velocity profiles, and thus heat transfer rates, that were independent of exit temperature.

Perhaps the most interesting observation was that for a given Reynolds number, the stagnation point value of the Nusselt number appeared to be independent of the jet exit diameter to cylinder diameter ratio (d/D). This was found to be true regardless of the impinging jet Reynolds number or the distance between the nozzle exit and target cylinder. It is possible that no apparent Nusselt-number dependence on the diameter ratio only means that the range of nozzle diameters used was not great enough to influence any heat transfer behavior. Quantitatively, the change from the smallest nozzle (1.27 cm) to the largest nozzle (2.54 cm) was less than 8% of the target cylinder diameter. Further study on this point is required before any conclusion can be drawn.

Based on the foregoing observations, the stagnation point Nusselt number was represented by a same-power law function of the Reynolds number and the impingement distance. The relationship obtained from a least-squares curve fit of the measured data was

$$Nu_0 = 2.48 Re^{0.74} (X/d)^{-1.26} \tag{2}$$

Statistically, all of the data correlated to within 9% of this equation.

This correlation compares remarkably well with others published for flat plate studies. A correlation given by Vallis et al.[14] incorporating work over a similar Reynolds number and axial distance range is plotted in Fig. 2 along with Eq. (2). It can be seen that beyond an axial distance of about 12 nozzle diameters, the two correlations start to vary significantly, with the Nusselt number decreasing faster for the cylindrical target. This suggests that with curved target surfaces, there is an influence of the curvature on the stagnation point heat transfer that becomes more pronounced at larger impingement distances.

This target geometry influence can be attributed to the behavior of the flow in the deflection region. In the immediate

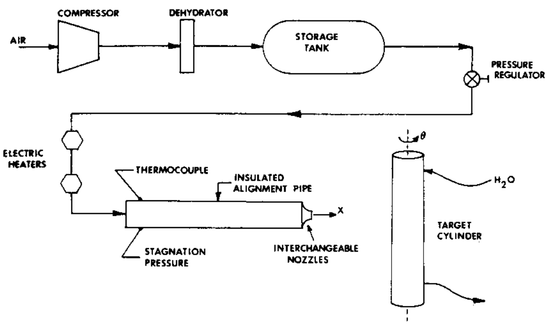

Fig. 1 Schematic of experimental facility.

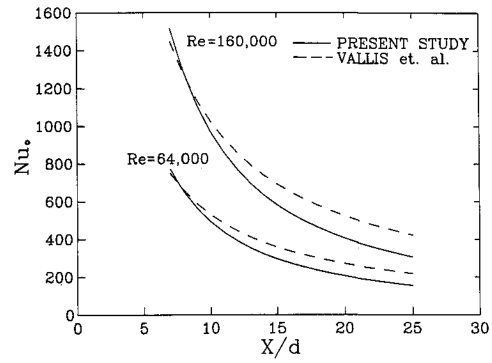

Fig. 2 Stagnation point Nusselt number.

vicinity around the stagnation point, the altered flow direction is essentially at right angles to the impinging jet centerline. This is true regardless of the shape of the target surface as long as the local diameter of the jet hitting the target is small. Under such conditions, the heat transfer at the stagnation point should also be independent of the target shape.

Increasing the distance between the nozzle and target increases the local diameter of the jet striking the target. If the local diameter of the impinging jet increases, the stagnation point actually becomes more of a stagnation region. This means that with a cylindrical target, some of the flow near the outer edges of the impinging jet would not have to be deflected a full 90 deg. Heat within the impinging jet could then be transported further around the cylinder and away from the stagnation point. If the local diameter were large enough, it is possible that heat within the jet would be carried right on past the target cylinder. With a flat plate target, however, all of the flow is turned 90 deg. This would result in the heat not being advected away from the stagnation point.

The majority of previous impingement heat transfer studies have been for two-dimensional geometries. The highly three-dimensional nature of the flowfield studied here complicated attempts at any type of comparison with these studies. Only sections of the local Nusselt number distributions that exhibited pseudo-two-dimensional behavior could be compared with any degree of certainty.

Areas of possible two-dimensional flow behavior on the target cylinder were along the longitudinal axis and around the circumference. It was assumed that the velocity and temperature fields along the longitudinal axis of the cylinder would be comparable to a jet impinging on a flat plate. Similarly, the velocity and temperature fields around the circumference of the cylinder would be comparable to those resulting from a circular jet impinging on a convex hemisphere.

Substantial justification for these assumptions was obtained by Potts[9] who measured the wall jet characteristics for a circular jet impinging on a cylinder. His measured velocity profiles and turbulence intensities along the longitudinal axis of the cylinder compared favorably to those found on a flat plate. He also found that velocity profiles and turbulence intensities around the circumference of the target cylinder were similar to those on the convex surface of spherical targets.

Further justification was provided in the present study by measuring surface pressures on the target cylinder. The measured pressure distributions along the axis compared very well with results Hrycak et al.[15] obtained with flat plate targets. Distributions around the circumference compared very well with the results Chan[16] obtained with spherical targets.

