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Effect of internal flux on the heat transfer coefficient in circular cylinders in cross flow

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Abstract

The present paper shows results of laboratory measurements carried out to study effect of perforations on the cooling of cylinders under forced convection. Measurements were carried out for Reynold's numbers between 2000 and 15,000. Results indicate that the presence of holes and their orientation relative to air flux have a significant effect on the cooling process. Nusselt numbers have been found to increase as much as 20% as compared to a corresponding smooth cylinder.

Keywords: Ventilation coefficient; Nusselt number; Internal flux; Circular cylinder

1. Introduction

The fluid flow and heat transfer from a cylinder have been studied analytically, experimentally and numerically by many researchers including Zukauskas [1,2], Roshko [3], Achenbach [4], Schlichting [5], Churchill and Bernstein [6], Eckert and Soehngen [7], Morgan [8], Hilpert [9], Perkins and Leppert [10], Asanitjai and Goldstein [11], Park and Lee [12], Nakamura and Igarashi [13].

The interest in this issue lies in the many and varied applications in which it is necessary to know the parameters describing the heat exchange (as well as other dynamic interactions, such as the drag coefficient) between a body and the fluid around it.

In general, the use of simple geometries makes experiments more feasible and results easier to analyze. Nevertheless, practical applications do not always make use of objects of simple geometry. Yajima and Sano [14] reported measurements of the drag coefficient in perforated circular cylinders. These authors found the drag coefficient to be as much as 40% less in these than in smooth cylinders, for a wide range of attack angles (from 20° to 60°) and for Reynolds numbers (Re) between 0 and 9000. These authors interpret the decrease in the drag coefficient in terms of the behaviour of the fluid surrounding the body observed for different attack angles.

The current work presents an experimental study of the cooling of perforated cylinders by the forced convection mechanism. The observations exhibit a good correlation with those reported by Yajima and Sano regarding the behaviour of fluid bathing the cylinders.

2. Experimental technique

When a body at an initial temperature T_i, is bathed by an air flux of temperature T_a and speed V, it is cooled by means of a process of conduction and forced convection (assuming $T_a < T_i$). In general, the heat lost by unit time and through a unit area can be expressed in the following way:

$$Q = h_m \cdot (T_s - T_a) \tag{1}$$

where h_m is the convection heat transfer coefficient, T_s the body surface temperature in contact with the fluid and T_a is the fluid temperature at an infinite distance from the body.

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The value of h_m represents the effectiveness of the fluid in removing heat from the body; it depends on the Reynolds number associated to the body and on other geometric factors, such as surface roughness.

In the case of material objects with high thermal conductivity, body temperature in Eq. (1) can be considered uniform, that is: $T_s = T$. For these cases, the lumped capacitance method may be used and the expression for body temperature (T) as a function of time has the form [15,16]

$$T(t) - T_a = (T_i - T_a) \cdot e^{-t/\tau} \tag{2}$$

where

$$\tau = m \cdot C_c / h_m \cdot S \tag{3}$$

S and m being respectively the area through which heat transfer takes place with the medium and the body mass, and C_c is the body's specific heat.

It is worth recalling that h_m and the Nusselt number (Nu) are related through [15,16]

$$h_m = Nuk_f / D \tag{4}$$

and

$$Nu = CPr^{1/3}Re^n \tag{5}$$

where k_f is the fluid's thermal conductivity coefficient, D is a length characteristic of the body, Pr is the Prandtl number for the fluid (when the fluid is air under atmospheric conditions $Pr \sim 0.71$), C and n are constants that in general do not depend on the fluid's characteristics, but do depend on body geometry, the kind of flux occurring around the body and on Re range.

The experiments were performed with the purpose of determining the Nusselt number of cylindrical aluminium bodies immersed in an air flux by measuring the time-variation in the body temperature. The Biot number (Bi) provides a measure of the temperature variation in the solid relative to the temperature difference between the surface and fluid. The Biot number estimated in the current experiments is $Bi < 0.01$, then it is reasonable to assume a uniform temperature distribution within the solid which ensures that the lumped capacitance method is adequate to be applied in this work. A scheme of the experimental device is presented in Fig. 1.

