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The near wall behaviour of an ortogonally impinging jet

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Abstract. *The present work investigates the applicability of scaling log-laws to the turbulent impinging jet. Both, the velocity and the temperature fields are studied under this assumption. To validate the proposed expressions, a thorough experimental program was carried out based on thermal anemometry. Measurements of local velocity and temperature distributions are presented as well as longitudinal turbulence profiles. The experiments were conducted for three nozzle-to-plate spacing and Reynolds number of 35,000. A constant wall heat flux condition was achieved by conducting electricity through thin resistors that were placed beneath an aluminum disk. Mean temperature was measured through thermocouples.*

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

1. Introduction

The problem of a turbulent jet impinging orthogonally onto a surface has been dealt with by the present authors in two recent publications (Guerra e Silva Freire (2003, 2004)). Those publications analyzed both the velocity and the temperature fields with an emphasis on a description of the inner layers of the flow. At the time, both works specifically analyzed the existence of the so called universal law of the wall, in view of its relevance for the calculation of the wall shear stress and of the wall heat transfer. For wall jets, we cannot positively say that the log-law is a well established concept. In fact, several authors (Patel (1962), Tailland and Mathieu (1967), Ozarapoglu (1973), Irwin (1973)) have reported a large range for values for the log-law constants. This certainly raises some important questions as to the validity on the use of the log-law for the estimation of surface friction and the heat transfer coefficient.

Almost at the same time however, other researchers (Özdemir and Whitelaw (1992)) showed that for an oblique impinging jet the law of the wall could be observed for both the velocity and the temperature fields. More than that, these authors proposed a functional behavior for the log-law parameters that resorted to a scaling procedure based on the stream-wise evolution of the flow by the maximum jet velocity. In fact, Narasimha et al. (1973) were the first to acknowledge that the traditional use of the nozzle diameter as the reference scaling for wall jet flows was not appropriate. They proposed a scaling length that would take into consideration the flow evolution.

The purpose of the present work is to carry out further investigations on scaling laws governing the motion of a orthogonal jet impinging onto a surface. Here, for the first time, we will present data for the longitudinal turbulent intensities. The law of the wall, for both the velocity and the temperature fields, will also be investigated under the light of some new data.

Thus, at this point, it important to make it clear to the reader that other authors have specifically studied the role of the scaling laws in wall jet flows. That is the case of the work of Wygnanski et al. (1992) where the relevance of the wall to the evolution of the large coherent structures in the flow was studied. Here, in the impinging jet, the problem is further complicated by a deflection of the streamlines and by the presence of a stagnation point.

A turbulent jet impinging orthogonally onto a surface is a geometrical arrangement commonly used in industry to promote high rates of heat exchange. The studies have concentrated on the investigation of different features of the phenomenon because of the several important aspects associated to the problem. These studies normally solve for the velocity and the temperature fields in regions around 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. Cases where the Reynolds number is low enough so that the flow can be rendered laminar, asymptotic methods can be used to find analytical solutions in all flow regions except near the stagnation point, which presents a strong singularity. Consequently, even for this simple flow condition, calculation of the heat transfer coefficient at the stagnation point is very difficult. The result is that a severe lack of information on the flow behaviour in the stagnation region exists. The reason for this is clear, due to the small scales that define this region, the placement of dedicated instrumentation is always very difficult.

The flow structure of an impinging jet produced by a nozzle can be highly complex due to the ambient fluid entrainment, flow separation, interaction of the flow with the impingement or confining walls, and generation of vortices. In this work, we will provide experimental data on turbulent semi-confined and unconfined impinging jets.

Once understood the flow structure, one can foresee how this dynamics affects the heat transfer process. Results are presented for the turbulent characteristics of a round jet. The work includes measurements for the radial mean and instantaneous velocity profiles and pressure distributions. The velocity field was measured through a hot-wire system.

In regard to 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 but near 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 much 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.

2. Short literature review

Because this paper is concerned with experiments on the impingement of a jet onto a heated plate, we will make a short literature review on the subject.

For the wall jet, the first studies were severally limited by the lack of any sophisticated instrumentation. As a result, the first experiments were limited to measurements of mean velocities in the vicinity of the nozzle, see e.g. Forthman (1934). Sigalla (1958) was the first to try to evaluate the skin-friction and also the first to measure the mean velocity at large distances from the nozzle. The development of the hot-wire anemometer in the sixties made it possible the development of much more detailed investigations.

