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Heat transfer between an under-expanded jet and a cylindrical surface

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Abstract

This paper presents a selection of data from an investigation that was concerned with the heat transfer which occurs when an under-expanded jet impinges onto a heated cylindrical surface. The purpose of the study was to establish the thermal boundary conditions for calculating thermal stresses in heat transfer surfaces when subjected to high-speed cleaning jets. The heat transfer in the impingement zone of a high-speed jet is extremely high and when the presence of the surface interferes with the expansion of the jet, the radial and circumferential distributions of the heat transfer coefficient become complicated. If a highly under-expanded jet impinges upon the surface while the nozzle-to-surface spacing is small, $z/D \approx 3$, there is no longer a maximum stagnation heat transfer coefficient on the geometric axis of the jet, instead a stagnation 'ring' is formed with a radius of about one nozzle diameter. A selection of data is presented that shows how, particularly for z/D less than 10, the Nusselt number distribution has a very high peak value at, or near to, the geometric stagnation point and then falls away steeply in both the axial and circumferential directions. The high values of Nusselt number, and the large differences between the peak values on the front edge of the cylinder and the values at the rear of the cylinder, could lead to very substantial differential cooling rates and hence to significant thermal stresses being generated when high pressure air cleaning jets are used on high-temperature tubes. However, when the nozzle exit is placed more than 20 nozzle diameters away from the surface of the cylinder there is a significant reduction in the maximum Nusselt number and the overall distribution is much smoother; this will alleviate potential problems from thermal stresses.

1. Introduction

The work reported in this paper was undertaken as a preliminary investigation within a wider study into the interaction between a high-speed cleaning jet (air sootblower) and high-temperature boiler tubes. Of particular concern are the thermal stresses generated in the tube material due to the intense cooling action of the jet, particularly when it is placed unusually close to the tubes. A survey of the literature quickly showed that there have been only a few studies into jet impingement onto cylindrical surfaces, and what there is has been confined to incompressible jets with low Reynolds numbers [1–5]. What was more surprising, however, was how very little work has been carried out on compressible jet impingement heat transfer onto flat surfaces. The authors therefore carried out a study, which has recently been reported [6], into the heat transfer between an under-expanded jet and a flat surface using a 12.7 mm diameter convergent nozzle, with ratios of nozzle supply pressure to ambient up to 5.08. Detailed heat transfer measurements were made for nozzle-to-surface spacings of 3, 6 and 10 nozzle diameters, with some further measurements being made for nozzle-to-surface spacings of 20, 30 and 40 nozzle diameters.

The study illustrated the complexity of the heat transfer distribution generated by an under-expanded jet impinging onto a plane surface, particularly when the spacing between the jet exit and the surface is

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Nomenclature

D	nozzle diameter	Re	Reynolds number
h	heat transfer coefficient	T_{aw}	adiabatic wall temperature
Nu	Nusselt number	T_w	wall temperature
P_0	pressure upstream of nozzle	w	jet velocity
P_∞	ambient pressure	y	axial distance on cylinder surface
q	surface heat flux	z	distance from nozzle exit plane
r	jet radius	θ	angular distance around cylinder
r_5	radius of jet where velocity is half of the maximum		

shorter than the length of the jet core. The source of the complexity is illustrated in Fig. 1. The nozzle used in both the study with plane surfaces, and the present one with cylindrical surfaces, was investigated in great detail by Gibbings et al. [7,8] who showed that the exit velocity profile was uniform, and that for a pressure