The characteristic longitudinal and cross sections of the cylindrical bodies employed are schematically shown in Fig. 2. C_0 is a smooth cylinder obtained from a solid cylindrical rod with a central hole along its axis (annulus). Its dimensions are: 11 mm diameter, 90 mm length and 5 mm diameter for its inner axial hole. Therefore, the wall of the annulus is 3 mm thick. C_1 is a cylindrical body of size similar to that of C_0. Unlike C_0, it has 17 through-holes, which are 1 mm in diameter. These perforations are located as shown in Fig. 2, i.e. with their axes along the radial direction and separated 4 mm from each other. C_2 is a cylindrical body of the same size as C_0. It also has holes, as does C_1, but in this case, holes do not run through the cylinder walls and are only 2 mm deep.

For each measurement, cylinder temperature was raised by means of a heating device located inside the central hole in the cylinder. Basically, this device consists of a nicrome resistance. Once the cylinder have reached the steady-state temperature (T_i), the heating device was removed from the central hole and the ends of the hole were closed in order to avoid flux circulation from there. Then the temperature measurements were carried out, as a function of time, whilst the body remained immersed within an air flux. The temperature data used in the data analysis were those measured after 30 s of removing the heating device in order to any residual temperature gradients inside the body were minimized.

Cylinder temperature ($T1$) was measured with a thermocouple in contact with the inner annulus wall. This thermocouple was mounted at the middle of the annulus wall to approximately 20 mm from the end of the cylinder as shown in Fig. 3. Another thermocouple was placed at the central hole of each cylinder, allowing for the determination of the air gap temperature ($T2$) inside the cylinders (Fig. 3).

Temperature values were recorded until values of approximately room temperature were reached. The measurements were performed for different angles of incidence of the air flux on the cylinders and different values of air flux speed (V). The different angles of incidence or attack of the air flux on the cylinders are shown in Fig. 4. The parallel configuration corresponds to the holes aligned with the air flow (angle of incidence equal to 0°), the perpendic-

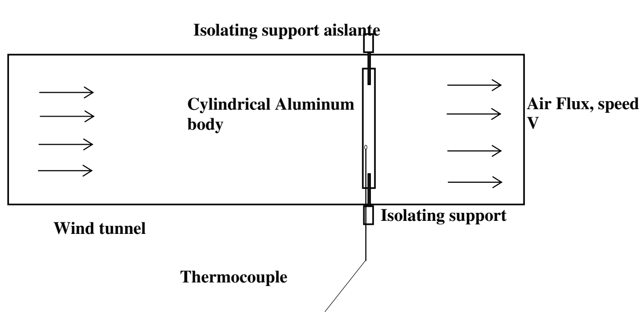

Fig. 1. Scheme for the experimental setup used in the present study.

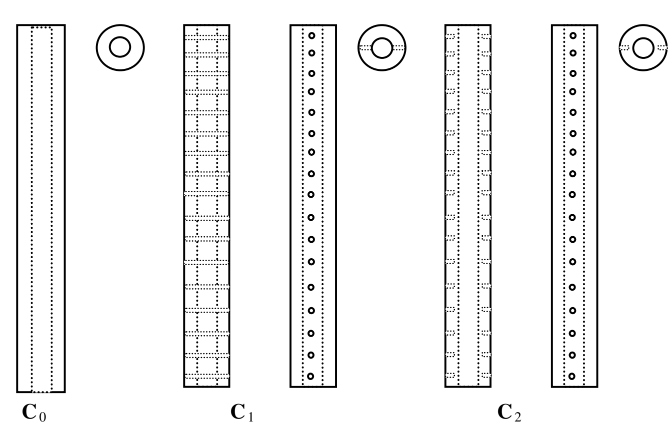

Fig. 2. Schematic view of the cylinders used to carry out the experiments.

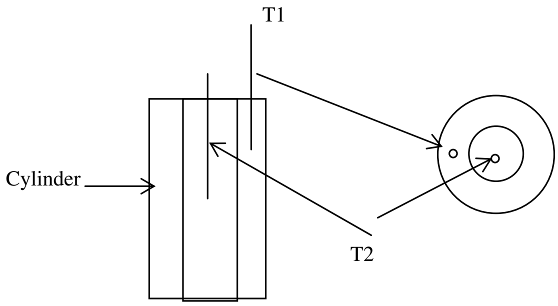

Fig. 3. Schematic view of the thermocouple localization.

ular configuration corresponds to the holes placed perpendicularly to the air flow (angle of incidence equal to 90°) and the slanted configuration to the holes placed at 45° respect to the air flow direction.

The air flow speed was varied between 4 and 22 m/s and was measured using a Pitot-tube type anemometer. The speed was determined with an error of ±0.5 m/s for velocities below 19 m/s and an error of ±1 m/s for velocities between 19 and 22 m/s. Temperature measurements were carried out using thermocouples bearing errors within the 0.5 °C range. The mass of the bodies was determined by using a balance with an error of 0.1%.