Tailland and Mathieu (1967) noticed that the rate of spread of a wall jet and the decay of its maximum velocity were dependent on the Reynolds number, a feature that is not observed in a free jet. That raised questions on the reason for such difference.

The scaling laws for wall jets were particularly studied by Patel (1962), Ozarapoglu (1973) and Irwin (1973).

In relation to the impinging jet, the following are the notable references.

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 an 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 too much 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.

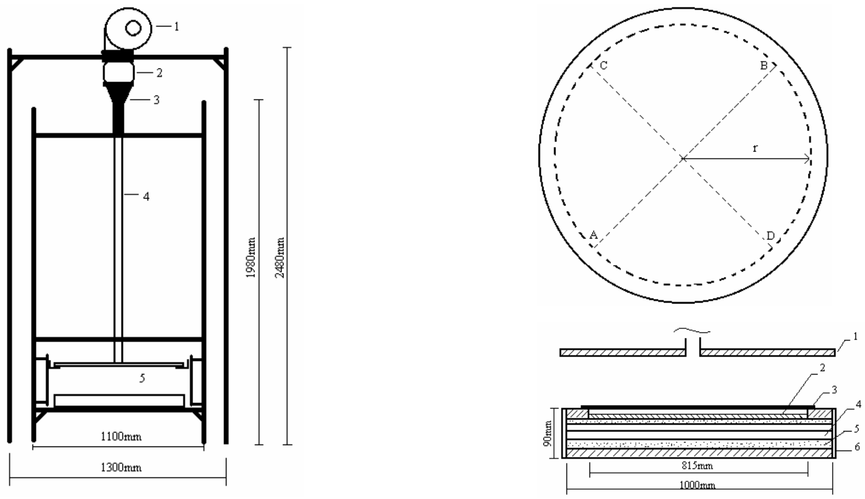
Lee and Lee (1999) experimentally studied the heat transfer behavior of a turbulent jet impinging on a wall with special attention to the stagnation region. For nozzle-to-wall ratios of $H/d = 2.0$ the local Nusselt number variation with r/d had two peaks and varied according to $Re^{0.5}$. For $H/d > 6.0$, 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 sub-atmospheric region occurs on the impingement wall at nozzle-to-wall gaps up to two and that there is a linkage between the sub-atmospheric 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 problem geometry is illustrated in Fig. (1) below. Air at 22 ^{o}C was pumped through a centrifugal blower and passed through a 1350 mm long pipe with 43.5 mm internal diameter. Inside the pipe, a flow-straightener honeycomb was fitted constructed from drinking straws glued together; screens were also set in place. The jet was set to emerge from the circular nozzle with a bulk velocity of 12 m/s.

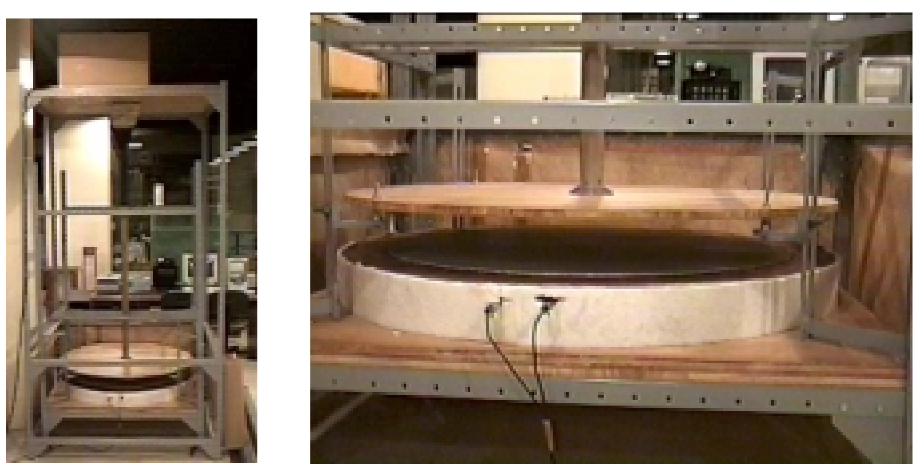


(a) (b)

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 of a 3 mm thick aluminum circular sheet. This sheet had 840 mm in diameter and was laid over a plenum chamber as shown in Figure 1. The plenum chamber was 20 mm height and 815 mm in diameter. At the bottom of the chamber a series of electrical resistances were placed that debited a maximum of 4.000 W. 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 were the nozzle-to-plate spacing, the resistance heat flux 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 through thermocouples. The readings of the thermocouples were routed to a AMD Athlon +2000 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 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 was calculated from

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

where T_w and T_{aw} are wall temperature and adiabatic wall temperature of the stream. q_v is the total heat flux imposed through the electrical resistance and 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 was 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 at a cross formation. 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 electric current was turned off and the temperature recorded. Next, the heater was turned on and the behaviour of the wall temperature recorded. Normally, 6 control 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 are required to reach steady state conditions for each test run.

