AN EXPERIMENTAL HEAT TRANSFER STUDY OF A COLD JET IMPINGING ON A HOT SURFACE

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Abstract. *This paper investigates the influence of some governing parameters on the heat transfer characteristics of a circular impinging jet on a heated flat plate. Measurements of local pressure and wall temperature distributions are presented as well as temperature profiles. The experiments were conducted at various nozzle-to-plate spaces and Reynolds number of 35,000. The constant wall heat flux condition is achieved by conducting electricity through thin resistors that are placed beneath an aluminum disk. Temperature was measured using thermocouples.*

Keywords. *Turbulence, jet, impingement, Nusselt number.*

1. Introduction

The flow of a cold jet impinging on a hot surface is a problem that has attracted a lot of interest due to its technological importance. Typical applications are the tempering and shaping of glass, the annealing of plastic and metal sheets, the drying of textile and paper products, the deicing of aircraft systems and the cooling of heated components in gas turbine engines and electronic instruments. Although the number of works devoted to this subject over the past thirty years has been consistently high; the recent times have witnessed a considerable increase in the number of published articles. Because of the high complexity of the flow structure resulting from an impinging jet, many recent works have been published which study specific aspects of the problem. The generation of vortices at the entrance nozzle, the entrainment of fluid from the surround medium, the interaction with confining walls, the flow separation, along with many other complicating factors make the delivery of a single and complete theory difficult to obtain.

Thus, because of the several important aspects associated with the problem, studies have concentrated on characterizing different features of the phenomenon. This will be reviewed in the next section. However, we can anticipate here that most of the works that deal with theoretical analyses resort to asymptotic techniques and similarity concepts. These studies normally solve the velocity and the temperature fields in regions about but not at the stagnation point.

In fact, a question that has been the object of many investigations is the behavior of the heat transfer coefficient at the stagnation point (Lee and Lee (1999,2000), Kendoush (1998), Nishino et al. (1996)). If the Reynolds number is low enough so that the flow can be rendered laminar, then asymptotic methods can be used to find analytical solutions in all flow regions except at the stagnation point, where a strong singularity is present. Thus, even for this simple flow condition, calculation of the heat transfer coefficient at the stagnation point is very difficult to achieve.

For turbulent flows, the correct description of the flow field is greatly complicated by the necessary specification of turbulence models that can capture all relevant characteristics of the problem. Frequently, turbulence models of the eddy viscosity type are used together with some heat transfer analogy consideration for the description of the temperature field (see, e.g., Behnia et al. (1998, 1999), Gibson e Harper (1997)). This leads to a serious difficulty at the stagnation point where the Reynolds analogy between eddy-diffusivity and eddy-viscosity breaks down. Indeed, when the equations of motion are integrated to the wall and the hypothesis of a constant turbulent Prandtl number is used, the calculated heat transfer rates at the stagnation point are observed to exceed by far the actual values.

Despite the critics of many researchers, the use of wall functions to by-pass the difficulties involved with the modeling of low Reynolds number turbulence is still an attractive means to solve problems in a simple way. Cruz and Silva Freire (1998) have proposed an alternative approach where new wall functions are used to describe the velocity and temperature fields in the wall logarithmic region. As the stagnation point is approached, these functions reduce to power-law solutions recovering Stratford's solution. The paper of Cruz and Silva Freire resorted to Kaplun limits for an asymptotic representation of the velocity and temperature fields. Results were presented for the asymptotic structure of the flow and for the skin-friction coefficient and Stanton number at the wall.

The objective of the present research is to investigate the influence of some governing parameters on the heat transfer characteristics of a circular impinging jet on a heated flat plate. Measurements of local pressure and wall temperature distributions will be presented as well as temperature profiles. The experiments were conducted at various nozzle-to-plate spaces and Reynolds number of 35,000. The constant wall heat flux condition is achieved by conducting electricity through thin resistors that are placed beneath an aluminum disk. Temperature was measured using thermocouples.

2. Short literature review

In the early nineties Özdemir and Whitelaw (1992) studied the problem emphasizing the large-scale transport of temperature by spatially coherent structures. These authors showed that oblique impingement introduced vertical velocities that rendered the boundary layer equation inapplicable and resulted in a flow structure with strong azimuthally dependence. The large structures improved the transport of temperature but led to an inactive zone near the vortex center.

Fox et al. (1993) also studied the influence of vortical structures on the temperature field of jets. Depending on the distance of the jet nozzle to an adiabatic wall, secondary vortical structures were observed that could result in a region of lower wall temperature. Thus, it is the competition that is established between the vortex rings formed at the jet periphery and the secondary vortices resulting from the impingement that determines the wall temperature.

