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Infrared thermography study of heat transfer in an array of slot jets

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Abstract

The paper describes a study of convective heat transfer in a multiple-jet systems composed of straight and inclined slot nozzles. The application concerned is the fast cooling of moving strip. The experimental approach involves the application of infrared thermography associated with the steady-state heated foil technique. The study aims to determine the effect on the average heat transfer coefficient of the slot Reynolds number up to the value of 100000, the nozzle spacing normalised by the slot hydraulic diameter in the range $6 \leq W/S \leq 18$, the normalised nozzle protrusion length, *E/S*, from 5 to 17 and the normalised nozzle to strip standoff distance *Z/S* from 3 to 10. The geometrical arrangements tested include perpendicular (90º) and tilted (60º) nozzles.The experimental findings are compared with existing correlation; deviations, which are observed at high values of the Reynolds number may reach 25%. Jet merging phenomenon is experimentally observed a low *W/S*-values.

1. Introduction

Impinging fluid jets are widely used in industrial processes where high momentum, heat and/or mass transfer rates have to be reached [1]. Typical applications are the jet wiping, the drying of paper and textiles, the cooling of turbine blades [2], the anti-icing of aircraft [3,4], the tempering of glass sheets and the cooling of moving metal strips [4,5,6]. This last application, which is the concern of the present paper, may require cooling rate up to 50°C/s, for a steel strip of 1 mm thick.

Therefore, the paper presents an experimental study of design parameters of such a gas cooling system. The aim is to determine the local and mean convective heat transfer coefficient in an arrangement of gas slot nozzles. The optimisation of such gas fast cooling systems requires identifying and modelling the design parameters that control convective heat transfer such as the Reynolds number, the spacing, the protrusion and the tilting of nozzles as well as the standoff distance with respect to the strip.

To access to a refined description of the thermal exchange between gas jets and strip a new facility accommodating the use of infrared thermography, has been developed at the VKI.

2. Experimental set-up

A model of a fast cooling system has been designed at scale 2/3. Figure 1-a shows a schematic of the tests set-up while the photograph in figure 1-b displays a general view. The model consists of a plenum (settling chamber) and a set of slot nozzles in the front side, both made in Plexiglas® for visualisation purpose. The jets impact on a vertical uniformly heated flat plate.

A ventilator delivers a nominal flow rate of 5 Nm^3/s at a pressure of 10000 Pa. The design of piping system and the vane diffuser makes that the variation of flow rate between the slots does not exceed 1%.

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The facility allows the investigation on the cooling rate of the jet Reynolds number *Re,* based on the hydraulic slot-diameter *S,* up to 100000.

The slots arrangement can be adjusted to investigate the effect of parameters such as the nozzle spacing *W* ($6 \leq W/S \leq 18$), the protrusion length of the nozzle *E* ($5 \leq E/S \leq 17$) and the impinging angle β (β=60° and 90°corresponding to perpendicular jet). More than 25 slot nozzles have been designed to cover the whole range of each parameter. The translation of the multijets system allows to change the stand-off distance, *Z*, between the nozzle and the flat plate in the range of $3 \leq Z/S \leq 10$.

The vertical flat plate is a constant heat-flux surface of 1.7m long and 0.27m wide. It consists of three adjacent electrical-circuit boards. Each circuit is made of copper foil of 27 μm thick, coating an epoxy sheet of 1 mm thick. The copper layer of the plates is machined to produce a greek fret shaped continuous electrical resistance. The design of the heated flat plate is optimised to provide uniform heat flux, q_J. The Joule heating is monitored by a potentiometer and measured with ammeters and voltmeters. The copper face is exposed to the impinging flow while an infrared camera scans the other side, which is painted black.

The IR scanner is an AGEMA Thermovision 900 system with a HgCdTe detector sensitive in the 8-12 μm wavelength range and cooled by liquid nitrogen. The standard optical set-up is 20° vertical x 10° horizontal giving an instantaneous field of view (IFOV) of 1.5mrad. The IR-camera is located at about 0.75 m. from the fat plate yielding a typical resolution on the skin of less than 1mm. The camera is mounted on a displacement carriage to scan the flat plate along the vertical direction. The thermograms are calibrated by measuring the surface temperature at dedicated points with T-type thermocouples flush-mounted on the heated element.

