Local convective heat transfer identification by infrared thermography from a disk mounted on a cylinder in air crossflow

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Abstract

In this study, the local convective heat transfer from a disk was evaluated using an infrared thermographic experimental setup. Solving the inverse conduction heat transfer problem allows the local convective heat transfer coefficient to be indentified. We used an inverse method, based on spatial regularization, in order to take radial and angular conduction into account. This model was tested using crossflow Reynolds numbers between 11350 and 39600, corresponding to the turbulent flow domain. In this paper, the local convective heat transfer distribution on the disk allows us to study the boundary layer development with Reynolds number upstream the cylinder such as horseshoe vortex and their impact on heat transfer around and downstream the cylinder.

1. Introduction

Convective heat transfer and fluid flow around a disk in air crossflow find an application in the standard heat exchangers as well as in engineering and electronic component cooling for railroad and automobile vehicles, turbomachinery, disk brake cooling and also computer industry. Many investigations on the local convective heat transfer and flow on this configuration have been performed. In the general case of a fixed disk in air crossflow, Wiesche [1] has numerically studied local convective heat transfer for air crossflow Reynolds numbers $Re$ varying from $10^4$ to $10^5$ by large-eddy-simulation. His results are in accordance with those of the flat plate, characterized by a symmetric Nusselt number. Indeed, a high convective heat transfer occurs at the beginning of the disk, in the developing boundary layer, and decreases along the disk diameter. These observations correspond to the laminar boundary layer theory described by Incropera [2] and Ochoa [3]. To determine the Nusselt number $Nu_x$ based on the distance from the leading edge $x$, for a uniform heat flux boundary conditions, a correlation is proposed by Incropera [2]:

$$Nu_x = \frac{h \cdot x}{k} = 0.453 \cdot Pr^{1/3} \cdot Re_x^{0.5} = 0.453 \cdot Pr^{1/3} \left( \frac{U \cdot x}{V} \right)^{0.5} \quad (1)$$

However, our study deals with a disk mounted on a shaft perpendicular to an air crossflow. In this case, the presence of the cylinder generates flow perturbations, at the disk/cylinder junction, studied by different authors [4-9] for the laminar flow domain. From experimental observations, they detected a boundary layer development from the leading edge of the disk associated to a reduction of velocity due to the adverse pressure gradient in the stagnation zone upstream of the cylinder. This causes the flow to separate and to form a horseshoe vortex system, consisting of counter-rotating vortices swept around the cylinder base. Experimental studies [10-12] dealing with the visualization of the horseshoe vortex system by Particle Image Velocimetry highlighted a significant influence of $Re$ on the size of the vortex horseshoe. Moreover, Fu et al. [11] show that the instability of the horseshoe vortex generates flow perturbations in the near wake and fluctuation level increasing in the horseshoe vortex leads to the development of coherent vortices earlier in the separating shear layer. Bey [13] studies experimentally the characteristics of the flow developed in plate-fin-and-tube heat exchangers. The author shows that for $1400 < Re < 3320$ and a dimensionless fin spacing $u' = u/D$ between 0.13 and 0.27, the vortex number and size increase with Reynolds number and the wake zone downstream the cylinder tends to grow up with the velocity.

In order to study vortex structure impact on local convective heat transfer on a fin, Abbadi [14] developed a transient identification method of the local heat transfer coefficient. For a fixed fin spacing and $1400 < Re < 3840$, the author observed that the position of the heat transfer peaks due to vortex is not very influenced by airflow velocity. However, heat transfer in the wake zone increases with the airflow velocity. Moreover, a previous thermal study using a radial identification method (1D) has been realized at the laboratory [15] and shows that, for $5700 < Re < 39600$, the zones of lower convective heat transfers correspond to the wake and to the flow separation located at $\theta = \pm 90^\circ$ from the front stagnation point. On the other hand, zones of higher heat transfers are located at $+110^\circ < \theta < +150^\circ$ and $-110^\circ < \theta < -150^\circ$, where the legs of the horseshoe vortex system appear.

The experimental works presented in this paper allow the airflow velocity influence on the convective heat transfer at the disk surface to be studied. Indeed, a bidimensional inverse method using infrared thermographic measurements has been developed to indentify the convective heat transfer coefficient on the disk. In this method, the infrared camera measures the temperature variations of the disk in the radial and angular directions over time. Solving
the inverse conduction heat problem allows us to identify the local heat transfer coefficient in all the experiments carried out while taking into account both radial and angular conductive fluxes as well as radiative flux [15,16].