4. Results

The work will present complete results for three different geometries defined by the aspect ratio H/D = 1.0, 1.5 and 2.0. In previous publications by the present authors, just a fraction of these results had been presented.

The radial pressure distributions on the impingement surface were measured for the three nozzle-to-plate spaces of H/D = 1.0, 1.5 and 2.0, these are shown in Fig. 3. The plot shows that the pressure coefficient depends on the nozzle-to-plate spacing. The pressure measurements were non-dimensionalized with the dynamic pressure, $\rho U^2/2$, where ρ is the density of air and U is the jet exit velocity.

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 Figure 4.

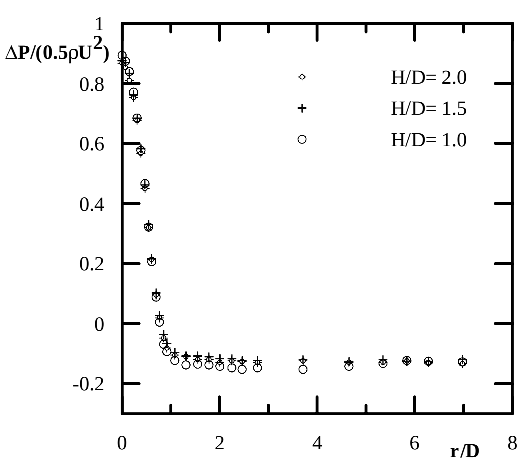


Figure 3. Radial pressure distributions of the jet.

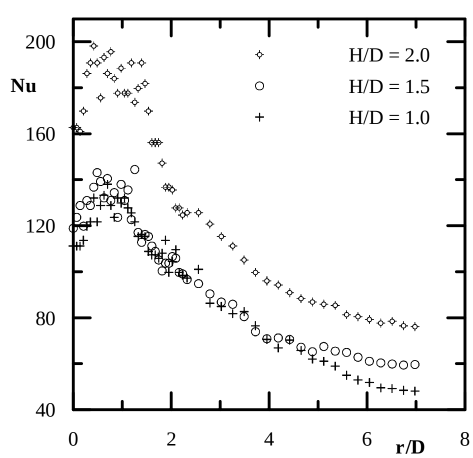


Figure 4. Nusselt number distributions.

Indeed, we know that for large H/D, nozzle-to-plate spaces, the Nusselt number presents a maximum at the stagnation point decreasing with increasing r/D. To small values of H/D, the Nusselt number is no-longer maximum at stagnation, being rather a local minimum. One or two local maxima may then be observed. Here, these trends have been confirmed. The local heat transfer begins to increase from the stagnation point towards a local first peak position near $r/D \cong 0.5$ then, it starts to decrease reaching a weak local maximum towards $r/D \cong 2.0$ for H/D = 2.0, and $r/D \cong 1.5$ for H/D = 1.0 and 1.5. After this point a second weak peak appears and then decreases.

To characterize the jet wall spreading, the radial component of the mean velocity was examined at various radial positions.

The velocity and turbulent intensity profiles are shown in Figures 5 and 6.

The study of Özdemir and Whitelaw (1992) has shown that a Weibull distribution represents well some of the global features of the profile, such as the position of the maximum and outer inflection points, but is not an adequate approximation for the near wall region. For this region, they showed that semi-log relation can be used to model the inner equilibrium layer. Thus, it follows that

$$\frac{u}{u_\tau} = \frac{1}{\kappa}\ln\left(\frac{y u_\tau}{\nu}\right) + A \tag{4}$$

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

The main contribution of Özdemir and Whitelaw (1992) was to show that, for the impinging jet, the inner layer appears to constitute a considerable part of the inner boundary layer and that if the outer edge of the equilibrium layer is attached to the point of maximum radial velocity, which is very close to the wall then, this maximum, u_M, should be the appropriate velocity scale. The conclusion, therefore, was that parameter A is not invariant but changes with a deviation function. To describe A, these authors proposed a simple relation of the form $A = 1.292\ (u_M - u_\tau) - 6.2$, where u_M denotes the point of maximum radial velocity.

