http://dx.doi.org/10.21611/qirt.1996.024

IR heat transfer measurements in a rotating channel

by T. Astarita, G. Cardone and G.M. Carlomagno

University of Naples - DETEC, P.le Tecchio, 80 - 80125 Naples, ITALY, Tel. ++39 81 7682178; Fax ++39 81 2390364

Abstract

The main aim of the present study is to develop a new experimental methodology that allows accurate measurements of the local heat transfer distribution nearby a 180deg sharp turn in a rotating square channel to be performed by means of infrared thermography. Another objective is to prove that the use of infrared thermography may be appropriate to experimentally study this type of problems. To perform heat transfer measurements, the *heated-thin-foil* technique is used and the channel is put in rotation in a vacuum tank so as to minimise the convective heat transfer losses at the surface of the foil on the channel outside. Some preliminary results in terms of temperature distributions and averaged Nusselt number *Nu* profiles are presented.

1. Introduction

To increase the thermodynamic efficiency of gas turbine engines is necessary to increase the gas entry temperature. Present advanced gas turbines operate at gas entry temperatures much higher than metal creeping temperatures and therefore require intensive cooling of their blades especially in the early stages. A classical way to cool turbine blades is by internal forced convection: generally, the cooling air from the compressor is supplied through the hub section into the blade interior and, after flowing through a serpentine passage, is discharged at the blade trailing edge. The serpentine passage is mostly made of several adjacent straight ducts, spanwise aligned, which are connected by 180deg turns. The presence of these turns causes separation of the flow with consequent high variations of the convective heat transfer coefficients. Furthermore the rotation of the turbine blade gives rise to Coriolis and much stronger buoyancy forces that may completely change the distribution of the local heat transfer coefficient. To increase the blade life, which depends also on thermal stresses, it is necessary to know the distribution of the local convective heat transfer coefficient.

In the case of radially outward flow, the Coriolis force produces a secondary flow (in the form of a symmetric pair of secondary vortices), in the plane perpendicular to the direction of the moving fluid, which pushes the particles in the center of the channel towards the trailing surface, then along the latter in the direction of the side walls and finally back to the leading surface. The presence of these two secondary cells enhances the heat transfer in the vicinity of the trailing wall and reduces it at the leading surface with respect to the non-rotating case. When the flow is reversed, i.e. radially inward flow, one has only to change the role played by the leading surface with that of the trailing one and *viceversa*. Furthermore, the heating at the walls causes a temperature difference between the core and the wall regions, so that the induced density difference and the strong centripetal acceleration due to rotation give rise to a buoyant effect. This effect magnifies the influence of the Coriolis force in the radially outward flow and reduces it in the opposite case.

The combined effects of Coriolis and buoyancy forces on the heat transfer has been investigated by many researchers; in particular,the works of Morris et al. [1,2,3], Wagner et al. [4,5], Iacovides and Launder [6], Han and Zhang [7], Mochizuki et al. [8] are acknowledged. All the previous experimental researches analyze regionally averaged heat fluxes and the acquired

http://dx.doi.org/10.21611/qirt.1996.024

data practically agree with the assertions of the previous paragraph.

The need to produce detailed and reliable local heat transfer distributions in rotating channels (including the 180deg turn) not only is important *per se* but is also relevant to validate computer programs which are often used to study these complex flows. The main objective of the present work is to develop a new experimental methodology in order to perform accurate measurements of the local heat transfer distribution nearby a 180deg sharp turn in a rotating square channel by means of infrared thermography. Another aim is to show that the use of infrared thermography in these type of problems may be advantageous on account of its relatively good spatial resolution and thermal sensitivity. Moreover, the use of thermography matches both qualitative and quantitative requirements. The essential features of this methodology are [9]: it is non-intrusive; it allows a complete two-dimensional mapping of the surface to be tested; the video signal output may be treated by digital image processing.

2. Experimental apparatus

The design of the experimental apparatus (*figure 1*) is a direct consequence of the *heated-thin-foil* technique [10] which is used to measure the convective heat transfer coefficient distribution at the channel inside. In fact, since the external surface of the foil (which is viewed by the IR camera) cannot be thermally insulated, the only way to prevent high thermal losses by forced convection at the channel outside is to have the channel itself rotating in the vacuum. Therefore, the apparatus consists of a confinement circular tank (vacuum tank) which contains a rotating arm. The tank is 850*mm* in diameter and its seals are designed so as to have the tank operating at an absolute pressure below 100*Pa*. The rotating shaft is connected, by a toothed belt, to an AC electric motor and its angular speed may be varied in a continuos way, in the range 0-2000*rpm,* by changing the pulleys and by means of an inverter. The rotating arm includes a two pass square channel, 22*mm* in side and 330*mm* in length, and is balanced by a counterweight. The shaft is hollow so as to feed and to exhaust the air passing through the channel. An AR/AR Germanium window, which is placed on a hood located on the side wall of the tank (not shown in *figure 1*), is used as optical access for the IR camera.

