Experimental assessment of a new technique for measuring heat transfer coefficients

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Abstract

A novel technique for measuring convective heat transfer coefficients is presented. The proposed steady-state approach can be applied to thermally thin sensors, made of relatively high conductivity material. Unlike the heated-thin-foil method, demanding a uniform heating of the slab, the sample can be externally heated, and the heat input is not required to be known. An experimental assessment of the technique is illustrated, dealing with the measurement of convective heat transfer coefficient distribution onto a thermally thin aluminium slab, subjected to a jet normally impinging on it. Tests are carried out by varying the Reynolds number (ranging from $2.0 \cdot 10^4$ to $5.0 \cdot 10^4$) and the nozzle-to-plate distance (from 2 to 6 nozzle diameters).

1. Introduction

The measurement of heat fluxes is of fundamental importance in countless industrial applications. Several techniques have been developed in order to achieve this goal, the most relevant part of them should be considered as zero-dimensional in terms of spatial resolution [1,2] (e.g. methods involving the use of thermocouples, pyrometers, resistance temperature detectors, etc.). On the other hand, IR thermography allows to obtain instantaneous surface temperature maps. This peculiarity makes IR thermography well suitable to the measurement of heat fluxes, even in presence of high temperature gradients, enabling the possibility to take into account the effects of the tangential conduction along the surface of the sensor.

For this reason, IR thermography has been widely used as a common tool for the measure of convective heat fluxes; e.g., it has been coupled with the heated-thin-foil method [3] for the investigation of heat transfer between a fluid and a solid surface. This technique requires uniform heating of a thermally thin sample; the main drawback is that precise information about the heating system are requested. Furthermore, it could be necessary the design of supplementary components, e.g. a printed circuit boards, in order to guarantee a uniform heating. Of course, for practical reasons, the exchanging surface should have a relatively simple geometry.

In the present work a novel steady-state approach for measuring convective heat transfer is presented. The proposed technique is mainly based on the possibility to capture two-dimensional temperature maps by means of IR thermography, and it can be applied to thermally thin sensors, made of relatively high conductivity material, i.e the Biot number $Bi=hs/k$ (where s is the slab thickness, and k is the thermal conductivity coefficient of the material) has to be small with respect to the unity. In this hypothesis it can be assumed that the temperature is constant across the slab thickness, and the temperature distribution related to the interaction between the fluid and the slab surface can be detected on the opposite face to that of heat exchange.

In Sec. 2 the main characteristics of the technique are described, and its theoretical model is illustrated. In Sec. 3 an example of application is presented, dealing with the well-known investigation of the distribution of the convective heat transfer coefficient on a slab subjected to a jet normally impinging on it. Lastly, the results are discussed and the conclusions are drawn.

2. Description of the technique

A simple characterization of the proposed heat flux sensor could be obtained by a two-dimensional energy balance, as illustrated in Eq. (1) and in figure 1. In the case of the heated thin foil sensor, it is necessary that the external heating is uniform, and this issue is typically performed by imposing a heat input associated to Joule effect. The proposed technique does not require uniform heating; furthermore, the heat input could be given externally from the measurement zone, and its value has not to be exactly known.