Cooper et al. (1993) and Craft et al. (1993) in two companion papers studied turbulent jets impinging orthogonally onto a plane surface. Cooper et al. reported an extensive set of measurements on the flow field; data for the mean velocity profile in the vicinity of the plate surface and also for the three Reynolds stress components lying in the *x-r* plane were presented. These data were used by Craft et al. to examine the performance of four different turbulence models: the κ–ε model and three-second order moment closures. The predictions obtained through the κ–ε model and one second order moment closure result in far too high turbulence levels near the stagnation point. As such, they also result in too high heat transfer coefficients. Adaptations on the other two models lead a better predictions. None of the models, however, could successfully predict the Reynolds number effects on the flow. The authors concluded that this ought to be due to the two-equation eddy viscosity model that was adopted for all cases to span the near wall sublayer. Numerical simulation of impinging jets using the κ–ε model was also performed by Knowles (1998). The author concluded that the Rodi and Malin corrections could not predict wall jet growth.

Colucci and Viskanta (1996) studied experimentally the effects of nozzle geometry on the local heat transfer coefficients of confined impinging jets. Low nozzle-to-plate gaps were considered in the Reynolds number range of 10,000 to 50,000. The results were compared with similar experiments for unconfined jets. An important conclusion was that the local heat transfer coefficients for confined jets are more sensitive to Reynolds number and nozzle-to-plate gaps than those for unconfined jets.

Dianat et al. (1996) have used the κ–ε model and one modified second-moment closure to make velocity field predictions in the stagnation as well as in the jet region. The second-moment closure was modified to account for the influence of the wall in distorting the fluctuating pressure field away from it. With this modification, the damping of normal velocity fluctuations was well predicted.

The heat transfer in the flow of a cold, two-dimensional, vertical liquid jet against a hot, horizontal, surface was given an approximate solution for the velocity and temperature fields by Shu and Wilkins (1996). The solution is valid for laminar flows and resorts to the hydrodynamic similarity solution of Watson. The results were compared with a numerical realization of the flow.

Meola et al. (1996) investigated the influence of shear layer dynamics on impingement heat transfer. Again, coherent structures and/or recirculating currents were observed to alter the distribution of heat transfer coefficients. Temperatures were measured with an infrared scanning radiometer whereas heat transfer coefficients were evaluated by the heated thin foil technique. The distribution of Nusselt number was discussed and a new explanation given for the local peak in the local Nusselt number.

Liu and Sullivan (1996) investigated the heat transfer and flow structures in an excited circular impinging jet with a small nozzle-to-wall gap. Enhancement or reduction of the local heat transfer coefficient in the wall-jet region was shown to be achieved by exciting the impinging jet near to its natural frequency or subharmonics respectively.

Nishimo et al. (1996) report the turbulence statistics in the stagnation region of an axisymmetric jet impinging vertically on a wall. They used particle-tracking velocimetry do measure the flow near the stagnation point and found that the turbulent normal stress of the axial component gave a substantial contribution to the increase in the static pressure near the wall. Turbulence was studied through an invariant map of the turbulent stress anisotropy. In the stagnation region, turbulence was close to an axisymmetric state.

The standard κ–ε model together with the logarithmic law of the wall was applied by Ashforth-Frost and Jambunathan (1996) to a semi-confined impinging jet; the nozzle-to-wall distance was two nozzle diameters and the Reynolds number 20,000. Laser-Doppler anemometry and liquid crystal thermography were used to determine velocity, turbulence and heat transfer data. In the developing wall jet, authors showed numerical heat transfer results to compare to within 20% of experimental data. However, at the stagnation point, heat transfer is over predicted by about 300%. The authors attributed this discrepancy to failure of the wall function to confirm to the physics of the flow.

San et al. (1997) reported local measurements of Nusselt number for a confined impinging jet. The recirculation and the mixing effect on the heat transfer were investigated by varying the jet diameter, the surface heat flux, the Reynolds number and the surface heating width.

Knowles and Myszko (1998) carried out turbulence measurements in a jet impinging on a flat wall. Different nozzle-to-wall gaps were investigated. Measurements were conducted using hot-wire anemometry. Nozzle height was found to have a large effect on turbulence peak level for distances up to r/d = 4.5; lower nozzle-to-wall ratios caused an increase in peak level measured in all turbulent stresses in the stagnation region.

Kendoush (1998) in a previous work had derived analytical solutions for the convective heat and mass transfer predictions in a laminar jet impinging on a plane wall. However, his solution was shown to have a strong singularity at the stagnation point. The subsequent paper of 1998 had, therefore, the objective of removing such singularity and find workable solutions for the problem. The results were compared with available experimental and numerical data.