Each measurement was repeated at least five times and the mean value of the thermal time constant (τ) and standard deviation were calculated. The results show good repeatability between tests and the values of τ were determined with an error of 3–5%.

3. Results

Fig. 5 shows temperature variation ($T1$) as a function of time for the cylinders C_0 and C_1 in the parallel configuration. For both, the air flow velocity was 5 m/s. The uncertainty attached to each determination is always smaller than the size of the plotted point. The fitted exponential curves and the corresponding thermal time constant are also included in the graphic. From Fig. 5, it is possible to observe that the temperature decreases exponentially as time goes on. This satisfies the equation defined by the lumped capacitance method. A similar behaviour than $T1$ was observed for the temperature measured in the gap ($T2$). This means that the thermal time constant was the same for both temperatures. In general it was observed that $T2 > T1$. In fact, we have measured a difference $T2 - T1 =$ 0.5 °C during the cooling of the cylinders C_0 and C_2, and $T2 - T1$ up to 2 °C for C_1.

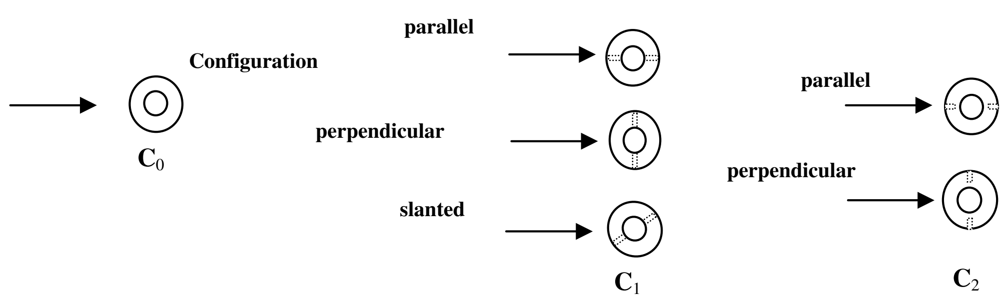

Fig. 4. Schematic view of the incidence of air flux on the cylinders.

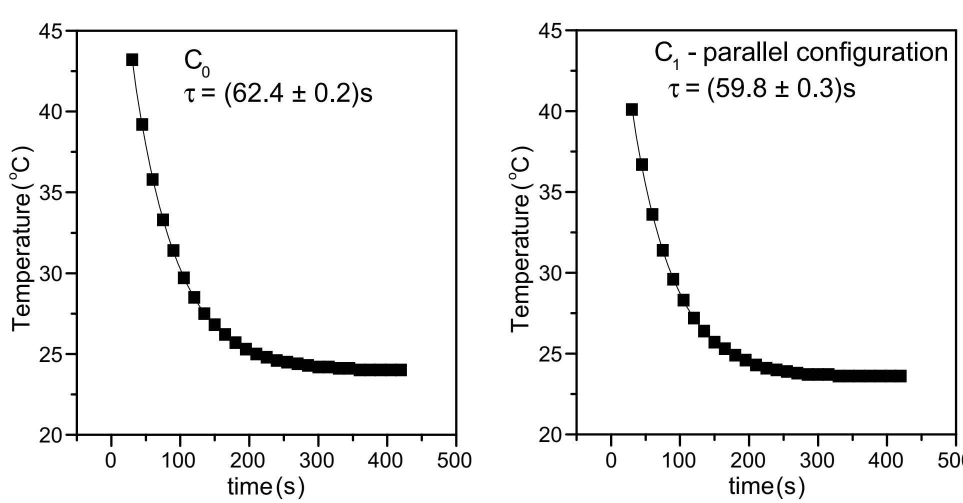

Fig. 5. Temperature vs time measurement for the smooth cylinder and C_1 in parallel configuration. The velocity was 5 m/s in both runs and τ is the thermal relaxation time.