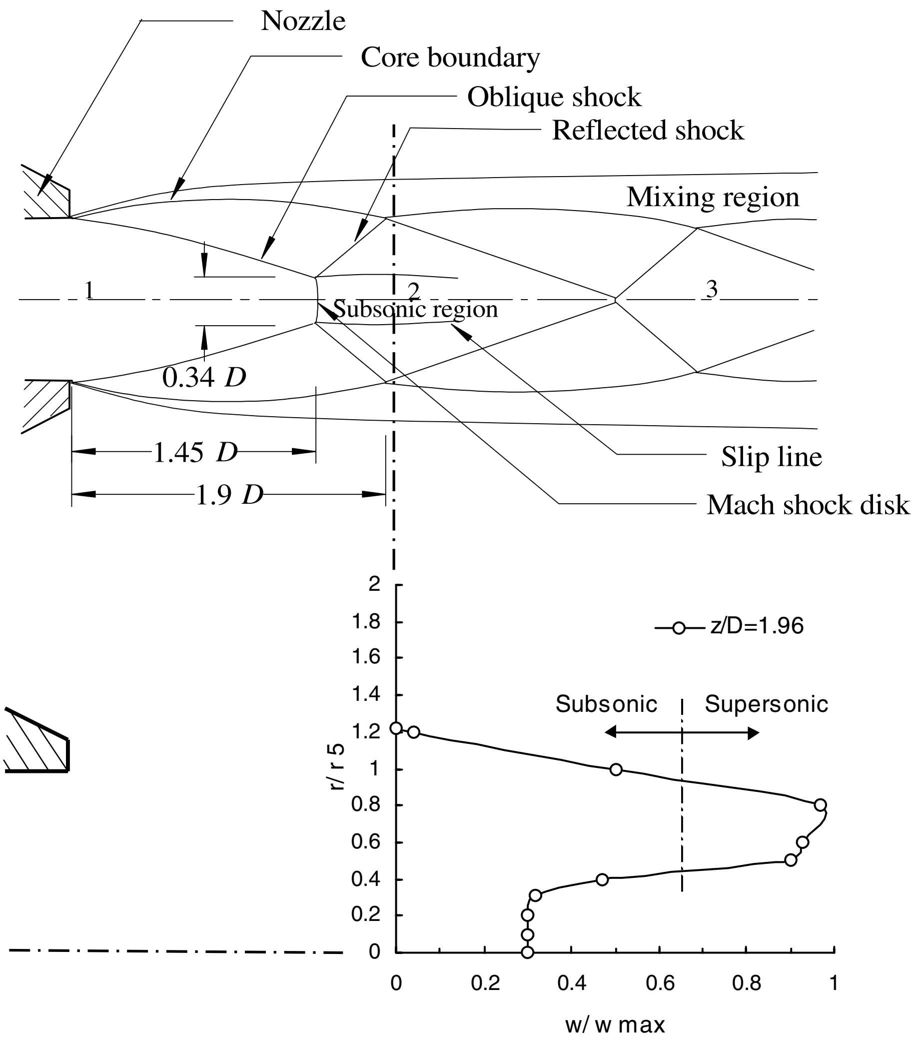

Fig. 1. Structure of the jet from a convergent nozzle [8] and an associated velocity distribution [9].

ratio of 5.15 the structure of the flow close to the nozzle was as depicted in the upper half of Fig. 1. In a different study of the compressible flow from a convergent nozzle, at a comparable pressure ratio, Donaldson et al. [9] measured the velocity profile within the jet at various axial positions. A normalised velocity profile in a highly under-expanded jet for a pressure ratio of 6.75, taken at a distance of about 2 jet diameters downstream of the jet exit, is shown in the lower half of Fig. 1. The radius is normalised in the conventional way using the radius r_5 at which the velocity drops to one-half of its maximum value. Although the pressure ratios corresponding to each figure are not identical, they are both representative of a highly under-expanded jet (as defined by [9]) and they are sufficiently close for the qualitative juxtaposition of the two figures. What these combined figures show is that for a few diameters downstream of the nozzle the core of the jet is subsonic, having gone through a disc shock closer to the nozzle, while the surrounding flow is supersonic. When a plane surface is introduced into this flow field the impingement pressure on the axis of the jet will be lower than that in the surrounding annulus. There will, therefore, be a region where the flow on the surface is radially inwards towards the central axis and the stagnation point no longer exists at the geometric axis of the jet, but as a ring. A more detailed study of this phenomenon, and its consequence for heat transfer on a plane surface can be found in [6]. The present paper is concerned with the heat transfer between the same convergent nozzle, operating under similar conditions, but impinging upon a cylindrical surface. Experiments were performed with the jet impinging onto a 50 mm diameter cylinder, initially with the jet axis being aligned with the diameter of the cylinder, but experiments were also conducted with the jet offset from the cylinder axis, and with a row of closely packed cylinders to represent the tubed wall of a boiler. A large amount of data was collected over a range of conditions. However, for the sake of brevity and clarity, this paper is confined to the case where the jet is arranged so that its axis is normal to the cylinder axis, and passes through it, as shown in Fig. 2. Furthermore, only a sample of the data is presented to illustrate the distribution and magnitude of the heat transfer coefficient, and to help explain the physics of the flow. For further details the reader is referred to [10].