$$\dot{Q}_r + \dot{Q}_a + \dot{Q}_c + \dot{Q}_k = 0 \,. \quad (1)$$

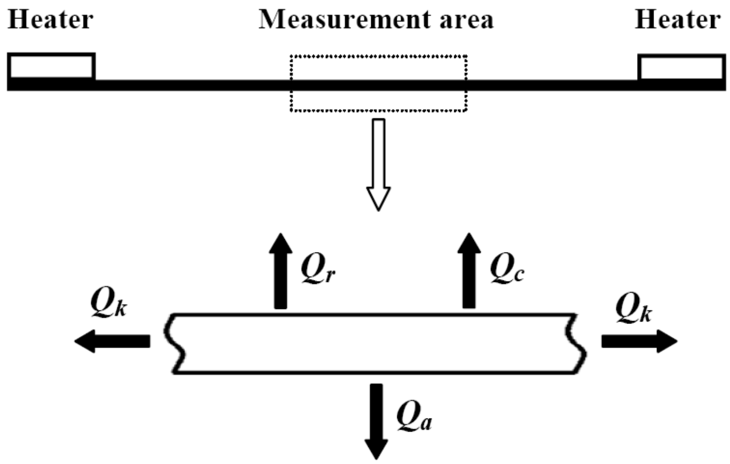

Fig. 1. *Heat flux sensor*

In Eq.1 the term $\dot{Q}_r$ is the radiative heat flux towards the flowing fluid; it can be estimated with the following formula:

$$\dot{Q}_r = \sigma\varepsilon\left(T_w^4 - T_{amb}^4\right). \tag{2}$$

In Eq. (2) σ is the Stefan-Boltzmann constant, ε is the total emissivity coefficient, T_w is the wall temperature and T_{amb} is the uniform temperature of the experimental environment.

The total heat flux to the external ambient is represented by the term $\dot{Q}_a$. Usually, this contribution is the sum of the radiative and natural convection heat fluxes from the sensor back surface, [4].

The term $\dot{Q}_c$ in Eq.(1) is the to be measured convective heat flux, expressed by means of Newton's law:

$$\dot{Q}_c = h\left(T_w - T_r\right). \tag{3}$$

where h is the convective heat transfer coefficient, whose distribution has to be determined, and T_r is a reference temperature. Different choices could be adopted for the reference temperature, depending on the flow regime: e.g. for compressible regime T_r is chosen to be equal to the adiabatic wall temperature, while for external iposonic flows it practically coincides with the fluid temperature.

The term $\dot{Q}_k$ represents the tangential conduction heat flux within the slab. Under the assumptions of thermally thin sensor (i.e. the temperature can be considered constant across the thickness of the sample) and of isotropic material, the tangential conduction heat flux can be evaluated by means of the Fourier law:

$$\dot{Q}_k = -ks\nabla^2 T_w. \tag{4}$$

The aforementioned term is the one that conveys internal energy from the heaters towards the measurement zone and, at steady state, it is spent by the other contributions. The estimation of the tangential conduction heat flux requires the knowledge of the Laplacian in Eq. (4). The measurement of two-dimensional temperature maps by means of IR thermography enables the numerical computation of the Laplacian. It's clear that this procedure is strongly affected by the unavoidable presence of both temporal and spatial noise. For this reason, temporal averaging and spatial filtering are applied as long as the desired level of signal to noise ratio has been achieved.

Taking into account all the contributions in Eq. (1), the distribution of the convective heat transfer coefficient could be determined as follows:

$$h = \frac{-\sigma\varepsilon\left(T_w^4 - T_{amb}^4\right) - \dot{Q}_a + ks\nabla^2 T_w}{\left(T_w - T_r\right)} . \qquad (5)$$

The main drawback of the technique is related to the importance of a correct estimation of the tangential conduction heat flux, which is the most relevant contribution in Eq. (5). The heated-thin-foil method is less affected by this issue, being $\dot{Q}_k$ usually a correction term, as long as the most important contribution is that of the imposed heating by Joule effect. For the aforementioned reasons, the numerical computation of the Laplacian in Eq. (5) involves some critical aspects to be considered. E.g., an improvement of the signal to noise ratio could be obtained by calculation of the numerical derivatives with a larger spatial distance; on the other hand, this expedient implies a reduction of the spatial resolution of the technique, and an operating compromise has to be found. Similar remarks can be argued regarding the amplitude of the spatial filtering window.

Another critical aspect is related to the minimum difference between the wall temperature and the reference temperature. As a matter of fact, a small difference gives rise to a lower Laplacian value and, therefore, to a higher importance of the noise affecting the computation. Furthermore, it determines also a greater relative importance of the error in the measurement of both the reference temperature and the wall temperature.

Summarising, the application of the proposed technique requires a careful preliminary study. The choice of the geometrical parameters and of the characteristics of the material is strictly connected to the phenomenology to be investigated.