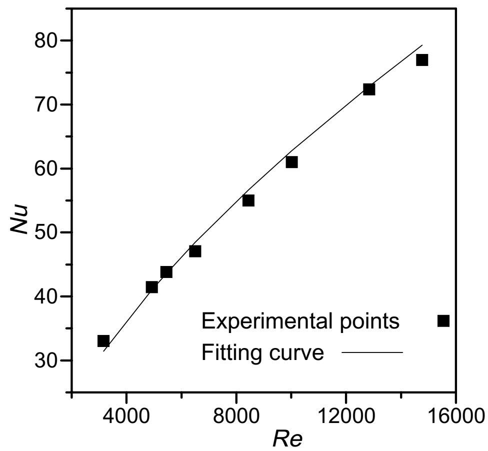

Fig. 6. Experimental values of Nusselt number vs Re for a smooth surfaced cylinder. The line represents the fitting function given by Eq. (5) with $C = 0.28$ and $n = 0.6$.

Furthermore, the experimental results obtained for the smooth cylinder C_0 are compared with results of a circular cylinder in cross flow existing in literature. Fig. 6 shows the experimental values obtained for Nusselt number (Nu) as a function of Re for the smooth cylinder, together with the fitting curve given by Eq. (5). The value $n = 0.6$ was fixed according to Incropera and De Witt [15] for circular cylinders and the value of C was adjusted by using the experimental points. The best value was $C = 0.28$ which is very close to the value of $C = 0.26$ reported by Incropera and DeWitt corresponding to $1000 < Re < 200{,}000$. It is possible to observe that the experimental data are well-fitted by the fitting curve, giving confidence in the experimental setup and showing that it is suitable to perform the measurements.

Fig. 7 presents values for the product Sh_{m} as a function of Re corresponding to cylinders C_0 and C_1, for different angles of incidence of the air flux. The origin of the error of Sh_{m} lies in the determination of τ and the mass of the cylinder. Since the error associated with the mass is less than 0.1%, then the main source of error comes from the determination of τ. On the other hand, the error on the determination of Re number involves uncertainties on the velocity and cylinder diameter. Only the errors associated to C_0 and C_1 in parallel configuration are displayed in the graphic for best clarity and detail. As can be observed, Sh_{m} values for C_1 are always larger than the ones corresponding to C_0. Also, values obtained in parallel configurations are always larger than those obtained using perpendicular or slanted configurations, these two, in turn, being generally indistinguishable from each other.

In Fig. 8, Sh_{m} values, obtained for cylinders C_0 and C_2 in parallel and perpendicular configuration as a function of Re, are presented. It is possible to observe that the values obtained in this case are indistinguishable within the experimental errors. These results indicate that no heat transfer enhancement is observed for the cylinder C_2.

In general, the presence of perforating holes seems to have produced greater heat transfer enhancement on parallel configurations. However, a heat transfer enhancement was also noted on the perpendicular configuration.

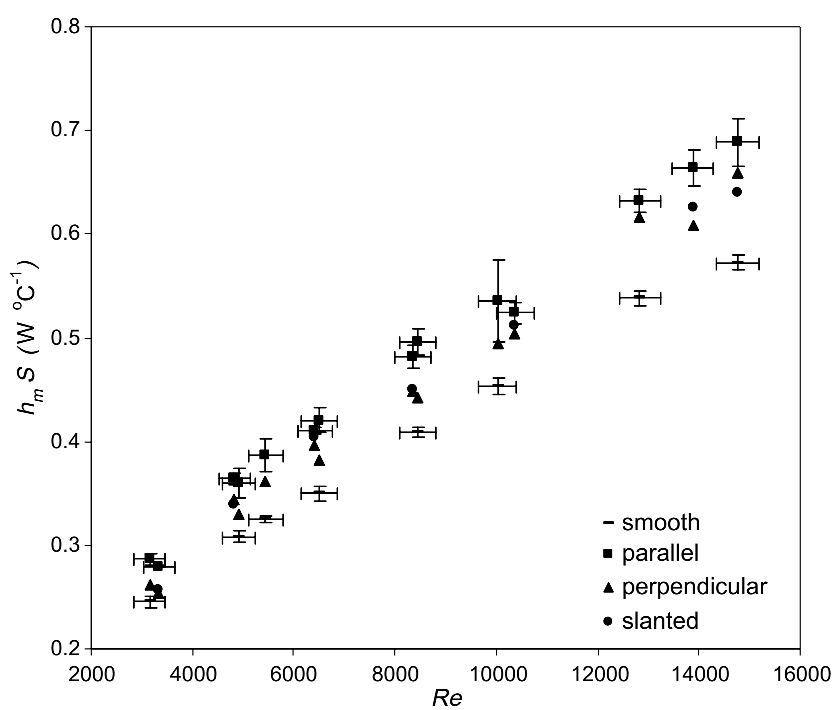

Fig. 7. Sh_{m} vs Re values obtained for C_0 (smooth) and for C_1 (air flux configurations: parallel, perpendicular and slanted).