3. Data Reduction

More than 17 images are necessary to reconstruct the complete thermogram of the total flat plate. An image processing software has been developed to restore all the sub thermograms. The convective heat transfer coefficient is inferred from the plate surface temperature $T_w(x,y)$ by application of the Newton relation:

$$h(x,y) = \frac{q_{cv}(x,y)}{T_w(x,y) - T_f} \qquad (1)$$

In the above expression, T_f is the temperature of the jet at the nozzle exit. The convective flux q_{cv} is obtained by subtracting the heat losses q_l to the Joule heating q_J. The heat losses include the contribution of the radiation q_{rad}, the foil conduction q_{cd} along the two dimensions , *Ox-Oy*, and the possible natural convection on the face not exposed to the flow. Making use of the two-dimensional fin theory, the following expression is applied:

$$q_\ell = -e\nabla.(k\nabla T_w) + h_{tot}(T_w - T_{amb}) \qquad (2)$$

e and *k* are the thickness and the thermal conductivity of the plate, respectively.

The total heat transfer coefficient, h_{tot}, models natural convection and thermal radiation effects together. The heat transfer coefficient by radiation can be inferred from the radiosity concept [6]. The relations (1) and (2) show that *h* can be determined readily once the foil properties and the temperature field T_{w} are known. It can be noticed that the ratio h_{tot}/h does not exceed 10% while the relative uncertainty assigned to *h* is about 10%.

4. Typical Results

Thermograms of typical configurations tested are shown in figure 2. They emphasise the effect of the jet-to-jet spacing on the final convective exchange. For large *W/S*-values

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the thermogramme is a faithful representation of the impinging flow as depicted in figure 2-a. The lowest temperatures are measured at the impingement of 90° jet. The cooling rate decreases as the jet angle decreases. The low spanwise distortion of the isotherm contours denotes the two-dimensional character of the impinging flow for this specific configuration. The axial distribution of the Nusselt number, *Nu*, based on the hydraulic slot diameter, *S*, and corresponding to the jet arrangement of figure 2-a, is plotted in figure 3-a. Peak of Nusselt number of 400 and more can be reached at the impingement of the peripheral perpendicular jets. However, the two 60°-tilted jets, designed to create additional moving strip stabilisation yield a lower cooling. For jet configuration characterised by small *W/S*-values, featured by figure 2-b, a jet merging phenomenon is observed and local distribution of the heat transfer coefficient does not reflect the nozzle arrangement as illustrated by the axial distribution of *Nu* plotted in figure 3-b. In such a nozzle array, the system may undergo an unstable jet behaviour if $W/Z < 1.5$.

For engineering purpose like the design of industrial heat exchanger, it is more convenient to calculate an average convective heat transfer coefficient, $<h>$ from the local distribution. To to transpose the final correlation to industrial designs, it is pertinent to compute $<h>$ from a peak-to-peak integration:

$$< h >= \frac{1}{(n-1).W} \int_{-(n-1).W/2}^{(n-1).W/2} h.dx \tag{3}$$

where *n* represents the number of slots.

However, flow visualisations and numerical simulations [6] prove the existence of an entrainment of air from the lateral environment surrounding the jet system into the cooling device. Such a situation has already been reported and analysed in literature [7-9]. When the ambient temperature differs from that of the jet, this lateral air entrainment will affect the cooling efficiency. This effect depends on the enthalpy of the entering air flux compared to the enthalpy of the jets. A corrective factor F_e by which the experimental average Nusselt number has to be multiplied to disregard this thermal entrainment effect has been established:

$$F_e = 1 - \frac{T_{jet} - T_{amb}}{T_{plate} - T_{jet}} f\left(\frac{A}{l^2}\right) \tag{4}$$

f is an increasing function of the ratio A/l^2 where *A* is the lateral entrance cross-section area and *l* the length of the slot.