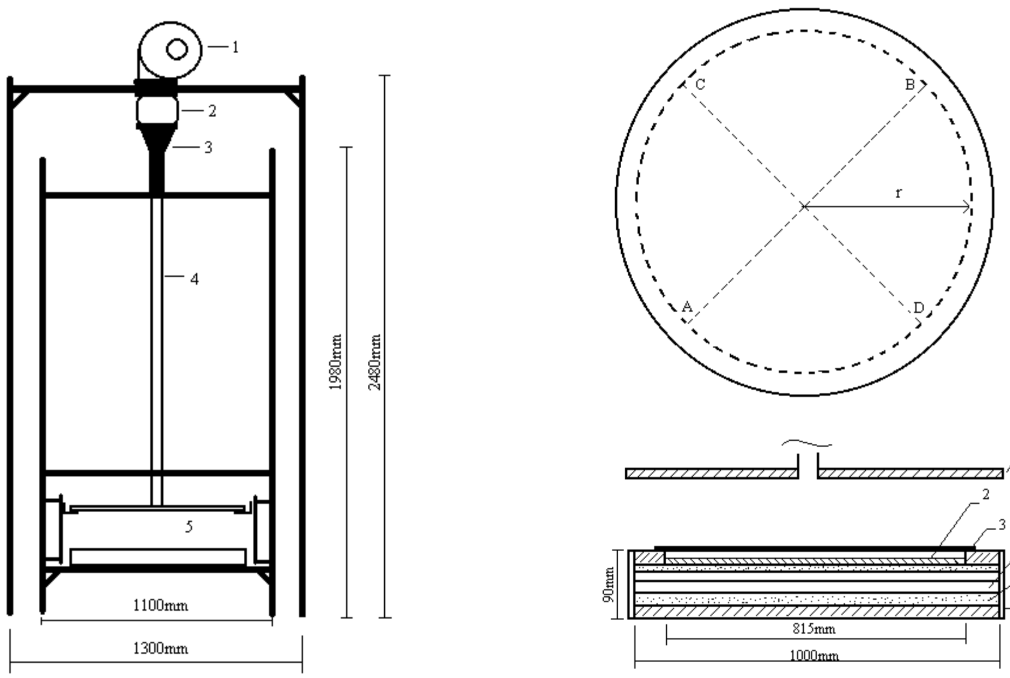
Lee and Lee (1999) experimentally studied the heat transfer behavior of a turbulent jet impinging on a wall with special attention to the stagnation region. For nozzle-to-wall ratios of H/d = 2.0 the local Nusselt number variation with r/d had two peaks and varied according to $Re^{0.5}$. For H/d > 6.0, Nusselt number decreased monotonically with r/d. The Reynolds number dependence was conserved to increase as H/d increases.

Confined impinging jets at low Reynolds number were experimentally studied by Baydar (1999) for a single and a double jet. The author concludes that a subatmospheric region occurs on the impingement wall at nozzle-to-wall gaps up to two and that there is a linkage between the subatmospheric region and the peak in the heat transfer coefficients.

In their next paper, Lee and Lee (2000) studied experimentally the local heat transfer characteristics of an elliptic jet impinging on a heated flat plate for various nozzle aspect ratios. The temperature distributions on the heated plate were measured using a thermochromic liquid crystal thermometry with a digital image processing system. A smoke-wire technique was used to visualize the flow. For small nozzle-to-wall gap, as the aspect ratio of the elliptic jet increases, the heat transfer rate was increased larger than that for the circular jet in the stagnation region. This was attributed to the large entrainment and large scale mixing of the elliptic jet.

3. Experimental methods

A schematic diagram of the experimental apparatus and the test section geometry is illustrated in Fig. (1) below. A centrifugal blower drives air at 22 ^{o}C through a 1350 mm long pipe and 43 mm inner diameter. Inside the pipe, the flow is straighted in a honeycomb made with glued drinking straws; screens were also placed. The flow emerges as a jet from the circular nozzle at bulk velocity of 12 m/s.

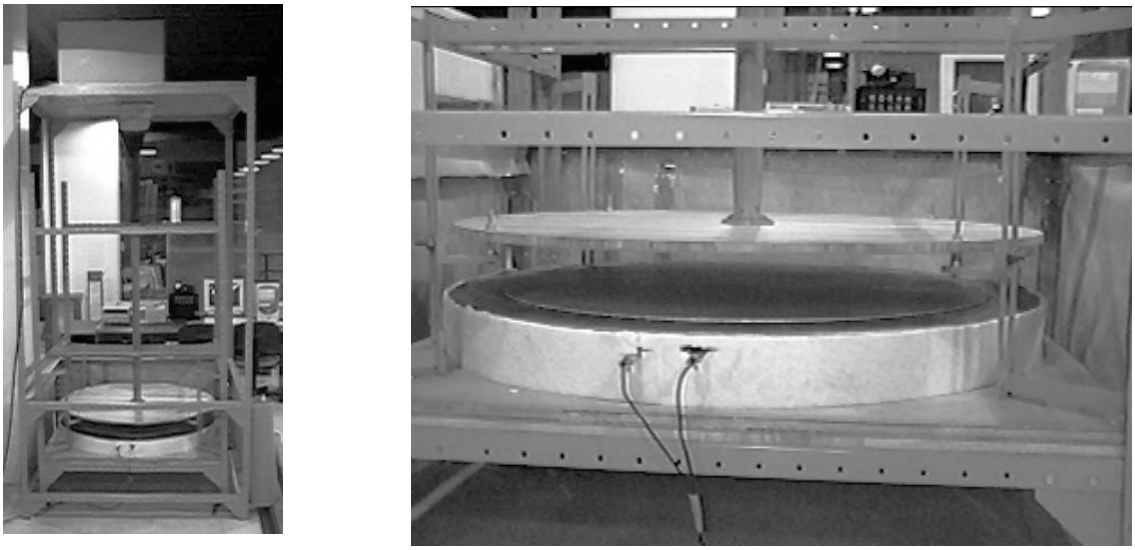


(a) (b)

Figure 1. Schematic diagram of (a) the experimental apparatus: (1) centrifugal blower, (2) flexible section, (3) contraction, (4) pipe, and (5) test section; and (b) test section and its heating system: (1) confinement plate, (2) electrical resistance, (3) impingement plate, (4), (5) and (6) thermal isolation.