2. Experimental study

2.1. Experimental setup

Local convective heat transfer coefficient at the surface of a disk in air crossflow was obtained by measuring the radial and angular temporal temperature variations on the disk (figure 1). The 2-mm wide and 60-mm high disk was made of aluminium ($\lambda_{al} = 200 \text{ W.m}^{-1}\text{.K}^{-1}$, $\rho_{al} = 2700 \text{ kg.m}^{-3}$, $C_{al} = 0.96 \text{ kJ.kg}^{-1}\text{.K}^{-1}$) and was mounted on a 58-mm diameter cylinder. The disk and cylinder were covered with a thin coat of black paint, whose relatively high emissivity allowed the radiative heat flux emitted to be more accurately determined, thus improving the accuracy of the relationship between the camera exit signal and the disk temperature. The experiments are performed in a 2.4 m long wind-tunnel. The test disk is positioned at 2-m distance from the fan in order to obtain a straightened airflow upstream from the disk. A diaphragm allows us to vary airflow velocity $U$ from 4 to 14 m.s$^{-1}$. In order to measure disk temperatures by thermography on the entire surface, the front wall of the wind tunnel includes a porthole made up of an infrared transparent film. The calibration law has been determined with an extended black body at the lab.

A radiant panel emitting short infrared waves was placed horizontally above the disk, heating them uniformly to temperatures about 120°C. Once the steady thermal state was reached, the heat source was shut off. The disk then cooled through radiative and convective heat transfers, depending on air crossflow velocity. The surface temperatures during the cooling of the disk were recorded using an infrared (IR) camera (JADE 3 from CEDIP INFRARED SYSTEMS). In our study, the camera was placed 2.50 m in front of the disk, with an acquisition frequency of 600 Hz and a window of 160x120 pixels, where each pixel corresponds to almost 2.5 mm$^2$ of the disk surface. Figure 2 shows the real image taken by the camera during the cooling.

The camera exit signals obtained during the cooling were collected and processed with the software MATLAB to determine the disk temperature profiles for each pixel. They are expressed in thermal levels and the processing of them is explained in detail in the next part, as well as the radiative flux. An infrared pyrometer was placed perpendicular to the cylinder's rotational axis in order to measure the cylinder's temperature evolution over time. The air temperature measured by a K-thermocouple 0.5 m from the disk while they were cooling was used as the reference temperature.

2.2. Disk temperature computation

During the cooling, for each time $t$, the infrared images taken by the IR camera show thermal levels $I_{meas}$, proportional to the radiosities reflected by the disk. The objective of this section is to determine the relationship between each thermal level $I_{meas}$ and the corresponding temperature of the disk $T_{disk}$. In this way, we used the camera calibration law and the expression of the radiative heat flux density $J_{disk}$ emitted by the disk surface. This one depends on the incident radiation on the camera sensors $J_{meas}$ and the atmosphere radiation $J_{atm}$. The radiosity $J_{meas}$ reaching the camera is weakened by the atmosphere, thus yielding:

$$J_{meas} = \tau_{atm} \cdot J_{disk} + \left(1 - \tau_{atm}\right) \cdot J_{atm}$$

(2)
Where $\tau_{atm}$ represents the atmospheric transmissivity. Moreover, in Eq. (2), the radiosity $J_{disk}$ depends on radiosities $J_i$ emitted by $n$ surfaces $S_i$ around the disk. So, the expression of $J_{disk}$ could be written:

$$J_{disk} = \varepsilon_n \cdot \sigma \cdot T_{disk}^4 + (1 - \varepsilon_n) \cdot \sum_{j=1}^{n} F_{disk-j} \cdot J_j$$ \hspace{1cm} (3)

The cylinder’s temperature $T_{cyl}$ is given by the pyrometer and the atmospheric temperature $T_{atm} = T_\infty$ is measured by the K-thermocouple. Black paint emissivity has been experimentally determined at the lab and depends on temperature $T$ as follows:

$$\varepsilon_n = 0.93 \text{ for } 293K \leq T \leq 353K$$

$$\varepsilon_n = 0.98 - 5.7829 \times 10^{-4} (T - 273) \text{ for } 353K < T \leq 413K$$ \hspace{1cm} (4)

Eq. (3) leads to the following equation:

$$J_{disk} = a_1 \cdot \sigma \cdot T_{disk}^4 + a_2 \cdot \sigma \cdot T_{cyl}^4 + a_3 \cdot \sigma \cdot T_{\infty}^4$$ \hspace{1cm} (5)