Relation (4) points out that the thermal entrainment effect will be more important for a cooling unit with a small width *l*.

Figure 4 plots the corrected average Nusselt number $<Nu>$ versus the jet Reynolds number based on the mean velocity, *Uj*, at the exit of the nozzle. Although the Martin's correlation [1] does not account for the effect of the nozzle protrusion length, *E/S*, a comparison with the IR data is proposed in figure 4. To be coherent with the applicability condition of the Martin's correlation, only perpendicular-slot arrangements are considered.

Figure 4 shows that the increase of the nozzle protrusion is not really a benefit to the heat transfer coefficient (line with bullet symbols). However, the *E/S*-effect lessens as the normalised nozzle spacing, *W/S*, decreases (bullet and square symbols).

It is worth noting that the Martin's correlation (dashed line) tends to underestimate the IR data. The absolute deviation grows as the Reynolds number augments. Notice that above the range of the validity of Martin's relation ($Re \leq 40000$) the relative deviation $(Nu_{IR}-Nu_{Martin})/Nu_{IR}$ is around 25% for the conditions relative to figure 4. At constant Reynolds number, the discrepancy enlarges as *W/S* increases.

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5. Conclusion

Local convective heat transfer coefficient in a multiple-jet systems composed of straight and inclined slot nozzles is experimentally determined by means of infrared thermography combined to the thermo-foil technique.

The decrease of the normalised nozzle-to-nozzle spacing *W/S* may lead to the jet merging. In such an event, the footprint of the multijet configuration on the thermal mapping smears out.

Effect of lateral air entrainment on the average heat transfer coefficient has to be corrected when the slot length is small compared to the typical size of the lateral entrance area.

No evident benefit of the nozzle protrusion to the heat transfer coefficient has been experimentally observed.

The nozzle tilting introduced to reduce strip flutter is accompanied by a decrease of the local heat transfer.

The effects on the average heat transfer coefficient of the slot Reynolds number, the nozzle spacing, the nozzle protrusion length and the nozzle-to-strip distance are appreciated. In this respect, deviations between the correlation established from the infrared data and the Martin's correlation have been observed for high Reynolds numbers.

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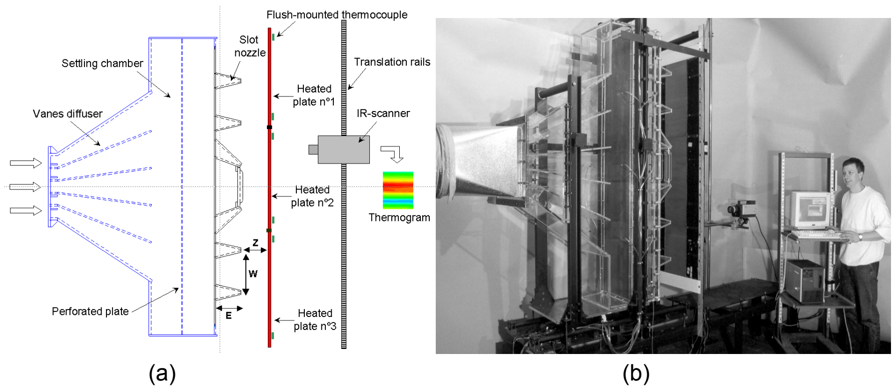