The impingement flat plate was made with 3 mm thick aluminum circular sheet. This plate has 840 mm in diameter and was laid over a plenum chamber as shown in Figure 1. The plenum chamber is 20 mm height and 815 mm in diameter. At the bottom of the chamber was installed a 4 kW of electrical resistance with its heat generation controlled by a rheostat. The walls of the plenum were completely insulated from the ambient as shown in Figure 1. Photos of the experimental apparatus are shown in Figure 2.

The controlled parameters in the experiments are the nozzle-to-plate spaces, electric current which controlls the dissipated heat at the electric resistance, the heat flux from the resistance and the stagnation pressure. At each test, the centerline of the jet was lined up with the center of the impingement surface.



(a) (b)

Figure 2. (a) Experimental apparatus and (b) impingement plate.

The temperature of the aluminum sheet was monitored by thermocouples. The readings of the thermocouples were routed to a K6 II 450 MHz personal computer via a Picolog acquisition system model TC-08.

The jet exit velocity was measured using a Pitot tube and an electronic manometer. Temperature measurements were made using thermocouples. Temperature profiles were measured using a chromel-constantan micro-thermocouple that was positioned using a traverse gear system with a sensitivity of 0.02 mm.

The local convective heat transfer coefficient is calculated from

$$h = \frac{q_v}{T_w - T_{aw}}, \tag{1}$$

where T_w and T_{aw} are the wall temperature and the adiabatic wall temperature measured with air flowing, and q_v is obtained by the total imposed heat flux input through the electrical resistance, that is given by

$$q_v = \frac{4\, I\, V}{\pi\, D^2}. \tag{2}$$

The conduction heat flux (q_c) was considered negligible and the radiation heat flux (q_r) was calculated and achieved 2% of the total imposed heat flux. The local surface temperature was converted to the local Nusselt number defined as

$$Nu = \frac{hD}{k}, \tag{3}$$

where D is the jet exit diameter and k is the thermal conductivity of air.

To perform the experiments, an elaborate procedure was devised. First, the flat plate was fitted with 27 pressure taps arranged in a cross shape. The readings of the pressure at these points were subsequently used to find the geometrical center of the jet; only when the pressure distribution was found to be completely symmetric the jet centerline was considered determined.

To find the adiabatic temperature, the resistance was turned off and the temperature recorded. Next, the heater was turned on and the behaviour of the wall temperature was recorded. Normally, 6 references points were used at this stage. Only when the plate was observed to reach a steady state the fan was turned on. In the steady state condition, the wall temperature variation was within 1 ^{o}C. Approximately 2.5 h was required to reach steady state conditions for each test run.

4. Results

The work will present results for three different geometries defined by *H/D* = 1.0, 1.5 and 2.0.

The measured pressure at the geometrical center of the jet is shown in Fig. 3. The radial pressure distributions on the impingement surface were measured for three nozzle-to-plate spaces of *H/D* = 1.0, 1.5 and 2.0, these are shown in Fig. 4. The plot reveals that the pressure coefficient depends on the nozzle-to-plate spacing having a maximum at the stagnation point. The regions of subatmospheric pressure corresponds to regions of recirculating flow. The pressure measurements are nondimensionalized with dynamic pressure $rU^2/2$, where r is the density of air and U is the jet exit velocity. Other authors (see, e.g., Colucci and Viskanta(1996)) have stated that, as nozzle-to-plate spacing decreases, the distance for which the pressure remains subatmospheric and the strength and size of the recirculation zone increase. This fact is clearly observed in Fig. 4.

Heat transfer data are reduced in dimensionless form in terms of Nusselt number, Nu, and the distributions for different nozzle-to-plate spaces are shown in Fig. 5. Indeed, we know that for large *H/D*, nozzle-to-plate spaces, the Nusselt number presents a maximum at the stagnation point decreasing as increase *r/D*. To small values of *H/D*, the Nusselt number is no-longer maximum at stagnation, being rather a local minimum. Two local maxima are then observed. Here, these trends have been confirmed. The local heat transfer begins to increase from the stagnation point toward the local first peak position near $r/D \cong 0.5$ then it decreases toward $r/D \cong 2.0$ for *H/D* = 2.0, and $r/D \cong 1.5$ for *H/D* = 1.0 and 1.5. After this point a second peak appears and then it decreases.