With:

$$a_1 = \frac{\varepsilon_n}{(1 - \varepsilon_n)^2 \cdot F_{disk-cyl} \cdot F_{cyl-disk}}$$

$$a_2 = \frac{\varepsilon_n (1 - \varepsilon_n) \cdot F_{disk-cyl}}{(1 - \varepsilon_n)^2 \cdot F_{disk-cyl} \cdot F_{cyl-disk}}$$

$$a_3 = \frac{(1 - \varepsilon_n) \left( (1 - \varepsilon_n) \cdot F_{disk-cyl} \cdot F_{cyl-atm} + F_{disk-env} \right)}{(1 - \varepsilon_n)^2 \cdot F_{disk-cyl} \cdot F_{cyl-disk}}$$ \hspace{1cm} (6)

By using $I(T)$ the thermal level of the camera exit signal when the IR camera receive only the radiation $\sigma \cdot T^4$ emitted by a black body at temperature $T$, obtained by the camera calibration law (figure 3), Eq. (5) can be expressed as:

$$I_{disk} = a_1 \cdot I(T_{disk}) + a_2 \cdot I(T_{cyl}) + a_3 \cdot I(T_{\infty})$$ \hspace{1cm} (7)

Using Eqs. (2) and (7), it is possible to write:

$$I_{meas} = \tau_{atm} \left[ a_1 \cdot I(T_{disk}) + a_2 \cdot I(T_{cyl}) + a_3 \cdot I(T_{\infty}) \right] + (1 - \tau_{atm}) \cdot I(T_{\infty})$$ \hspace{1cm} (8)
Finally, the spatio-temporal evolution of the disk temperature $T_{\text{disk}}(r,\theta,t)$ (figure 4) has been computed by Eq. (8) and the camera calibration law $I(T)$, determined using the reference temperature of the extended black body:

$$ I(T) = \frac{I_{\text{max}} - (1 - \varepsilon_s) \cdot I_{\text{cal}}}{T_{\text{amb}} \cdot \varepsilon_s} $$

(9)

3. Identification method

The objective of these works is to identify the convective heat transfer variations on the disk with radial and angular position and air crossflow velocity using spatial and temporal variations of the disk temperature $T_{\text{meas}}(r,\theta,t)$ measured by infrared thermography. This identification method requires the development of:

- a direct model which allows the spatio-temporal evolution of the disk temperature $T_{\text{cal}}(r,\theta,t)$ to be computed by fixing a local distribution $h(r,\theta)$,
- an inverse method, which allows the distribution $h(r,\theta)$ to be determined by comparing the computed and measured temperature evolutions during the cooling process.

3.1. Direct model

The direct model involves solving partial differential equations related to the cooling of the disk. The 2D equation system used in this direct model allows taking both the radial and angular conductive fluxes into account:

$$ \frac{\rho \cdot C_{\text{al}}}{\lambda_{\text{al}}} \cdot \frac{\partial T_{\text{cal}}(r,\theta,t)}{\partial t} = \frac{\partial^2 T_{\text{cal}}(r,\theta,t)}{\partial r^2} + \frac{1}{r} \cdot \frac{\partial T_{\text{cal}}(r,\theta,t)}{\partial r} - \frac{1}{r^2} \cdot \frac{\partial^2 T_{\text{cal}}(r,\theta,t)}{\partial \theta^2} + \frac{\phi_{\text{conv}}(r,\theta,t) + \phi_{\text{rad}}(t)}{e} $$

(10)

In this model, the convective heat flux density $\phi_{\text{conv}}(r,\theta,t) = h(r,\theta) \cdot (T_{\text{cal}}(r,\theta,t) - T_{\text{env}})$, characterized by a local time-averaged heat transfer coefficient $h(r,\theta)$, and the radiative heat flux density $\phi_{\text{rad}}(t)$ appear in the heat equation. The temperature distribution at the initial time and the boundary conditions on the inner and outer radii ($r_1$ and $r_2$) of the disk are obtained from IR camera thermal levels. At $\theta = 0 = 2\pi$, the temperature is obtained by a revolution condition. To solve the direct model, the local distribution of heat transfer coefficient $h(r,\theta)$ is assumed to be known. The equations in the system (Eq. (10)) are discretized and then solved by finite differences, with an implicit scheme using right differences for first-derivative terms and central differences for second-derivative terms.