Fig. 1. *Experimental Set-up: (a) Schematic, (b) Photography*

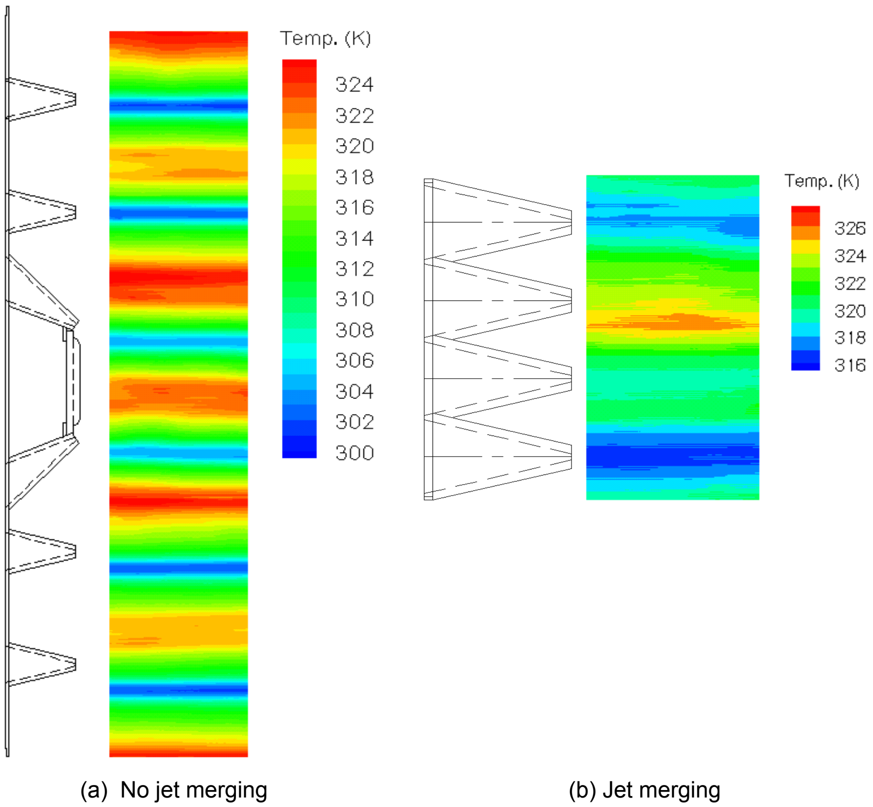

Fig. 2. *Typical Infrared Thermogrammes*

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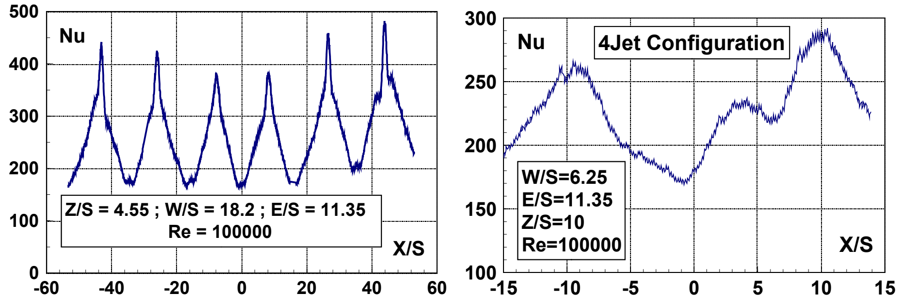

(a) No jet merging (b) Jet merging

Fig. 3. *Typical Nu-distribution*

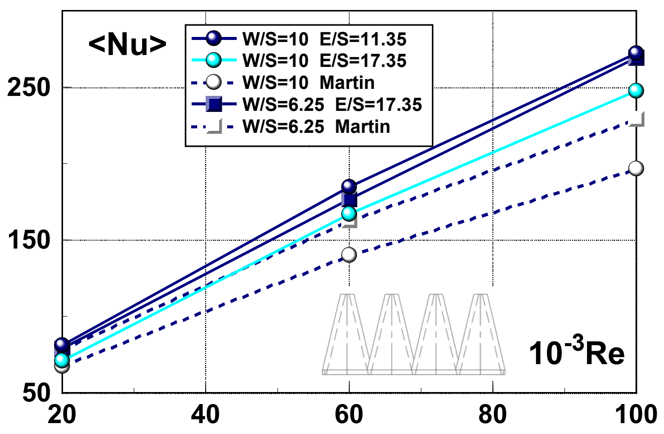

Fig. 4. *Effect of Reynolds number on the mean Nusselt Number: comparison with the Martin's correlation*