The temperature profiles were taken at six different positions for r = 0, 70, 120, 170, 220 and 270 mm. Dimensional graphs of these quantities are shown in Figure 6. The origin of the coordinate axes is the point of stagnation flow. The temperature profiles show a weak peak away from the wall. This has also been observed by other authors and is related to the temperature stratification.

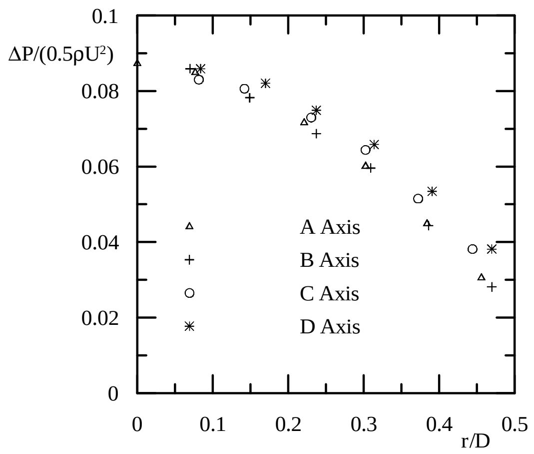


Figure 3. Pressure distribution at the geometrical center of the jet.

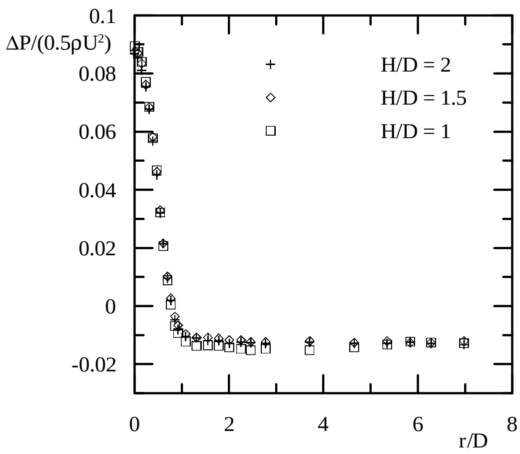


Figure 4. Radial pressure distributions

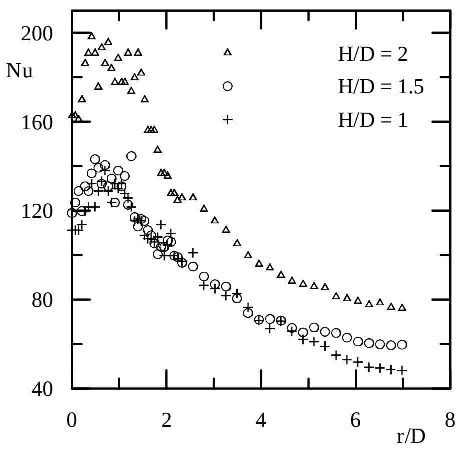


Figure 5. Nusselt number distributions.

Before we consider the behaviour of the temperature let us look at some classical concepts for the velocity profile.The inner equilibrium layer is renowned for its inherent difficulty in description. The law of the wall defines an equilibrium velocity profile

$$\frac{u}{u_t} = \frac{1}{k} ln\left(\frac{yu_t}{n}\right) + A \tag{4}$$

where u_τ is the friction velocity and k is the von Karman constant.

Özdemir and Whitelaw (1992) remarked that, for the impinging jet, the inner layer appears to constitute a considerable part of the inner boundary layer. They also state that if the outer edge of the equilibrium layer is attached to the point of radial velocity maximum then this maximum should be the velocity scale to make all velocity profiles coincide. The conclusion, therefore, was that parameter A was not invariant anymore and that in fact it followed a single relation of the form

$$A = 1.292\ (u_M - u_t) - 6.2 \tag{5}$$

where u_M denotes the point of radial velocity maximum.

The decay of the radial velocity along *"r"* direction is given

$$u_M = \frac{C}{r^n} \tag{6}$$

where C and n where found to be 0.095 and the exponent n to be 1.459. Özdemir and Whitelaw (1992) reported that other authors had found the values of n about 1.1.