3. Experimental setup and application

An example of application of the proposed technique is presented. The task is the measurement of the distribution of the convective heat transfer coefficient onto a thermally thin slab subjected to a jet normally impinging on it. A sketch of the experimental apparatus is reported in figure 2.

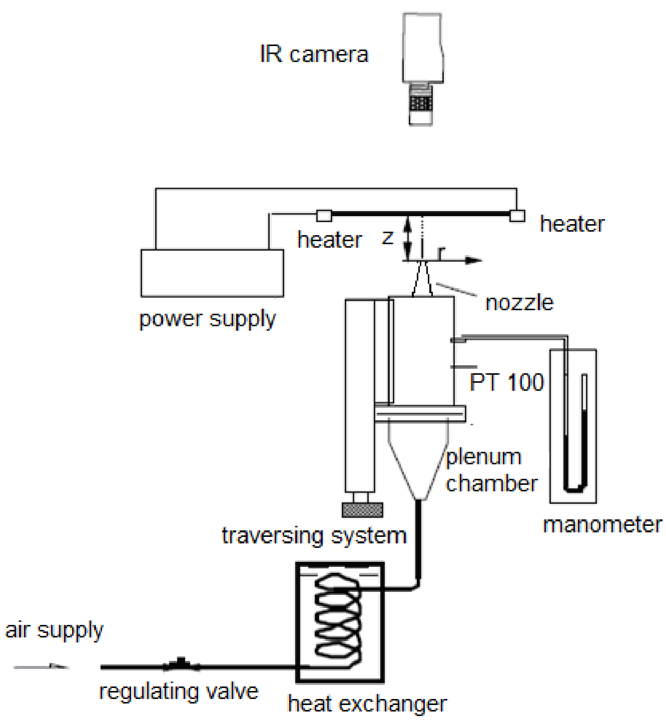

Fig. 2. *Sketch of the experimental setup*

The air supply is provided by a compressor; subsequently the fluid flow passes through a pressure regulating valve and a heat exchanger, in order to obtain a jet temperature close to the ambient temperature. The control on the jet temperature is of great importance for the reasons explained in the previous section. The total pressure and temperature are measured in the plenum chamber by means of a manometer and a Pt 100 RTD, respectively. Finally, the air expands through a convergent axisymmetric nozzle with an exit diameter of 18.7 *mm*, and it impinges on the plate, which characteristics are reported in table 1.

Table 1. *Physical and geometrical material properties*

Alluminium alloy	Al - 3105
Thermal conductivity	185 W/m•K
Length	300 mm
Width	300 mm
Thickness	1.1 mm

The plate is horizontally placed in order to reduce the effects of hot air recirculation in the measurement zone. Because of the strong convection due to the jet, the natural convection contribution is practically negligible in all tested conditions.

The plate is heated by means of two electric heaters, disposed at opposite positions with respect to the central point of impingement (i.e. the projection of the nozzle axis), at a mutual distance of 150 *mm*. In order to demonstrate that the feasibility of the technique is not strictly connected to the symmetry of the problem, the heating of the plate is performed with a non axisymmetric configuration, as illustrated in figure 3.

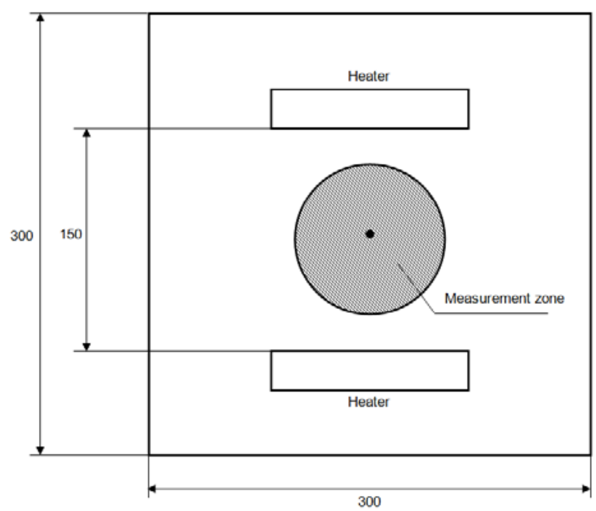

Fig. 3. *Scheme of the configuration of the heaters in the plan view of the plate*