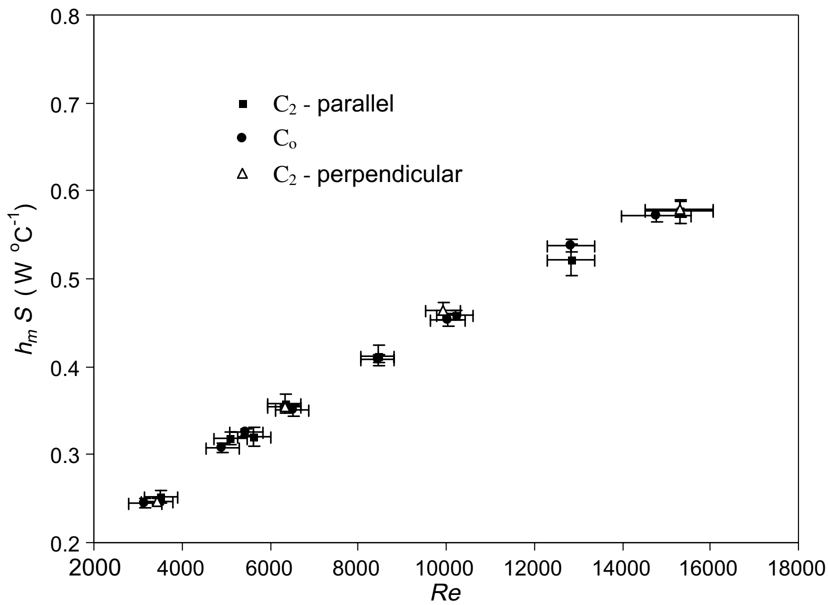

Fig. 8. Sh_m vs Re values for C_0 and C_2 (parallel and perpendicular air-flux configuration).

4. Discussion

As can be seen in Fig. 7, the values of Sh_m corresponding to C_1 in the perpendicular configuration are larger than those corresponding to C_0. The cause for this behaviour must be related to the presence of holes.

The heat transfer enhancement, as indicated by the increase of Sh_m values, could be a consequence of the heat transfer surface increment due to the presence of holes or changes in the heat transfer coefficient due to flow variations around the body; or both effects simultaneously.

The increase in the ventilation area could be related to the areas of the through-holes and the surface of the central cylinder hole. It is reasonable to assume that the heat exchanged between the body and the fluid inside the central hole do not contribute to cool the body because the temperature in the gap (T_2) was always slightly higher than the temperature at the core of the body (T_1). Then the area of the central gap does not contribute to enhance the cylinder ventilation. With a similar argument it is possible to neglect the contribution of the through-holes surface in which the air inside the hollow cylinder is ejected. Then the increase in the ventilation area comes from those holes that allow incoming airflow to enter into the cylinder. Considering the effective area contributing to the body ventilation we estimate that the area of C_1 is increased by 4% respect to the area of the smooth cylinder C_0. Although, the ventilation area increases respect to C_0, the increment is clearly not enough to explain the heat transfer enhancement observed in the current experiments. Therefore, the heat transfer enhancement for perforated cylinders should be mainly a consequence of the increment of its heat transfer coefficient.

Yajima and Sano [14] observed the behaviour of the flow around perforated circular cylinders in cross flow at similar Reynolds numbers. They have reported that for angles of incidence larger than 80°, fluid enters into the cylinder from one of the end of the holes and is ejected into the wake region from the other end, in an alternative way. They also observed turbulence in the wake region next to the holes. Then, in the present experiments, the air flux next to the holes, generated turbulence in the wake, and the turbulence improves the heat transfer process [15]. Thus, the increase in the product Sh_m, occurring in the perpendicular configuration with respect to a smooth cylinder, could be assumed to be due to the increase of both the ventilation area and value of h_m. The increase of the heat transfer coefficient would be a consequence of the presence of a turbulent flux in certain zones in the wake.

On other hand, Yajima and Sano [14] observed that for angles of incidence less than 60° the flow was sucked into the cylinder at the front holes and ejected to the wake region through the rear holes. This produces that the boundary layer near the upstream holes becomes thinner, which would be associated to an increase in cylinder heat transfer process, i.e. higher h_m values [15]. These results are in agreement with those found in the present report for cylinder C_1 in the parallel and slanted configurations, where an increase in the product of Sh_m was observed, compared to the values for a smooth cylinder.