A fair amount of work thus supported the comparison of certain velocity profiles, turbulence intensities, and pressure distributions from jet impingment on a cylinder with the appropriate two-dimensional studies. Local Nusselt number distributions were then compared in a similar manner.

Local Nusselt number distributions around the circumference were found to decrease as the surface distance from the stagnation point increased, and all curves appeared to asymptote to zero. Overall, the profiles decreased and became flatter as the impingement distance was increased. Beyond a circumferential surface distance of about 2.5 to 3 nozzle diameters, the local Nusselt number distributions were independent of the impingement distance. This behavior was thought to indicate the existence of a true wall jet at these locations.

The local Nusselt number distributions along the longitudinal axis of the cylinder exhibited trends similar to those found around the circumference. One major difference between the two is that along the axis, the Nusselt numbers did not decrease as rapidly as they did around the circumference. This can be explained if it is assumed that heat transfer is proportional to wall jet turbulence intensity. Thomann[17] has shown that convex curvature in the streamwise direction has a stabilizing effect on the boundary layer adjacent to the wall. This stabilization prevents the mixing of fluid and thus decreases the transport of heat.

It was found that with the appropriate normalization, the local heat transfer distributions just discussed could be reduced to yield curves independent of jet Reynolds number and jet exit to cylinder diameter ratio. Local Nusselt numbers were normalized against the stagnation point Nusselt number, and surface distance from the stagnation point was normalized by the impingement distance. Representative results are shown in Fig 3. Correlations for these normalized curves were obtained using a least-squares curve fit of the measured data. Along the longitudinal axis, the data fit the following expression to within 8%

$$Nu/Nu_0 = 1.00 - 2.36(S/X) + 2.41(S/X)^2 - 0.89(S/X)^3 \quad (3)$$

Around the circumference, the data fit (to within 6%)

$$Nu/Nu_0 = 1.00 - 3.15(S/X) + 3.42(S/X)^2 - 1.25(S/X)^3 \quad (4)$$

Both of these correlations, along with results from other investigations involving flat plate targets, are shown in Fig. 4 for comparison. Again, it is seen that around the circumference, the Nusselt numbers decreased at a faster rate than they did along the axis. Closer comparison also reveals that close to the stagnation point, both distributions decreased at rates similar to those obtained from impingement on a flat plate impingement. This further supports the hypothesis that flow near the stagnation point behaves in a manner independent of the target geometry.

Practical engineering calculations usually require space-averaged heat transfer coefficients. The key to providing useful results lies in the realization that average Nusselt numbers are dependent on the averaging area selected. Many investigators, however, simply calculate average Nusselt numbers based on one area and never study the effect different areas may have. This severely limits the usefulness of their results. Typically, their correlations do not allow for a target area to be entered as a parameter.

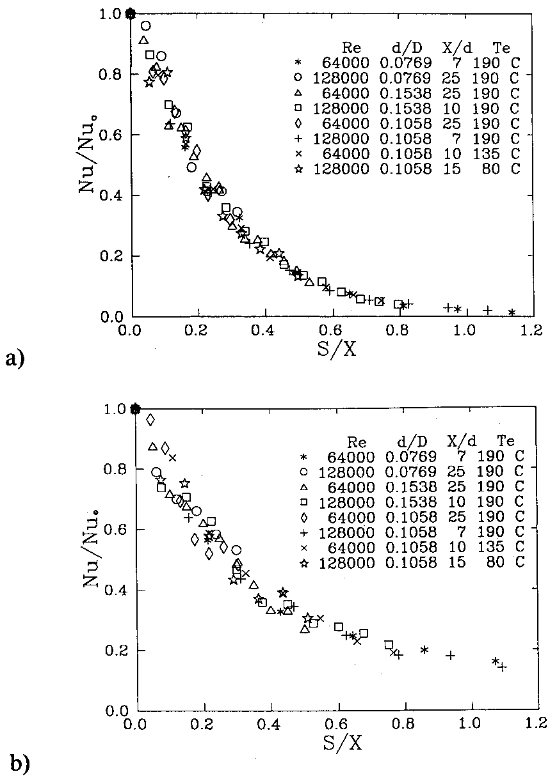

Fig. 3 Normalized Nusselt number distribution. a) Local *Nu* around circumference. b) Local *Nu* around axis.

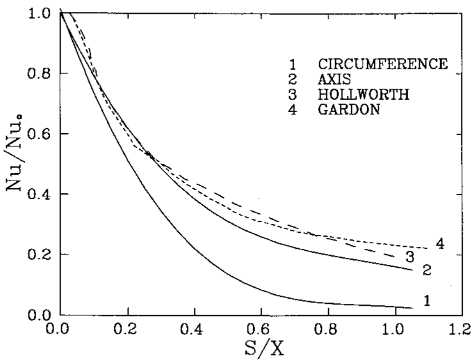

Fig. 4 Local Nusselt number correlation.