Even if compared to a relatively high local convective heat transfer coefficient, the rather small thermal thickness of the aluminium slab determines a very small Biot number. For this reason the temperature is practically constant across the thickness of the slab, allowing the possibility to measure the surface temperature distribution by means of an IR camera system observing the opposite face to that of exchange. An IR camera FLIR SC 6000 LW operating in 8-12 *µm* is employed in order to measure the surface temperature map.

Tests are carried out by varying the nozzle-to-plate distance; three dimensionless distances *z/D* are considered, i.e. 2, 4 and 6, being *D* the nozzle exit diameter. Tests are also performed by varying the Reynolds number. Four different values are tested, i.e. $2.0 \cdot 10^4$, $3.0 \cdot 10^4$, $4.0 \cdot 10^4$, $5.0 \cdot 10^4$.

For each test, the heat input is chosen in order to obtain on the slab a minimum temperature difference of at least 15 *K* with respect to the jet temperature.

The physical and the geometrical parameters of the to be investigated phenomenology have to be carefully chosen. E.g. the distance between the heaters, the thermal thickness of the slab and the outflow conditions directly influence the distribution of the convective heat transfer, and, as a consequence, the required heat input in order to obtain a suitable temperature distribution for the numerical computation of the Laplacian. For this reason, a careful preliminary study of the phenomenology must always forego the application of the technique.

4. Results and discussion

Convective heat transfer coefficient maps are presented in non-dimensional form, in terms of the Nusselt number, based on the jet diameter *D*, as expressed in the following:

$$Nu = \frac{hD}{\lambda}. \quad (6)$$

where λ is the thermal conductivity of the fluid.

Firstly, the effect of different source of measurement noise, affecting the computation of the Laplacian, has been investigated.

The results have shown that the temporal noise can be consistently reduced if each temperature map is obtained by means of averaging a relative high number of images. In the present application 100 instantaneous images are averaged for each temperature map.

The employment of IR cameras with Focal Plane Array introduces another critical source of error, i.e. the spatial measurement noise, generated by differences in the response of each pixel. For this reason the application of an accurate and reliable process of Non Uniformity Correction (NUC) is of fundamental importance. The results have demonstrated that both spatial filtering and computation of the derivatives with a relative wide step are not sufficient to compensate the decay of the signal-to-noise ratio due to the non uniformities of the response of each pixel. In order to make clear this statement, the steady-state temperature map for the case of Re = $5.0 \bullet 10^4$ and *z/D* = 2 is presented in figure 4. A white cross identifies the central point of impingement. In the IR image, the two electric heaters are disposed at the top and the bottom, at the boundaries of the field of view. Obviously the temperature distribution is not axisymmetric because of the non axisymmetric location of the heaters.

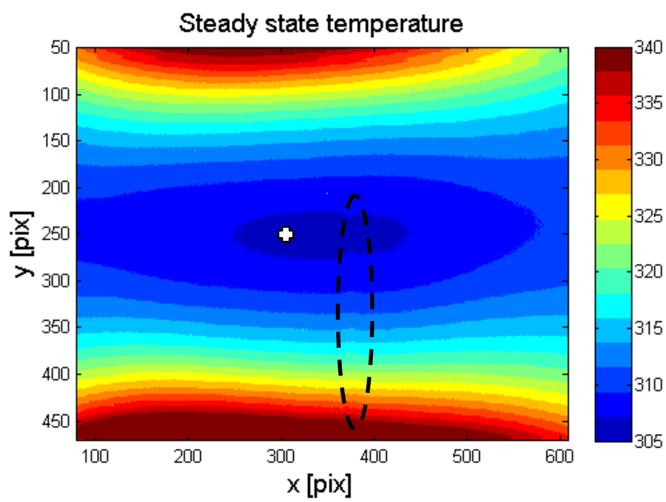

Fig. 4. *Steady-state temperature distribution for Re=$5.0 \bullet 10^4$ and z/D=2. The white cross indicates the central point of impingement. The dashed black oval encloses the area mainly affected by the non uniform response of the pixels.*