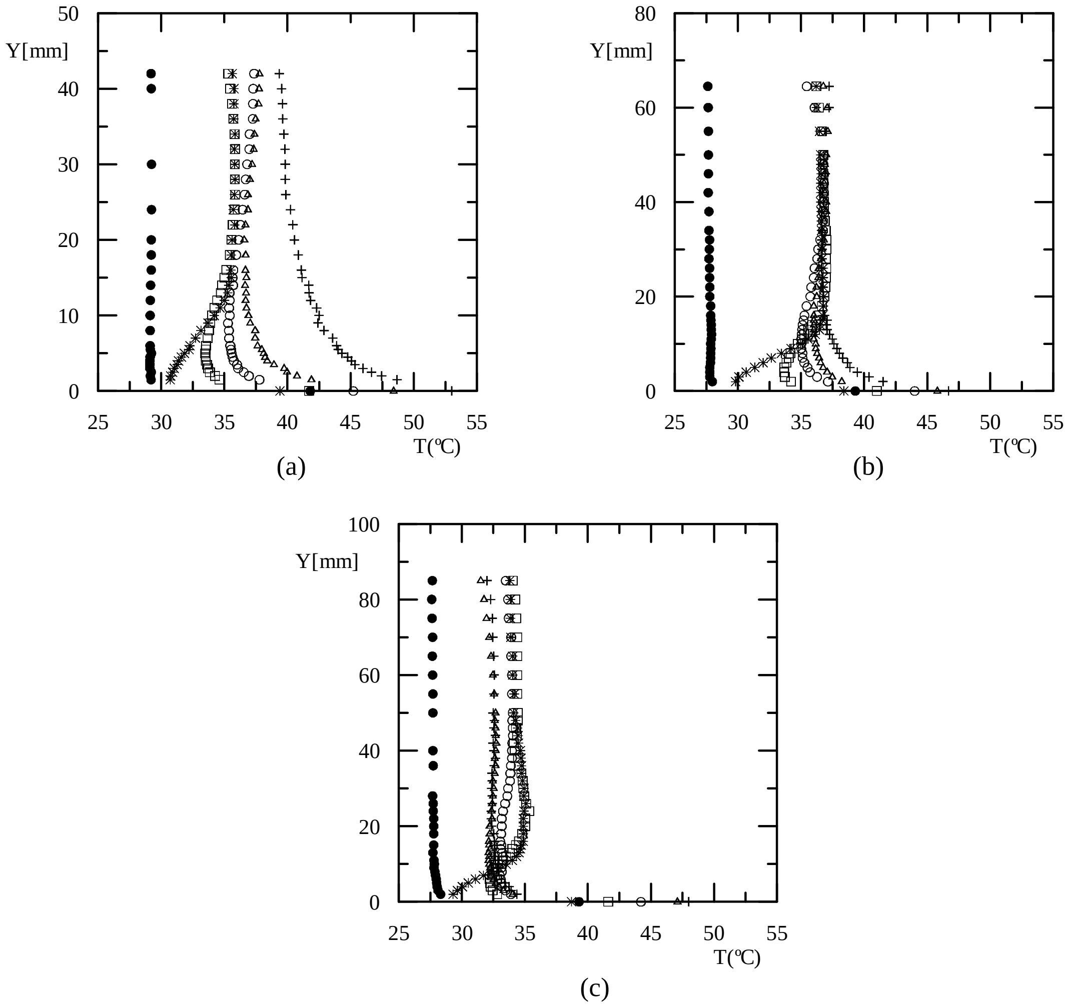


Figure 6. Mean temperature profiles in physical coordinates: (a) H/D = 1.0, (b) H/D = 1.5 and (c) H/D = 2.0. Legend: •, r = 0 mm; ∗, r = 70 mm; , r = 120 mm; °, r = 170 mm; Δ, r = 220 mm; +, r = 270 mm.

The growth of the wall jet evaluated as a function of its half-velocity thickness can be written as $y_{0.5} = K\, r^m$, where $K = 0.242$ and $m = 1.156$.

The existence of a temperature equilibrium layer was also investigated here. The law of the wall for the temperature profile can be written as

$$\frac{T_w - T}{t_t} = \frac{1}{k_t} ln\, y + \frac{1}{k_t} ln\left(\frac{P_r u_t}{n}\right) + B \tag{7}$$

where t_t is the friction temperature and k_t is the von Karman constant for the temperature field.

Equation (7) can be rewritten in the form

$$\frac{T_w - T}{t_t} = \frac{1}{k_t} ln\, y + B' \tag{8}$$

where

$$B' = \frac{1}{k_t} ln\left(\frac{P_r u_t}{n}\right) + B.$$

The important aspect to notice here is that since P_r and n are properties of the fluid and since u_t is supposed to vary slowly outside the stagnation point, the log term is almost constant. Therefore the behaviour of *B'* will be dictated by the behaviour of *B* and vice-versa. Thus, if the trends observed by Özdemir and Whitelaw (1992), and depicted by Eq. 5, are to be met for the temperature field, this should be captured by any of the parameters *B'* or *B*.

The temperature profiles in inner variables are shown in Figure 7.