When a high-speed jet impinges onto an unheated adiabatic surface there may be a localised cooling of the surface due to the reduced static temperature within the flow and this has to be allowed for in the measurement and definition of the heat transfer coefficient. Some of the earliest measurements of heat transfer to an impinging jet were made by Gardon and Cobonpue [11] who recognised that heat transfer coefficient should be defined in terms of the temperature difference between

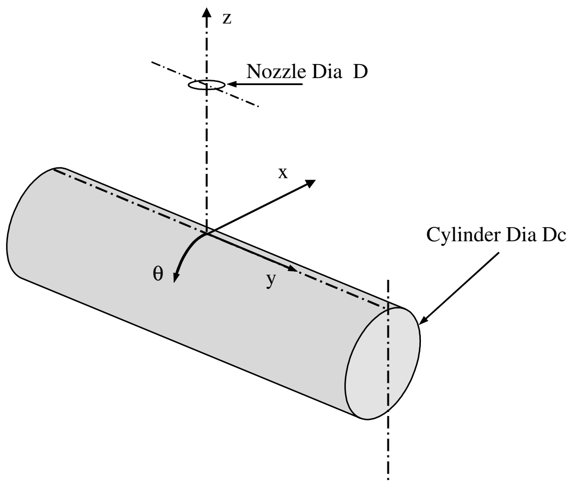

Fig. 2. Cylinder geometry.

the impingement plate temperature, and the adiabatic wall temperature. Thus:

$$h = \frac{q}{(T_w - T_{aw})} \tag{1}$$

2. Experimental apparatus and procedure

The 12.7 mm diameter convergent nozzle was held vertically above the cylinder, which was mounted on a three-dimensional traverse (y and z; x was used to offset the jet and is not relevant to the results presented in this paper). The cylinder could also be rotated about its own axis (θ), see Fig. 2. The temperature and pressure of the air were measured just upstream of the nozzle by a calibrated pressure gauge and a resistance thermometer with uncertainties of ±0.1 bar and ±0.5 °C respectively. The test cylinder, Fig. 3, was 50 mm in diameter and 280 mm long and made of wood. Thirty-three K-type thermocouples were installed on the test cylinder along three lines, placed every 60° of rotation, parallel to the cylinder axis. The distance between the thermocouples was constant and equal to the nozzle outlet diameter, 12.7 mm. The thermocouples were covered by a 50.8 μm

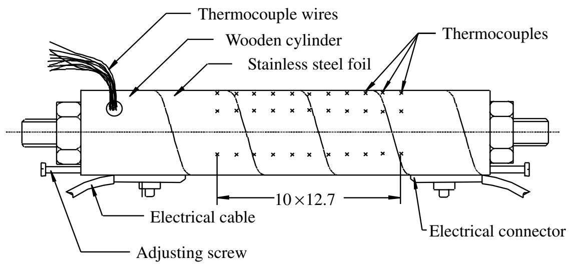

Fig. 3. Experimental cylinder.

thick strip of stainless steel foil, 660 mm long and 48 mm wide, which was spirally wrapped around the cylinder surface. The gap between adjacent turns was less than half a millimetre to reduce the discontinuity in the heating surface. The wooden cylinder was split along its diameter and adjusting screws were used to prise the two halves apart so that when the heated foil expanded, the good thermal contact between the thermocouples and the foil could be maintained.

An electrical connector was fixed at each end of the foil and heating was applied by passing an electric current through the foil. Currents and voltages up to 63 A and 13.1 V respectively were used, the greatest uncertainties in these measurements were ±1 A and ±0.1 V respectively. Because of unavoidable conduction heat loss through the wood, a detailed analysis of the heat conduction through the wooden cylinder was carried out. The greatest effect was found at the greater axial distances from the jet in the heated case (i.e. hottest temperatures) and adjacent to the impingement zone for the unheated case where the coldest temperatures were found. At these extreme conditions the greatest effect on the heat flux was found to be less than 3% and this was compensated for in the calculations of heat transfer coefficient. Calculations were also made of the lateral conduction through the foil due to temperature gradients in the foil; this was found to be negligible. A detailed uncertainty analysis showed the uncertainty in the Nusselt number to be about ±5%, with a similar uncertainty for the Reynolds numbers which were calculated at the jet exit conditions and assuming an isentropic expansion from the upstream stagnation conditions.