In the temperature map of figure 4 the presence of a strip of temperature discontinuity, related to a non complete correction of the non uniform behavior of the pixels, can be observed nearby $x \approx 370$. The effect is more clearly highlighted in figure 5, in which the horizontal temperature profile passing through the central point of impingement is illustrated. The measured and filtered temperature profile are compared; the latter has been obtained by applying a Gaussian spatial filter with a kernel 30x30 pixels and with a standard deviation equal to 6. Even if the error is relatively small (the order of magnitude is 0.1 K), it determines noticeable consequences in the computation of the Nusselt number distribution (obtained by adopting a step of 30 pixel, i.e. about one nozzle diameter for the chosen optical configuration, for the computation of the numerical derivates), which does not appear axisymmetric, as shown in figure 6. Evidently all the measurement points with a distance from the pixels affected by the offset minor than the adopted step for the computation of the numerical derivatives are involved, and their corresponding value of Nusselt number is clearly invalidated.

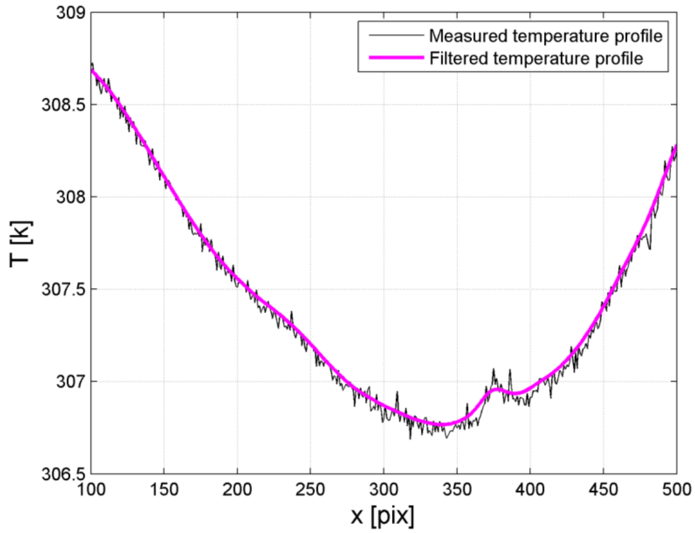

Fig. 5. *Temperature profile before and after the filtering for Re=5.0•10^4 and z/D=2*

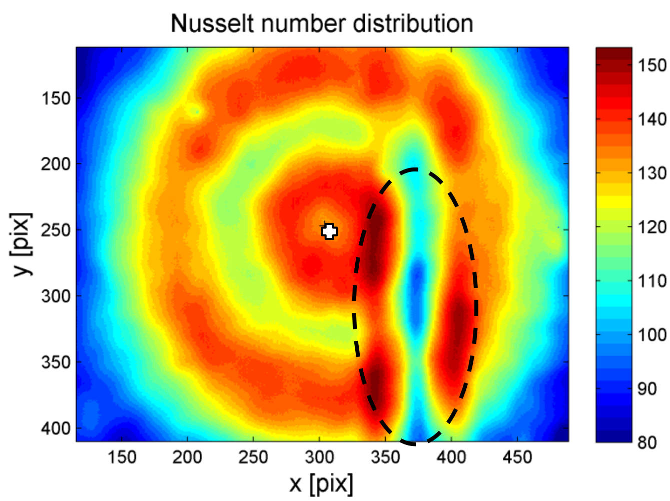

Fig. 6. *Nusselt number distribution for Re=5.0•10^4 and z/D=2. The dashed black oval encloses the area mainly affected by the non uniform response of the pixels.*