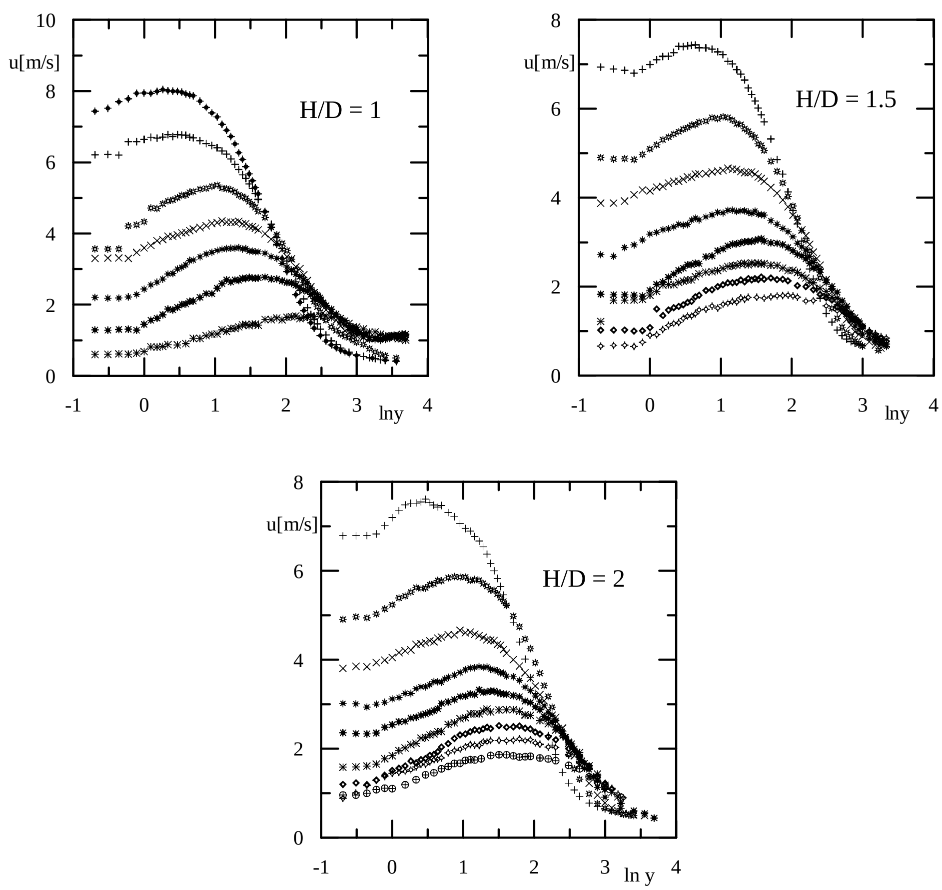


Figure 5. Mean velocity profiles in dimensional variables. Y is given in millimeters.