The jet was carefully aligned on the upper surface of the cylinder (so that the jet axis coincided with the vertical diameter of the cylinder as shown in Fig. 2). Once the distance was set, the horizontal cylinder was moved in the direction of the cylinder axis and at each axial position, y, the cylinder was rotated in 10° increments (θ). This procedure allowed the surface temperatures to be measured in both the axial and circumferential directions. Each test was carried out in two parts, once for the adiabatic wall and then again for the heated wall. No heating was applied to the foil in the first part of the experiment and the temperature distribution due to the flow field itself was recorded. Heating was then applied and the procedure repeated. Temperatures were measured by scanning the thermocouples using a data-logger once conditions had become stable and the temperature fluctuations at any thermocouple were no more than 0.1 °C.

3. Results and discussion

The Nusselt number was calculated using Eq. (1) and is based on the nozzle diameter, D, and air properties at film temperature ($0.5(T_w + T_{aw})$). A considerable number of surface temperature and heat transfer distributions were obtained for different jet pressure ratios, ranging from 1.68 to 5.08, and for nozzle-to-surface spacings, z/D, of 3, 6 and 10 with some limited data for 20, 30 and 40. Measurements were also taken for off-axis jet alignment and for a row of cylinders packed closely together. However, it was found that the highest heat transfer rates were obtained when the jet impinged on the top of the cylinder as shown in Fig. 2 and this is the reason why the paper concentrates on these results. The other results can be found in [10]. Fig. 4 is an example of the adiabatic wall temperature, the heated wall temperature, and the resultant Nusselt number distribution for the case of $z/D = 6$ and $P_0/P_\infty = 3.04$. Over most of the surface the adiabatic wall temperature, Fig. 4a, is close to ambient except in the area around the impingement zone where the high jet velocity has a low static temperature and its greatest cooling effect on the surface. The cooling effect of the jet on the heated surface is shown clearly in Fig. 4b, while the resultant Nusselt number distribution is shown in Fig. 4c. When a jet impinges upon a flat surface normal to the jet axis it spreads evenly over the surface. However when it impinges on a cylinder there is some deflection in the axial direction but the jet is able to flow around the cylinder in the circumferential direction. This circumferential flow will reduce the flow deflected along the axial direction. This flow pattern is reflected in the data shown in Fig. 4b and c. There is a sharp peak in the Nusselt number at the stagnation point, as might be expected. The cooling effect falls away in both the axial and circumferential direction as the velocity of the spreading jet decreases. A feature worthy of note is that the circumferential distribution of heat transfer coefficient is smooth and does not display the complex distribution found over a cylinder placed in a cross-flow where laminar/turbulent transition and flow separation are common occurrences, particularly at high Reynolds number (see, for example, [12]). The reason for this is that the flow from a jet impinging onto a cylinder remains attached for most of the circumference. Tabrizi [13] investigated the flow from an under-expanded jet impinging normally onto a cylinder using a series of experiments involving surface pressure measurements and a visualisation technique employing a mixture of lampblack and oil being painted onto the cylindrical surface. For a similar geometric ratios and jet pressure ratios to those under discussion here, he showed that the jet remained attached to the surface of the cylinder for most of the semi-circumference and a separation zone of just ±15° was formed at the rear of the cylinder, compared with the usual ±60° for a cylinder in cross-flow at high Reynolds number (e.g. [14]). The attachment of jets to curved surfaces has received considerable attention in relation to aircraft wings and jet flap devices (e.g. [15]) and at higher ve-