In the following the results of the performed tests are illustrated. The profiles of the Nusselt number, obtained by means of an azimuthal average only in the region not affected by the problem of a non correct NUC, are presented for the several tested conditions. The results are obtained by adopting a Gaussian spatial filtering window with a kernel 30x30 pixels, and a standard deviation equal to 6; a further increase of the signal-to-noise ratio is recovered by computation of the numerical derivatives with a spatial step of one nozzle diameter. In figure 7 several profiles are plotted for the tested values of Reynolds number and with a fixed nozzle-to-plate distance. The results are in agreement with that of the literature, [5-9]. Depending on the jet Reynolds number and on the distance between the outflow section of the nozzle and the plate, one or more radial peaks are detected. For z/D = 2 an inner radial peak is observed at approximately r/D = 0.5. Experimental results show that this peak becomes slightly more intense with increasing of the Reynolds number [9]. An outer peak is observed at approximately r/D = 1.8, and it also appears to boost in intensity with increasing of the Reynolds number For z/D = 4 an inner peak is also revealed at approximately r/D = 0.4, while a region of maximum Nusselt number could be observed close to a radial distance of two diameters.

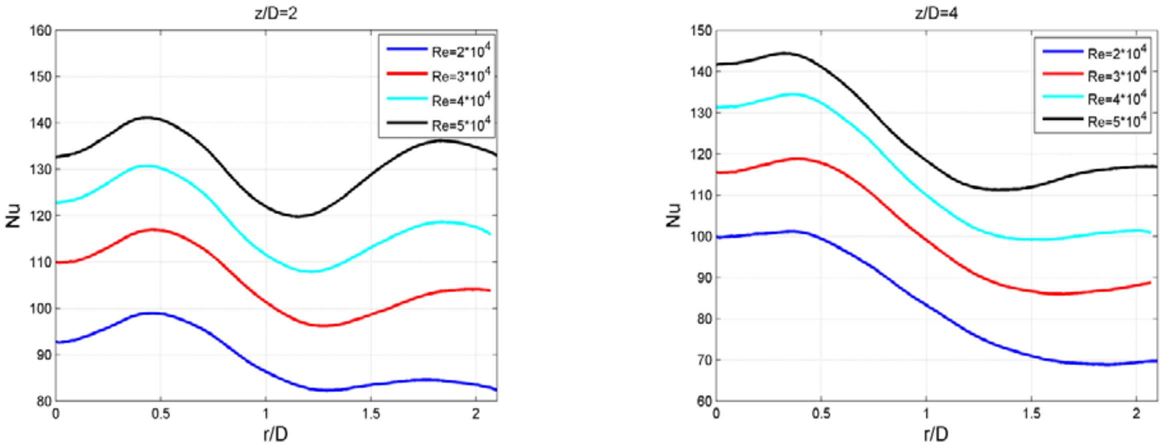

Fig. 7. *Azimuthal average of the radial distribution of Nusselt number in the tested range of Reynolds numbers with fixed nozzle-to plate distance: on the left z/D=2; on the right z/D=4.*