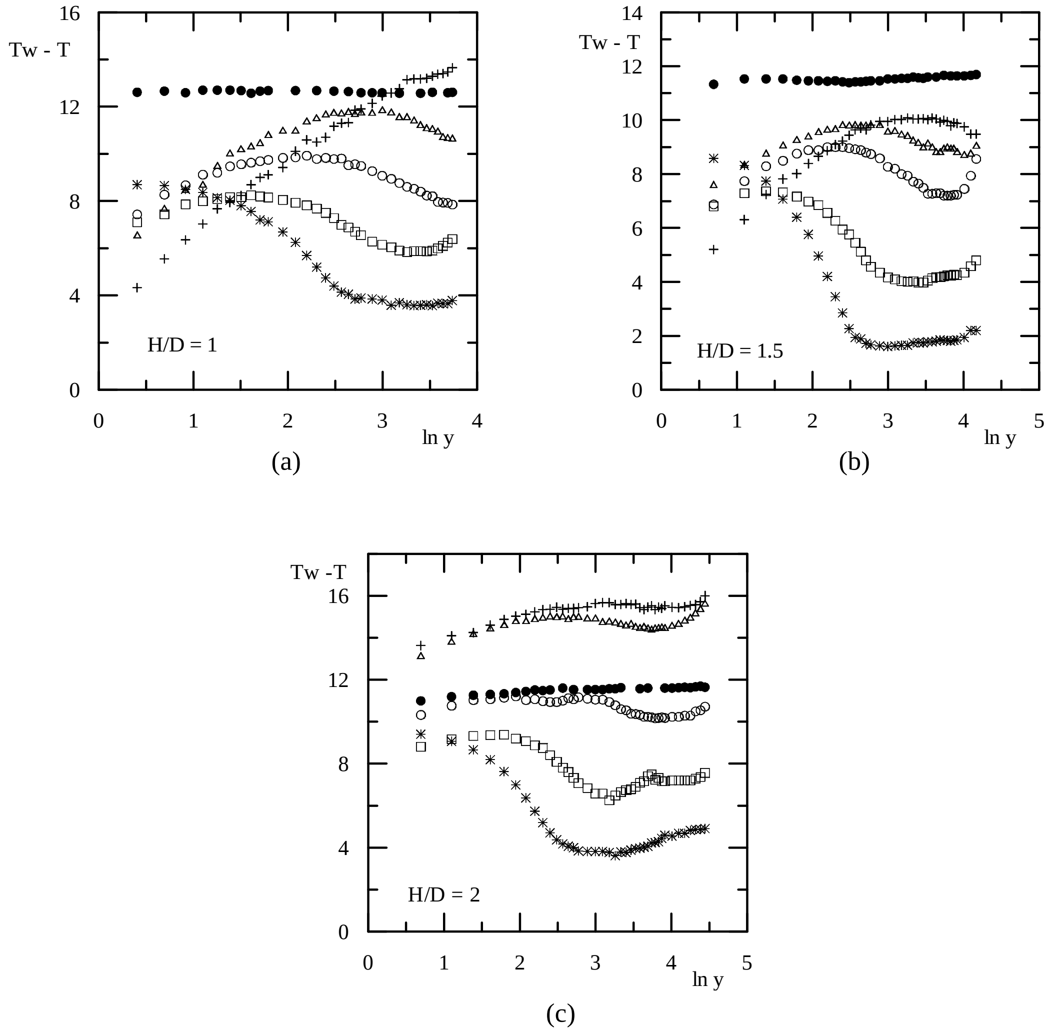


Figure 7. Temperature profiles in inner variables: (a) *H/D* = 1.0, (b) *H/D* = 1.5 and (c) *H/D* = 2.0. Legend: •, *r* = 0 mm; ∗, r = 70 mm;   , *r* = 120 mm; °, *r* = 170 mm; Δ, *r* = 220 mm; +, *r* = 270 mm.

The existence of a conductive layer can be observed. The equilibrium layer is represented by the logarithmic curve.

Thus, and in accordance with the previous remarks related to the temperature profile, for *H/D* ratios of 1 and 1.5, full logarithmic profiles can be observed for distances over r = 220 mm. For *H/D* = 2, the boundary layer thickness is very small but, for distances over 170 mm, log profiles can already be observed.

Table 1 below shows the typical values founded for the flow parameters.

Table 1. Flow parameters.

H/D	Station (mm)	t_t (°C)	*B'*
1	270	1.51	2.70
1.5	270	1.13	4.16
2	220	0.576	21.31
2	270	0.458	28.20

Despite the small number of temperature profiles that were analyzed here, just four, the following findings are remarkable: 1) the variation of *B'* for all three flow configurations is marked, 2) as recognized by Eqs. 5 and 6 the level in the logarithmic expression for the law of the wall seems to diminish with increasing r; this has also been observed here for the temperature profiles.

Thus it appears that the trend observed by Özdemir and Whitelaw (1992) for the behaviour of the velocity law of the wall is also followed by the temperature law of the wall.

5. Conclusion

The present work has described the behavior of a semi-confined impinging jet over a heated flat plate. Experimental data for the pressure distribution and temperature fields were obtained. The heat transfer data confirmed the existence of a peak in temperature profile away from the wall. The existence of a velocity and a temperature equilibrium layer was also investigated. The results found at this preliminary investigation indicate that the level of the logarithmic portion of the temperature law of the wall decreases with increasing radial distance. This fact has been observed in literature here for the first time

Acknowledgements. DRSG is grateful to CAPES (Ministry of Education) for the award of a D.Sc. scholarship in the course of the research. APSF is grateful to the Brazilian National Research Council (CNPq) for the award of a research fellowship. The work was financially supported by CNPq and by FAPERJ.