The current results show that no heat transfer enhancement is observed for the cylinder C_2 with non-penetrating holes (Fig. 8). This seems to indicate that an essential feature for holes practiced on a cylinder to have an effect on its ventilation coefficient is that they be "through-holes". Penetrating holes ensure flow circulation inside the cylinder and allow for a reduction of the boundary layer to take place near the holes or for a significant increase in the wake turbulence during the perpendicular configuration. Thus, the air is practically stagnated inside the holes of the cylin-

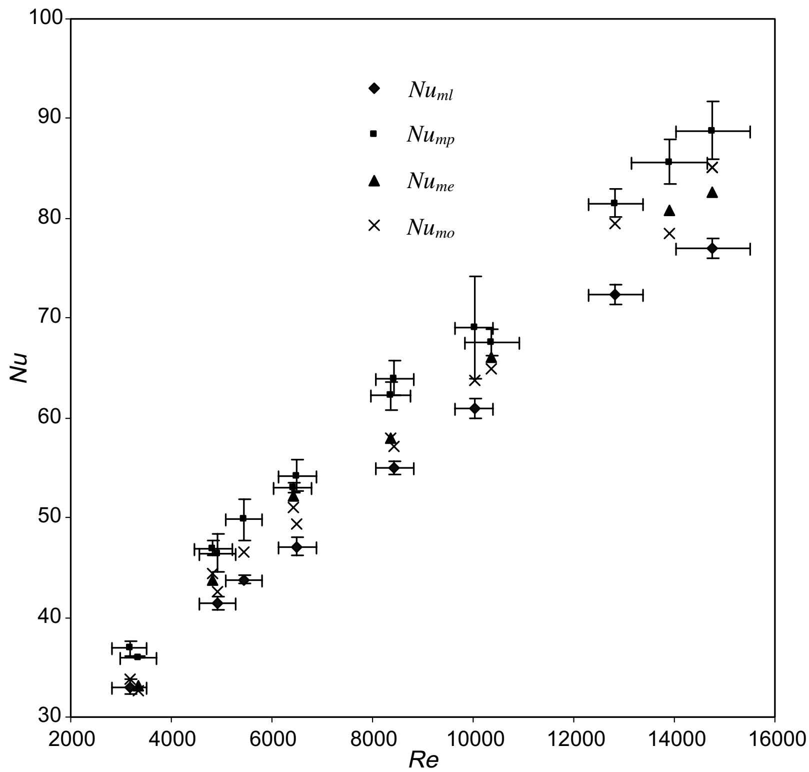

Fig. 9. Nu_{ml}, Nu_{mo}, Nu_{mp} and Nu_{me} values as a function of *Re*.

der C_2 and heat transfer process takes place by diffusion mechanism, which is less efficient than the convective mechanism.

The results indicate that small modifications on cylinder surface such as the presence of holes on cylinder C_2 do not significantly affect cylinder ventilation. This suggests that the thermal boundary layer is not substantially modified by the presence of holes. This is in agreement with results reported by Aguirre Varela et al. [17] who have observed that the ventilation coefficient for circular cylinders with slight surface modifications varies little compared to that of a smooth cylinder.

Therefore, the results obtained for C_1 using the parallel, perpendicular and slanted configurations can be directly associated to an increase in the value of the heat transfer coefficient. Fig. 9 shows Nusselt number values as a function of *Re* for a smooth cylinder (Nu_{ml}), together with: Nusselt numbers associated to h_{m} values obtained for C_1 in the parallel (Nu_{mp}), perpendicular (Nu_{mo}) and slant (Nu_{me}) configurations. These values were calculated by assuming that the increased ventilation area calculated above is involved on the heat transfer process. The figure shows that both the presence and orientation of perforating holes in bodies have a key influence on the heat transfer mechanism. Results found for C_1 using the parallel configuration exhibit an increase between 12% and 16% in the value of Nu_{mp} as compared to Nu_{ml}. In the perpendicular configuration, the increase in Nu_{mo} is between 2% and 10% as compared to Nu_{ml} and in the slanted configuration the increment is between 1% and 10% respect to Nu_{ml}.

We can conclude that the increase in the value of the heat transfer coefficient (h) as a consequence of the presence of holes could be due to different mechanisms. In the case where the holes are parallel to the air flow the increment of h could be caused by a thinning of the boundary layer, whereas in the other cases the increment of h could be due to an increase in the turbulence in the vicinity of the body.

Acknowledgements

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