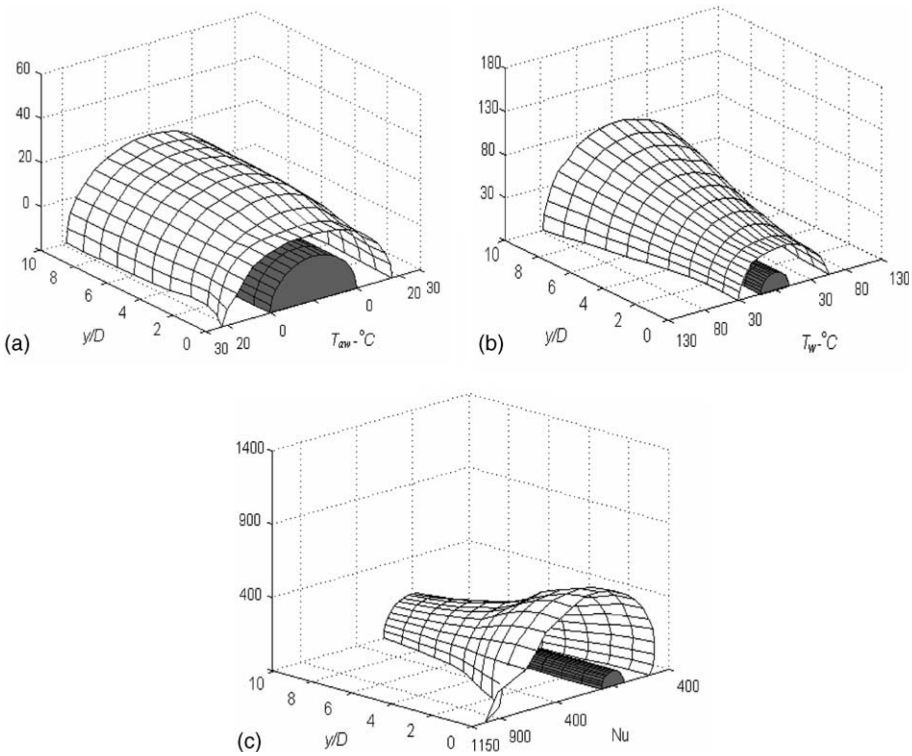

Fig. 4. Temperature and Nusselt number distributions on cylinder surface $z/D = 6$, $P_0/P_\infty = 3.04$, $Re = 615,000$. (a) Adiabatic wall temperature, (b) heated wall temperature and (c) Nusselt number distribution.

locities in relation to Coanda gas flares used on petrochemical plants [16]. In these studies, however, the jets are from slot nozzles and are two dimensional, unlike the circular nozzle of the present work which produces a highly three-dimensional flow.

The study carried out by the authors into the impingement of an under-expanded jet onto a plane surface [6] showed the complexity in the heat transfer distribution due to the interaction of the jet core with the surface. A similar complexity is found when the highly under-expanded jet impinges onto the cylinder with small nozzle-to-surface spacings. Fig. 5 shows the Nusselt number distribution over the cylinder for $z/D = 3$ and $P_0/P_\infty = 5.08$. The peak in the heat transfer coefficient is not on the jet axis but is displaced from it at $y/D \approx 1$ in the axial direction and at an angle of about 20° in the circumferential direction. This is due to the phenomenon discussed earlier in relation to the jet flow characteristics depicted in Fig. 1. On a flat surface this jet/surface interaction leads to a higher surface pressure being generated in an annular zone than at the jet axis, and there is a radial inward flow on the surface. On the cylindrical surface the flow is no longer symmetrical in that the surface 'falls away' in the circumferential direction. Flow visualisation using grease smeared onto the surface showed the surface flow to be as depicted in

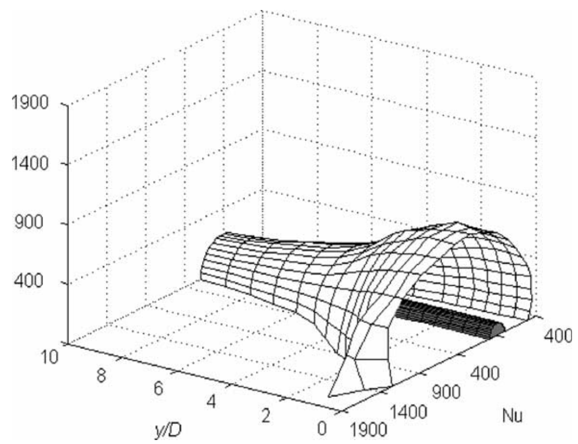

Fig. 5. Nusselt number distributions on cylinder surface $z/D = 3$, $P_0/P_\infty = 5.08$, $Re = 1,028,000$.

Fig. 6, which refers to the same conditions as in Fig. 5. In the impingement zone (indicated by the dashed circle) there is an inward flow along the top of the cylinder in the axial direction (just as there is on a flat plate) while in the circumferential direction the radial inflow is discouraged by the fact that the surface is receding in the angular direction. Thus the effect of the axial inward flow and the mainly outward flow in the circumferential direction was to leave an area of grease in a line around the circumference of the cylinder for about ±35°, as indicated in Fig. 6. Also shown in Fig. 6 are the axial and circumferential distributions of the adiabatic wall temperatures which show the higher stagnation temperatures to be away from the jet axis due to the velocity distribution shown in Fig. 1.