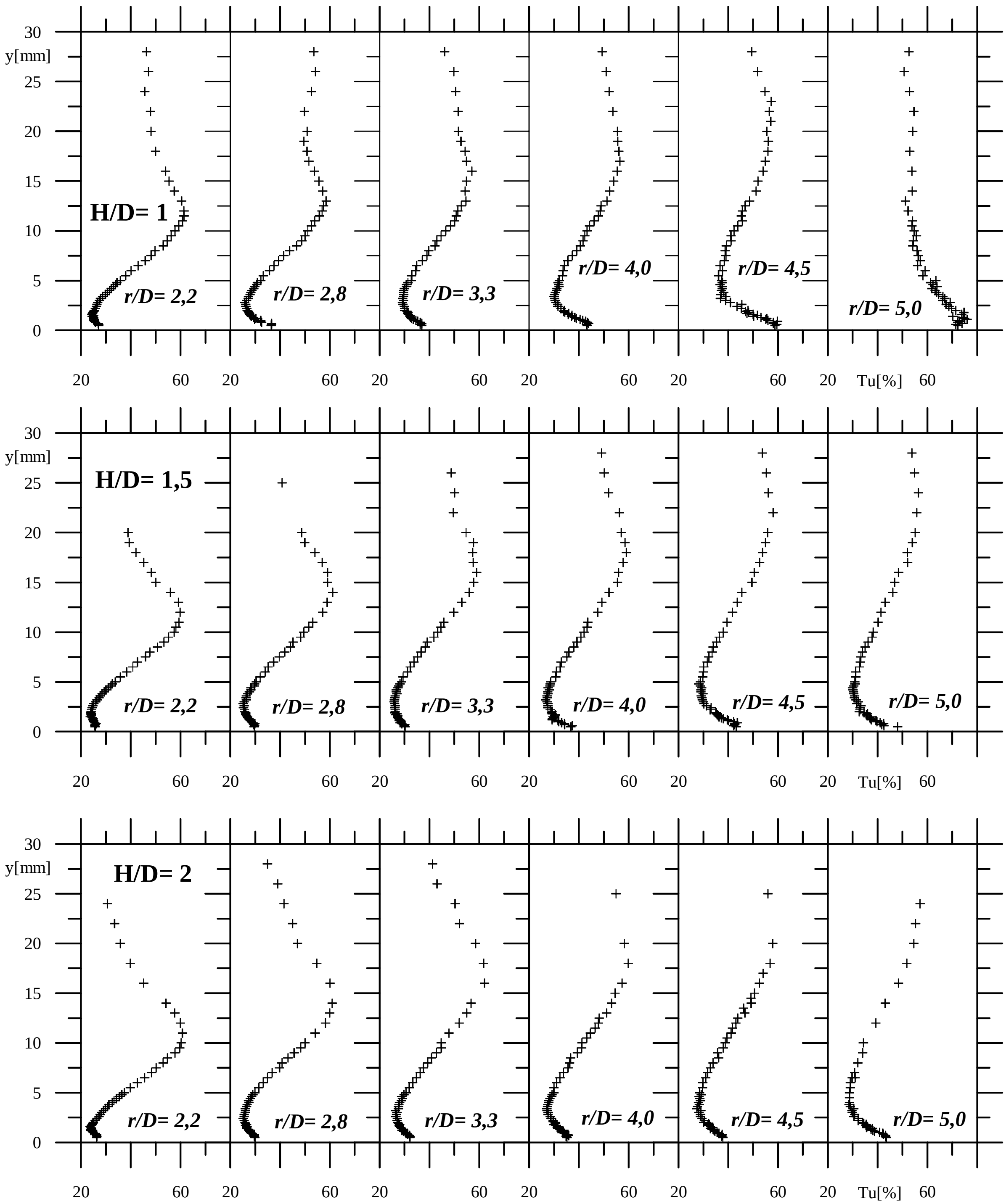


Figure 6. Longitudinal turbulent intensity profiles.

The establishment of these concepts for the velocity field clearly raises the same questions for the temperature field. This surely must lead to an investigation on the existence of a temperature equilibrium layer

The law of the wall for the temperature profile can be written as

$$\frac{T_w - T}{t_\tau} = \frac{1}{k_t} \ln\left(P_r \frac{y u_\tau}{\nu} \right) + B \tag{5}$$

where t_τ is the friction temperature and κ_t is the von Karman constant for the temperature field.

To assess the behaviour of *B*, Figure 7 was analyzed.