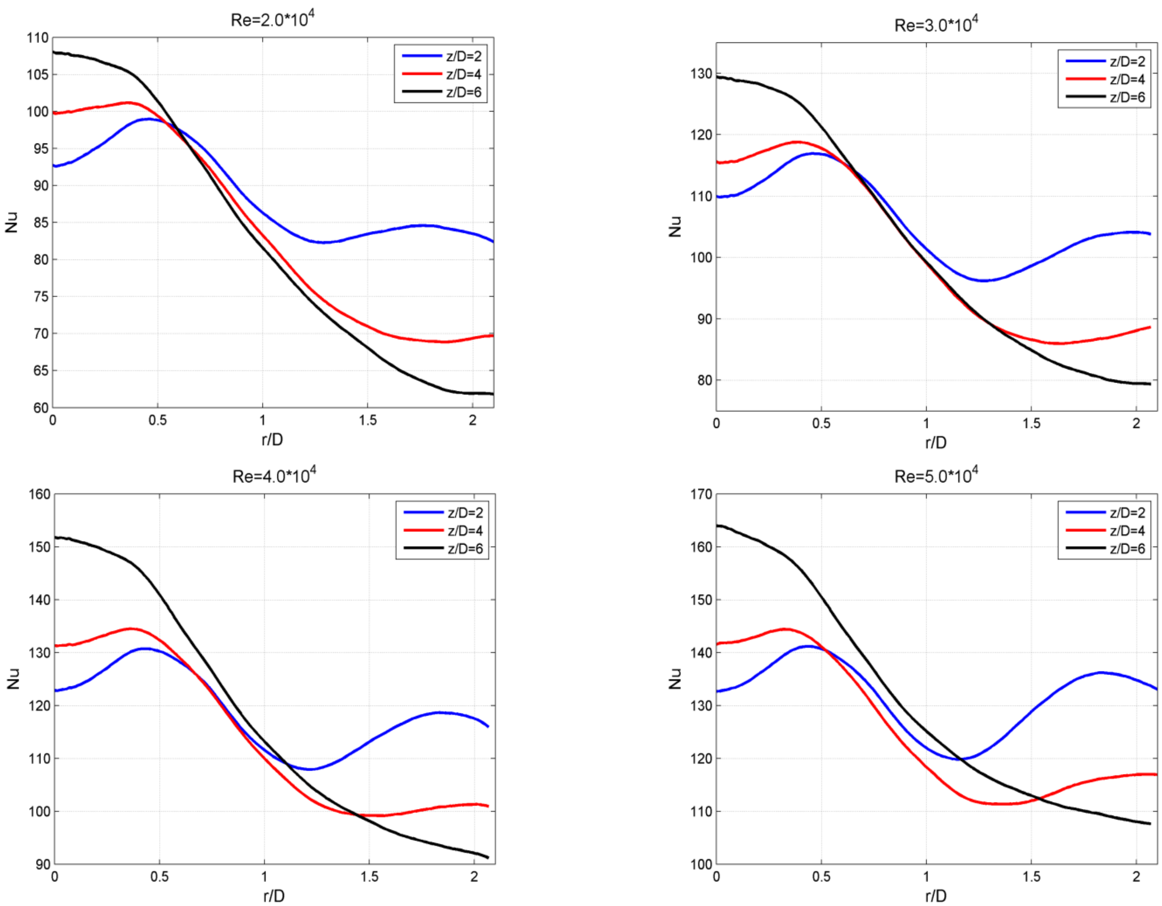

Fig. 8. *Azimuthal average of the radial distribution of Nusselt number in the tested range of Reynolds numbers and of distance between the nozzle exit and the slab.*

Finally, for *z/D* = 6, the maximum heat transfer point coincides with the stagnation point of the jet., as illustrated in figure 8.

In spite of the reduction in spatial resolution connected to the employment of a smoothing Gaussian filter and the calculation of numerical derivative with a relatively wide step, the technique shows to be able to detect these slight local variations, providing reliable results.

5. Conclusion

A new steady-state approach for measuring convective heat transfer coefficients is presented. The characteristics of the proposed heat flux sensor have been illustrated, and the relative drawbacks and critical aspects are reviewed. A preliminary analysis shows that a crucial aspect regards the numerical computation of the Laplacian of the temperature distribution. A smoothing of the signal with a Gaussian filter and the calculation of the numerical derivative with a relative wide step are suggested. Another critical aspect is connected with the use of Focal Plane Array infrared cameras, for which it is necessary to compensate the different response of each pixel of the array. A single detector camera would probably perform better.

An example of experimental assessment of the technique is illustrated, concerning the heat transfer between a plate and a jet normally impinging on it. The results show that the dimension of the Gaussian filtering window of the basis step for the calculation of the derivative, and the geometrical parameters of the experimental setup have to be chosen in strict relation with the physics of the problem.

To identify the heat flux sensor that makes use of the above described technique, the authors suggested to call it with the name *Laplacian sensor,* [10].

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