3.2. Inverse method

The inverse model allows the distribution $h(r,\theta)$ to be determined by comparing the computed and measured temperature evolutions during the cooling process. In order to reduce the effect of measurement noise on the parameter to be identified, we used 2D spatial regularization. As explained by Beck et al. [17] and Tikhonov et al. [18], this model involves searching for the distribution $h(r,\theta)$ that will minimize the following function:

$$ S = \sum_r \sum_\theta \left[ T_{\text{cal}}(r,\theta,t) - T_{\text{meas}}(r,\theta,t) \right]^2 + \alpha_1 \cdot \sum_r \sum_\theta \left[ \nabla(h(r,\theta)) \right]^2 $$

(11)

Where $\alpha_1$ is the regularization parameter of the model. The regularization term $S_1$ is added to a function specification method in order to obtain a stable solution in spite of the measurement noise. The $S_1$ term that we used was a first-order term that reduces the wide variation of the solution due to the noise [14,15,18]. To minimize $S$, we added a correction to the imposed distribution $T_{\text{imposed}}(r,\theta)$ (which was arbitrarily chosen), using an iterative identification process that stops when the derivative with $h$ of Eq. (11) tends to zero [14,15,18].

The procedure to validate this method was the same as explain in a previous article [15]. A spatial variation of the heat transfer coefficient was chosen (figure 5) and the direct model was used to calculate the temperature histories at radial locations. Then these data were contaminated by noise to simulate real temperature measurements. Next, these
simulated measurements were introduced in the inverse method to recover the evolution of the heat transfer coefficient. The correctness of the inversion was evaluated by comparing the exact and estimated heat transfer coefficients. Figures 6 and 7 present the results obtained from temperatures contaminated by the noise of standard deviation $\sigma = 0.1$, respectively with and without regularization. Adding the regularization term $S_1$ to the specification function allowed us to recover the theoretical profile of the heat transfer coefficient (figure 5) in spite of the measurement noise.

The value of the hyper-parameter $\alpha_1$, occurred in Eq. (11), was determined from the “U-curve” presented on figure 8 [15]. This method consists of tracing the condition number $\text{cond}(Y)$ of the matrix to be inversed versus $\alpha_1$. The optimal value is located at the minimum of the “U-curve” which has a parabolic shape (figure 8). The spatial resolution applied to this method is defined by the radial step $\Delta r = 3.1$ mm and angular step $\Delta \theta = 6^\circ$.

3.3. Uncertainties

The uncertainty computation related to the convective heat transfer coefficient $h(r, \theta)$ requires the evaluation of measurement uncertainties on the transfer. Like our previous study [15], we use a sequential disturbance method [19] to determine the overall uncertainty in the results by combining the uncertainties induced by entry parameter variations. For example, we present the results obtained for the case $Re = 33900$. The disturbance generated by each entry parameter is computed by the MATLAB identification program and presented in table 1. Finally, for this case, the overall uncertainty is evaluated at 21.2 %.

<table>
<thead>
<tr>
<th>$x_i$</th>
<th>$\rho_{al}$</th>
<th>$C_{al}$</th>
<th>$\lambda_{al}$</th>
<th>$T_\infty$</th>
<th>$\varphi_{rad}$</th>
<th>$T_{\text{disk}}(r_i, \theta, t)$</th>
<th>$T_{\text{disk}}(r_e, \theta, t)$</th>
<th>$T_{\text{disk}}(r, \theta, t=0)$</th>
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<td>$dx_i$</td>
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<td>20</td>
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<td>0.7</td>
<td>29</td>
<td>1</td>
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<td>0.0282</td>
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<td>0.0114</td>
<td>0.0071</td>
<td>0.14</td>
<td>0.15</td>
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4. Results and discussion

We present in this section the analysis of convective heat transfer on the disk for the cases representative of the physical phenomena observed for $11300 < Re < 39600$. The thermal results are expressed in terms of Nusselt number $Nu$, characteristic of the convective heat transfer. Physical properties were evaluated at the air film temperature, defined by $T_f = (T_{disk} + T_{∞})/2$. The identification method presented in section 3, representative of the disk thermal behaviour with the radial and angular conductive fluxes taking into account, allows the determination of the local convective heat transfer coefficient, leading to a image of the local Nusselt number $Nu(r, θ)$ on the disk for $34 \text{ mm} < r < 84 \text{ mm}$.