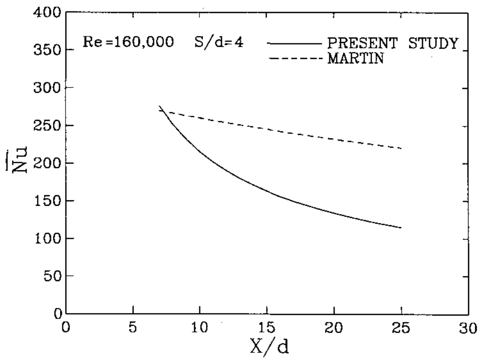

Fig. 5 Comparison of average Nusselt number correlations.

To completely investigate average Nusselt number behavior in the present study, different averaging areas were selected. An area was defined as the points enclosed by a boundary located a constant radial distance from the stagnation point. Based on the number of local data points available and the results of local Nusselt number distributions, the areas chosen for study were enclosed by constant radial distances of $S/d = 2$, 4, 6, and 8.

In general, the average Nusselt number is a function of Reynolds number, impingement distance, averaging area, and diameter ratio. It also follows that any correlation obtained for the average Nusselt number should have some identifiable trends. For example, in the limit, as the target cylinder diameter approaches some large value, the resultant geometry would be similar to a jet impinging on a flat plate. Therefore, any correlation obtained should also exhibit this behavior and approach a flat plate correlation as d/D approaches zero.

Based on these ideas and correlations previously published for flat plate impingement, the average Nusselt number data were correlated in a form

$$\overline{Nu} = C_1 Re^a (X/d)^b (S/d)^{c(d/D)} \tag{5}$$

As shown, the exponent on the area ratio was expected to be some function of the diameter ratio. As the diameter ratio approaches zero (e.g., large cylinder diameter), this exponent would be expected to approach the value obtained for a flat plate correlation.

Values were fit to Eq. (5) and the following expression was obtained:

$$\overline{Nu} = 0.62 Re^{0.73} (X/d)^{-0.69} (S/d)^{c(d/D)} \tag{6}$$

where

d/D	$c(d/D)$
0.0769	−0.461
0.1058	−0.478
0.1538	−0.447

It is important to notice the small difference in exponent values for the S/d term in Eq. (6). Satisfactory results, well within experimental error, could be obtained using a common value of −0.46. This suggests that the range of diameter ratios used was not great enough to influence heat transfer behavior. The analysis of the stagnation point heat transfer also led to this conclusion. It is also apparent that the influence of the averaging area term (S/d) is increased due to the convex curvature of the target. An explanation for this is found in the behavior of the local Nusselt number distributions. As discussed previously, the Nusselt numbers around the circumference of the cylinder decayed at a higher rate. Increasing the averaging area incorporated more and more of these low value Nusselt numbers into the overall calculation and thus lowered the average Nusselt number.

For comparison, Eq. (6) is plotted in Fig. 5 along with a correlation published by Martin.[18] Martin's equation is the result of compiling the correlating data from numerous flat plate investigations. Figure 5 shows that there is a strong influence of axial impingement distance on the average Nusselt number. This influence is also more pronounced with the cylindrical target. It is thought that this effect is due to the local diameter of the impinging jet. As seen with stagnation point Nusselt numbers, larger impingement distances caused heat to be advected further around the cylinder. Due to interaction between the wall jet and surrounding air, there is then less heat available to be transferred to the surface.

Conclusions

Heat transfer at the stagnation point was found to only be a function of jet Reynolds number and axial impingement distance. The range of diameter ratios (d/D) studied did not produce any observable influence. It was also found that for impingement distances less than 12 nozzle diameters, the resultant Nusselt numbers compared very well with values obtained from flat plate impingement studies. This was attributed to flow behavior near the stagnation point being independent of target geometry for those impingement distances.

Around the circumference of the cylinder, the Nusselt number distributions decayed faster than they did along the longitudinal axis. Increasing the impingement distance negated slightly this effect. The transition from a deflection region to a wall jet occurred at 3 nozzle diameters from the stagnation point along the axis and 2.5 to 3 nozzle diameters around the circumference. These locations also corresponded to where the Nusselt numbers reached a constant minimum value.

It was also shown that local flowfield and heat transfer exhibited two-dimensional behavior around the circumference and along the axis of the target cylinder. Local Nusselt numbers were found to reduce onto single curves when normalized properly. Correlations were obtained for nondimensional Nusselt number distributions along the axis and around the circumference. The correlation for distributions along the axis compared very well with others published for flat plate impingement. Nusselt number behavior in a region near the stagnation point was independent of target geometry.

Average Nusselt numbers were calculated from the local values for various surface areas. Different areas were selected to determine any effect they might have on the calculations. The average Nusselt numbers were found to be a function of jet Reynolds number, axial impingement distance, and averaging area. They were also found to be very weak functions of the diameter ratio. Convex target curvature increased the importance of both the averaging area and the axial impingement distance when calculating average values.

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