In Fig. 7 the nozzle is further from the cylinder at $z/D = 6$ and the jet core has decayed sufficiently that the maximum heat transfer is to be found once more on the jet axis and it has a higher value than that seen for the closer spacing. As the nozzle is moved further from the cylinder, Fig. 8, the peak heat transfer remains at the jet axis but reduces in value. These figures show the general characteristics of high heat transfer rate at the stagnation point with the values falling away quickly in both the axial and circumferential directions. In Fig. 7, where $z/D = 6$, although the peak heat transfer is at the jet axis, there is still a discontinuity in the distribution close to the stagnation point ($y/D \approx 1$) due to the jet core interacting with the surface. In Fig. 8 the cylinder is now sufficiently far from the nozzle that the jet core has decayed and the surface heat transfer distribution is smooth.

It is difficult to characterise the overall heat transfer distribution, other than to note its general form and to pick out the features resulting from the interaction of the jet core that have been described above. Recalling the original reason for carrying out this work, i.e. the thermal stresses generated in boiler tubes by the cooling action of air sootblowers, the distribution of heat transfer coefficient shows the potential for significant differential cooling and local thermal contraction. The very substantial cooling under the impingement zone at

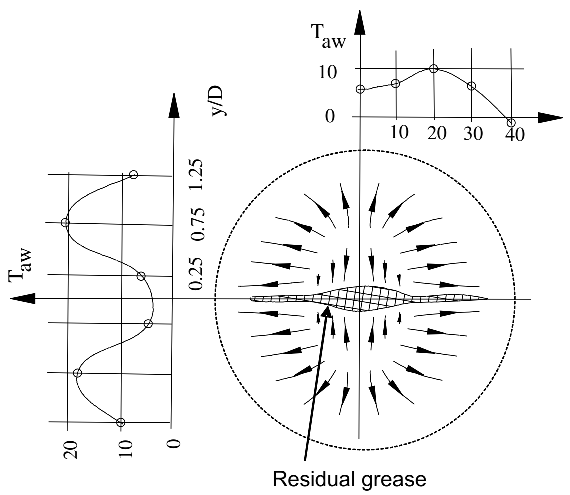

Fig. 6. Schematic representation of impingement flow pattern on surface, and measured axial and circumferential temperature distributions. $z/D = 3$, $P_0/P_\infty = 5.08$.

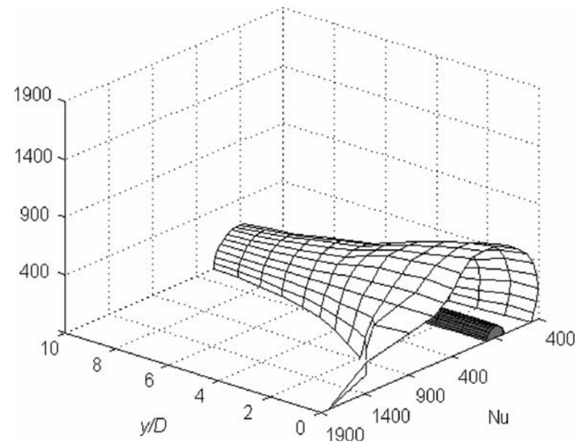

Fig. 7. Nusselt number distribution on cylinder surface $z/D = 6$, $P_0/P_\infty = 5.08$, $Re = 1,028,000$.

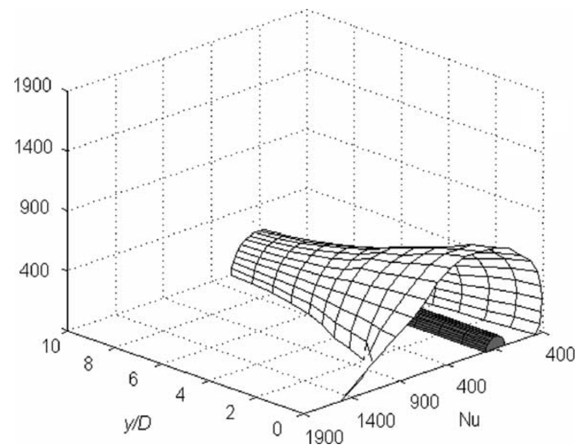

Fig. 8. Nusselt number distribution on cylinder surface $z/D = 10$, $P_0/P_\infty = 5.08$, $Re = 1,028,000$.