![Fig. 9. Local Nusselt number on the disk](image-url)
Figure 9 highlights a horizontal symmetry for thermal results in spite of turbulence phenomena which appear near the cylinder and in the wake zone. For all the cases, the spatial resolution, defined by $\Delta r = 3.1$ mm and $\Delta \theta = 6^\circ$, allows studying the real thermal behaviour of the disk. On the disk front part, for all the tests carried out, we can identify two zones. In the first zone, the high convective heat transfer is identified at the leading edge, corresponding to the boundary layer development and characterized by $Nu = 78$ for $Re = 11300$, $Nu = 132$ for $Re = 22600$ and $Nu = 205$ for $Re = 33900$ (figure 8).

\[
\frac{r_{e}}{r_{crit}} = 0.062 - 3 \cdot 10^{-7} \cdot Re
\]  

Then, the convective heat transfer decrease with the boundary layer thickness increases to a critical radius, corresponding to the convective heat transfer minima position, $r_{crit} = 0.059$ m for $Re = 11300$ (figure 10), $0.056$ m for $Re = 22600$ and $0.053$ m for $Re = 33900$. From our measurements $11350 < Re < 39600$, we can deduce an expression of $r_{crit}$ with $Re$ (figure 11):

From our tests, we can deduce an equation which describes this ratio evolution with the Reynolds number for $11350 < Re < 39600$:

Moreover, our study allows quantifying the convective heat transfer drop, characterised by the ratio of minimum and maximum Nusselt number, respectively identified at $r = r_{crit}$ and $r = r_{e}$ (figure 10). From our tests, we can deduce an equation which describes this ratio evolution with the Reynolds number for $11350 < Re < 39600$:
\[ \frac{Nu_{\text{min}}}{Nu_{\text{max}}} = 9.3 \cdot 10^{-6} \cdot \text{Re} + 0.303 \]  

(13)

The less the Reynolds number, the more the Nusselt number drop. Indeed, for \( \text{Re} = 11350 \), the Nusselt number decreases at about 63\% whereas for \( \text{Re} = 33950 \), the degradation do not exceed 35\% (figure 12). In the second zone, for \( r_i < r < r_{\text{crit}} \), the Nusselt number increase near the cylinder corresponds to the complex flow structure at the cylinder/disk junction [6,7,8,9]. Indeed, the horseshoe vortex, due to boundary layer interactions, generates a high air renewal leading to the convective exchange increase noticed on figures 9 and 10.

On the disk rear part, the horseshoe vortex legs development and the turbulent wake zone generate a Nusselt number increase (figure 9). Downstream the cylinder, the disk area submitted to this high convective heat transfer, defined by \( Nu/Nu_{\text{mean}} > 1 \), represents 46.5\% of the half-disk area for \( \text{Re} = 11350 \), and 55.9\% for \( \text{Re} = 33950 \). These thermal results correspond to the literature results [10,11,12,13] which illustrate a significant influence of \( \text{Re} \) on the size and number of the vortex horsehoe and the wake zone development downstream the cylinder. Indeed, this high turbulent zone yields significantly the convective heat transfer increase. However, the convective heat transfer intensification generated by the turbulent effects and characterized by the ratio \( Nu_{\text{mean}}/Nu_{\text{max}} \), is more important for weak \( \text{Re} \) values as shown on figure 13. For \( \text{Re} = 11350 \), the Nusselt number maximum value \( Nu_{\text{max}} \) is more twice important than the Nusselt number mean value \( Nu_{\text{mean}} \). Then, this ratio decreases with the Reynolds number \( \text{Re} \) to reach, at \( \text{Re} = 39600 \), a value of 1.53 from our experimental results and 1.42 from the trend curve.

![Fig. 13. Convective heat transfer intensification with Re on the disk rear part](image)

5. Conclusion

The experimental setup presented in this paper allowed us to evaluate the local distribution of the convective heat transfer coefficient on a disk mounted on a cylinder in air crossflow. In each test, the evolution of the thermal levels was recorded by the IR camera for each pixel on the disk while the system was cooling during 30 seconds. The thermal levels were converted to temperature values using the camera calibration law and the radiative balance of the surface scanned. A 2D inverse method based on spatial regularization allowed us to take both the radial and angular conduction and radiative fluxes into account. This method allows us to determine local variations of the Nusselt number at the disk surface. For all the tests carried out, we can identify on the front part of the disk a critical radius \( r_{\text{crit}} \) corresponding to the boundary between the boundary layer development influence zone and horseshoe vortex influence zone on convective heat transfer. The value of this critical radius depends on the Reynolds number and a correlation is given. Moreover, the Nusselt number decrease due the boundary layer thickness increase is quantify by comparing the minimum and maximum Nusselt number ratio with the Reynolds number. A correlation is also given. On the disk rear part, our thermal results correspond to the flow structure evolution described in the literature with an exchange area increase submitted to a high heat transfer in the turbulent wake zone. However, the convective heat transfer intensification generated by the turbulent effects is more important for weak airflow velocities and decreases with the Reynolds number.

REFERENCES