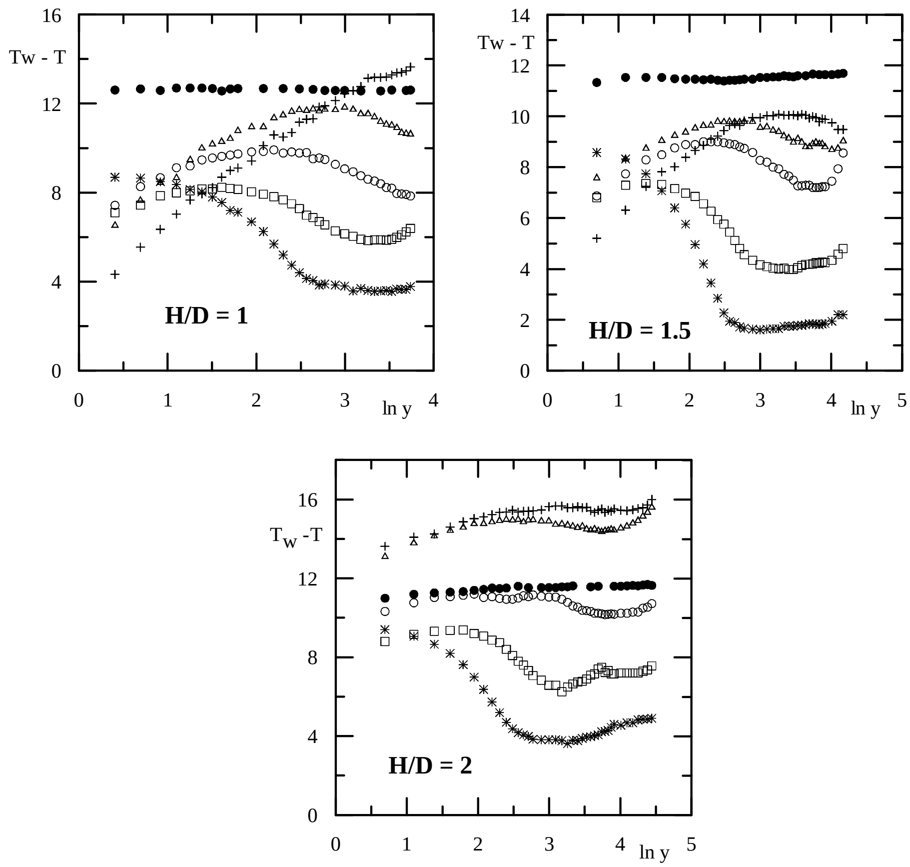


Figure 7. Mean temperature profiles.

To find the values of *A* and of *B*, the graphical method of Coles (1956) was used. Here, we must point out that the thickness of the inner turbulent region for an impinging jet is very thin, what makes the fitting of a straight line a difficult affair. The analysis of Wygnanski et al. (1992), however, implies that von Karman's parameter can be considered constant and that *A* varies from 5.5 to 9.5.

A data reduction from Figures 5 and 7 furnishes Figure 8.

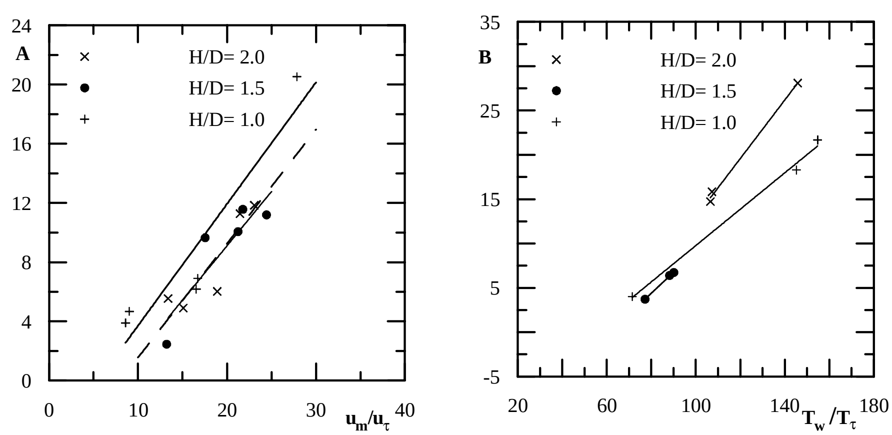


Figure 8. Deviation function for the velocity and the temperature profiles.

This figure seems to indicate that both *A* and *B* increase as the maximum jet velocity and the wall temperature increase respectively. Despite the scatter in the data these trends seem to be well defined.

Thus, the trends observed by Özdemir and Whitelaw (1992) are repeated here. Furthermore, the analysis, carried out separately for each condition of nozzle-to-plate space, gave us a strong hint that a possible linear behaviour of A and of B as a function of the maximum jet velocity and of the wall temperature would be in order. However, at this stage of the research, the authors do not feel adequate to commit themselves with a definite relation to describe this behaviour.

Despite our brief account of the problem of an orthogonal jet impinging on a wall, the following findings are remarkable: 1) the variations both A and B for all three flow configurations is marked, 2) the level in the logarithmic expressions for the laws of the wall have a weak tendency to increased with increasing maximum jet velocity and wall temperature.

Thus, it appears that the trends 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 velocity and the temperature laws of the wall increases with increasing maximum jet velocity and wall temperature. This fact has been observed for the first time in the course of the present research.

The present research is particularly relevant because, in their research, Wygnanski et al. (1992) have decided that the most reliable method for measuring the wall stress is to use the slope of the mean velocity profile near the surface. This method was tested against floating drag balances, Preston tubes and the momentum integral equation.

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