the front of the tube will cause high local tensile stresses and, because of the lower cooling at the rear of the cylinder, there will be a tendency for the tubes to bend. The calculation of the thermal stresses arising from the jet cooling is the subject of further investigations and will be published in due course. The magnitudes of the heat transfer coefficients are extremely high in the impingement zone and they reduce quickly away from the impingement point, and as the nozzle is moved further away from the surface. Fig. 9 shows how the maximum Nusselt number (which for high pressures and small spacings will not be on the jet axis) reduces as the nozzle is moved further from the surface. Also shown in this figure are the corresponding maximum values of Nusselt number for jet impingement onto a flat surface. The values are similar, which is not unexpected, and the slight differences can be accounted for by experimental

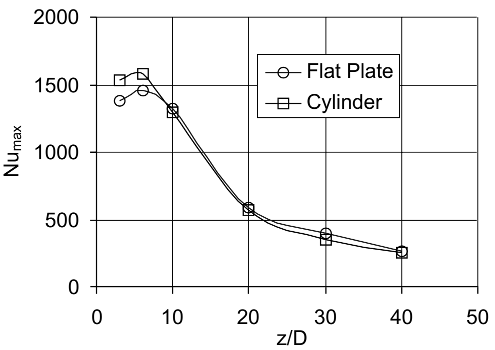

Fig. 9. Maximum Nusselt number on surface due to impinging jet, $P_0/P_\infty = 4.40$.

uncertainty in each value (±5%) and the fact that the curved surface will have some effect on the impinging flow on the jet axis. It is interesting that the maximum value for the spacing $z/D = 6$ is higher that that for the closer spacing of $z/D = 3$. This is because of the subsonic central region with the closer spacing, and it should be borne in mind that for the closer spacing the maximum heat transfer coefficient occurs in an annular region around the jet axis so a greater surface area, and hence heat transfer, will be involved.

The cooling intensity is also affected by the driving pressure of the jet. Fig. 10 shows how the maximum value of the Nusselt number increases with increasing pressure ratio. Taking Figs. 9 and 10 together it can be seen that a combination of short nozzle-to-surface spacings and high pressures leads to significant cooling, as would be expected, but as the nozzle is moved away to $z/D = 20$ the cooling rate reduces significantly. This has practical implications for the design of sootblowing systems which, conventionally, have nozzles with typical values of z/D of about 25. However, should the plant designers seek to move the nozzle closer to the tubes, for greater compactness of plant, or increased intensity of cleaning, the potential for generating thermal stresses

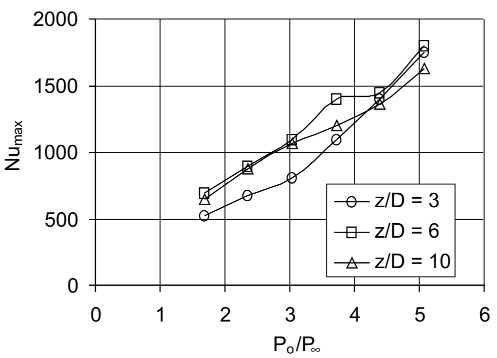

Fig. 10. Maximum Nusselt number on surface due to impinging jet.

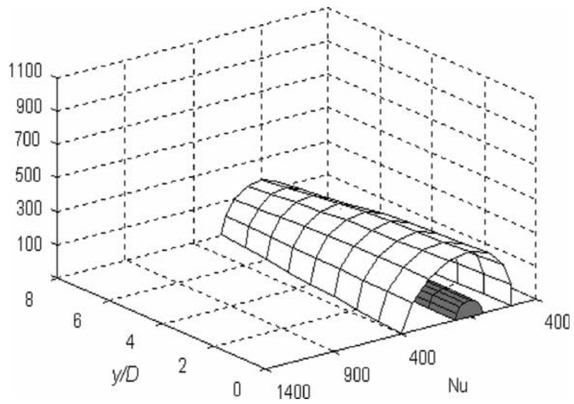

Fig. 11. Nusselt number distribution on cylinder surface $z/D = 30$, $P_0/P_\infty = 4.40$, $Re = 890,000$.

will increase greatly. Fig. 11 shows the heat transfer distribution around the cylinder for $z/D = 30$ and the greater uniformity, in all directions, is clearly seen. This implies that the cooling effects of the air jets from nozzles placed at this distance from the tubes, although significant, will be reasonably uniform and less likely to lead to problems from thermal stresses.