6. References

Ashforth-Frost, S. and Jambunathan, K., 1996, "Numerical prediction of semi-confined jet impingement and comparison with experimental data", Int. J. Numer. Methods Fluids, Vol. 23, pp. 295-306.

Baydar, E., 1999, "Confined impinging air jet at low Reynolds numbers", Exp. Therm. Fluid Sci., Vol. 19, pp. 27-33.

Behnia, M., Parneix, S. and Durbin, P. A., 1998, "Prediction of heat transfer in an axisymmetric turbulent jet impinging on a flat plate", Int. J. Heat and Mass Transfer", Vol. 41, No. 12, pp. 1845-1855.

Behnia, M., Parneix, S., Shabany, Y. and Durbin, P. A., 1999, "Numerical study of turbulent heat transfer in confined and unconfined impinging jets", Int. J. Heat and Fluid Flow, Vol. 20, pp. 1-9.

Colucci, D. W. and Viskanta, R., 1996, "Effect of nozzle geometry on local convective heat transfer to a confined impinging air jet", Exp. Therm. Fluid Sci, Vol. 13, pp. 71-80.

Cooper, D., Jackson, D. C., Launder, B. E. and Liao, G. X., 1993, "Impinging jet studies for turbulence model assessment-I. Flow-field experiments", Int. J. Heat Mass Transfer, Vol. 36, No. 10, pp. 2675-2684.

Craft, T. J., Graham, L. J. W. and Launder, B. E., 1993, "Impinging jet studies for turbulence model assessment-II. An examination of the performance of four turbulence models", Int. J. Heat Mass Transfer, Vol. 36, No. 10, pp. 2685-2697.

Cruz D. O. A. and Silva Freire A. P., 1998, "On single limits and the asymptotic behavior of separating turbulent boundary layers", Inter. J. Heat and Mass Transfer, Vol. 41, pp. 2097-2111.

Dianat, M., Fairweather, M. and Jones, W. P., 1996, "Predictions of axisymmetric and two-dimensional impinging turbulent jets", Int. J. Heat and Fluid Flow, Vol. 17, pp. 530-538.

Fox, M. D., Kurosaka, M., Hedges, L. and Hirano, K., 1993, "The influence of vortical structures on the thermal fields of jets", J. Fluid Mech., Vol. 255, pp. 447-472.

Gibson, M. M. and Harper, R. D., 1997, "Calculation of impinging-jet heat transfer with the low-Reynolds-number q-z turbulence model", Int. J. Heat and Fluid Flow, Vol. 18, pp. 80-87.

Kendoush, A. A., 1998, "Theory of stagnation region heat and mass transfer to fluid jets impinging normally on solid surfaces", Chem. Eng. Processing, Vol. 37, pp. 223-228.

Kline, S. J.; 1985, The Purpose of Uncertainty Analysis, J. Fluids Engineering, 107, 153-160.

Knowles, K., 1996, "Computational studies of impinging jets using k-e turbulence models", Int. J. Numer. Methods Fluids, Vol. 22, pp. 799-810.

Knowles, K. and Myszko, M., 1998, “Turbulence measurements in radial wall-jets”, Exp. Therm. Fluid Sci, Vol. 77, pp. 71-78.
Lee, J. and Lee, S. -J., 1999, “Stagnation region heat transfer of a turbulent axisymmetric jet impingement”, Exp. Heat Transfer, Vol. 12, pp. 137-156.
Lee, J. and Lee, S. -J., 2000, “The effect of nozzle aspect ratio on stagnation region heat transfer characteristics of elliptic impinging jet”, Int. J. Heat Mass Transfer, Vol. 43, pp. 555-575.
Liu, T. and Sullivan, J. P., 1996, “Heat transfer and flow structures in an excited circular impinging jet”, Int. J. Heat Mass Transfer, Vol. 39, No. 17, pp. 3695-3706.
Meola, C. and Carlomagno, G. M., 1996, “Influence of shear layer dynamics on impingement heat transfer”, Exp. Therm. Fluid Sci., Vol. 13, pp. 29-3.
Nishino, K., Samada, M., Kasuya, K. and Torii, K., 1996, “Turbulence statistics in the stagnation region of an axisymmetric impinging jet flow”, Int. J. Heat and Fluid Flow, Vol. 17, 193-201.
Ozdemir, I. B. and Whitelaw, J. H., 1992, “Impingement of an axisymmetric jet on unheated and heated flat plates”, J. Fluid Mech., Vol. 240, pp. 503-532.
San, J.-Y., Huang, C.-H. and Shu, M.-H, 1997, “Impingement cooling of a confined circular air jet”, Int. J. Heat Mass Transfer, Vol. 40, No. 6, pp. 1355-1364.
Shu, J.-J. and Wilks, G., 1996, “Heat transfer in the flow of a cold, two-dimensional vertical liquid jet against a hot, horizontal plate”, Int. J. Heat Mass Transfer, Vol. 37, No. 16, pp. 3367-3379.