4. Conclusions

The heat transfer between a highly under-expanded jet and a surface is known to be more complicated when the surface is placed within the development length of the jet core. This paper shows how this complexity manifests itself when the surface is a cylinder placed so that its axis is aligned with the jet axis and normal to it. A selection of data has been presented that shows how, particularly for z/D less than 10, the Nusselt number distribution has a very high peak value at, or near to, the geometric stagnation point and then falls away steeply in both the axial and circumferential directions. The jet flow from the centre of the nozzle remains attached to the cylinder's surface for about 165° and this leads to a smooth circumferential Nusselt number distribution, unlike that found for the more common situation of a cylinder in cross-flow. The high values of Nusselt number, and the large differences between the peak values on the front edge of the cylinder and the values at the rear of the cylinder, could lead to very substantial differential cooling rates and hence to significant thermal stresses being generated when high pressure air cleaning jets are used on high-temperature tubes. However, when the nozzle exit is placed more than 20 nozzle diameters away from the surface of the cylinder there is a significant reduction in the maximum Nusselt number and the overall distribution is much smoother; this will alleviate potential problems from thermal stresses.

References

[1] C. Gau, C.M. Chung, Surface curvature effect on slot-air jet impingement cooling flow and heat transfer process, ASME J. Heat Transfer 113 (1991) 858–864.

[2] T. Pekdemir, T.W. Davies, Mass transfer from stationary cylinders in a submerged slot jet of air, Int. J. Heat Mass Transfer 41 (1983) 2361–2370.

[3] M. Kumada, I. Mabuchi, Y. Kawashima, Mass transfer on a cylinder in the potential core region of a two-dimensional jet, Heat Transfer—Japanese Res. 2 (3) (1973) 53–66.

[4] E.M. Sparrow, C.A.C. Altemani, A. Chaboki, Jet impingement heat transfer for a circular jet impinging in crossflow on a cylinder, ASME J. Heat Transfer 106 (1984) 570–577.

[5] A.A. Tawfek, Heat transfer due to a round jet impinging normal to a circular cylinder, Heat Mass Transfer 35 (1999) 327–333.

[6] M. Rahimi, I. Owen, J. Mistry, Impingement heat transfer in an under-expanded axisymmetric air jet, Int. J. Heat Mass Transfer 46 (2003) 263–272.

[7] J.C. Gibbings, The combination of a contraction with a supersonic nozzle for a wind tunnel, Ingenieur—Archiv 35 (4) (1966) 269–275.

[8] J.C. Gibbings, J. Ingham, D. Johnson, Flow in a supersonic jet expanding from a convergent nozzle, ARC CP no. 1197, July 1968.

[9] C.D. Donaldson, R.S. Snedeker, D.P. Margolis, A study of free jet impingement. Part 1: mean properties of free and impinging jets, J. Fluid Mech. 45 (2) (1970) 281–323.

[10] M. Rahimi, Heat transfer and thermal stress distribution due to the impact of a high-speed jet on a hot surface, Ph.D. thesis, The University of Liverpool, UK, 1996.

[11] R. Gardon, J. Cobonpue, Heat transfer between a flat plate and jets of air impinging on it, in: International Developments in Heat Transfer, ASME, New York, 1963, pp. 454–460.

[12] F.P. Incropera, D.P. DeWitt, Fundamentals of Heat and Mass Transfer, fourth ed., Wiley, 1996, p. 369.

[13] S.P.A. Tabrizi, Jet Impingement onto a circular cylinder, Ph.D. thesis, The University of Liverpool, UK, 1996.

[14] B.R. Munson, D.F. Young, T.H. Okiishi, Fundamentals of Fluid Mechanics, Wiley, 1990, p. 365.

[15] D.J. Wilson, R.J. Goldstein, Turbulent wall jets with cylindrical streamwise surface curvature, J. Fluid Eng. 98 (1976) 550–557.

[16] D.G. Gregory-Smith, The discharge from a thin slot over a surface of convex curvature, Int. J. Mech. Sci. 24 (6) (1982) 329–339.