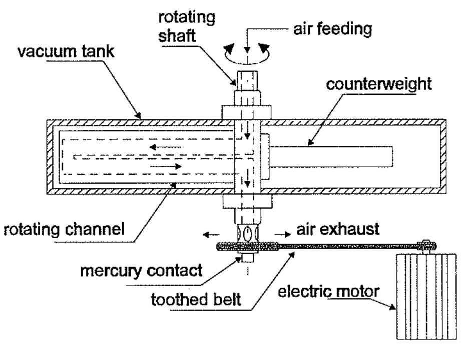

Fig. 1 . Experimental apparatus

In order to reduce the rotating weight and the wall thermal conductance, both skeleton and cover of the two pass channel (*figure* 2) are made of composite material (about 1*mm* thick): epoxy resin and kevlar mat. The thickness of the cover is chosen so as to have a deformation, smaller than 0.1*mm* under the effect of the pressure difference between the inside of the channel and the vacuum tank. Two 10μm thick stainless steel foils are glued on the channel cover in correspondence of the walls of each pass. The foils are used to generate an uniform heat flux by Joule effect and therefore are con-

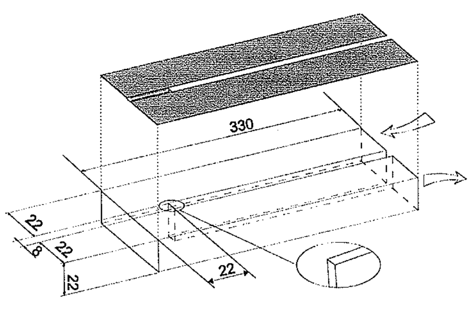

Fig. 2 . Channel skeleton and cover

http://dx.doi.org/10.21611/qirt.1996.024

nected to a DC power source via a mercury rotating contact attached to the shaft. Across the turn, i.e. in correspondence of the partition wall which is present between the two passes of the channel, the two stainless steel foils are electrically connected by a two pass copper plate which, however, generates an average Joule power dissipation per unit area slightly higher than the one dissipated by the stainless steel foils.

3. Experimental procedure

The steady state *heated-thin-foil* technique [10] is chosen to correlate the measured temperature to the convective heat transfer coefficient h. In particular, for each pixel of the digitised thermal image, h is calculated as:

$$h = (q_w - q_r) / (T_w - T_b) \qquad (1)$$

where q_w is the Joule heat flux, q_r the radiative flux to ambient and channel, T_w and T_b are the wall temperature and the local bulk temperature, respectively. Because of the low value of the pertinent Biot number, the heated wall may be considered isothermal across its thickness.

The radiative thermal losses q_r are computed from the measured T_w, and the losses due to tangential conduction are neglected because of the very low thermal conductivity of kevlar (structural material of the cover) and the low thickness of the heating foils.

As already mentioned, the main problem to perform accurate measurements of the convective heat transfer coefficient at the channel inside is necessary to reduce the thermal losses from the surface viewed by the IR camera, i.e. at the channel outside. In the case of turbulent flow the convective heat transfer coefficient h_e from a rotating arm to a fluid may be expressed by:

$$h_e\, r / \lambda = a\, (\rho\, \omega\, r^2 / \mu)^{0.8} \qquad (2)$$

where a is a dimensionless constant, r is the radius from the axis of rotation, ω is the angular speed, ρ is the fluid density, λ and μ are the thermal conductivity and the viscosity coefficients of the fluid evaluated at film temperature. Therefore, in the present case, the only way to reduce the convective thermal losses is to reduce the fluid density, i.e. the air pressure in the tank.

The local bulk temperature T_b is evaluated by measuring the stagnation temperature T_1 at the channel entrance and by making a one-dimensional energy balance along the channel, i.e. along the channel main axis; triangular heating sections are considered in the turning zone. By measuring T_1, the temperature at the channel outlet and the air mass flow rate for each test run, an overall energy balance is also performed so as to compare the energy received by the fluid with the net electric power input.

The infrared thermographic system employed is the AGEMA Thermovision 900. The field of view (which depends on the optics focal length and on the viewing distance) is scanned by the Hg-Cd-Te detector in the 8-12μm infrared window. Nominal sensitivity, expressed in terms of noise equivalent temperature difference, is 0.07$°C$ when the scanned object is at ambient temperature. The scanner spatial resolution is 235 instantaneous fields of view per line at 50% slit response function. A 10°x20° lens is used, during the tests, at a viewing distance of 1.1m which gives a field of view of about 0.12x0.24m^2.

Since the channel is rotating during the tests, it is not possible to take its whole thermal picture in one shot; it is then necessary to make use of the line scan facility of the AGEMA 900 in order to take advantage of the much higher acquisition frequency of a line (2500Hz instead of 15Hz for the full frame). In particular, when the channel wall reaches thermal steady state, the acquisition of a single line at a time starts. Each time the channel passes in front of the field of view of the camera (an optical trigger connected to the main unit monitors the passage of the

http://dx.doi.org/10.21611/qirt.1996.024

channel), the thermographic system *signes* the measured line and, after 32 acquisitions of said line, an application software picks up all the signals of the *signed* line, averages them and puts the averaged signal in a blank image. The procedure is automatically repeated by changing the measured line until the whole thermal image is reconstructed. Each image is digitised in a frame of 136 x 272 pixels at 12 bits. An application software can then perform on each thermal image: noise reduction by numerical filtering; computation of temperature and heat transfer correlations.

4. Results and discussion

In this section some preliminary results, in terms of temperature maps (or profiles) and dimensionless averaged convective heat transfer coefficient distributions are presented. The temperature maps may be considered as surface flow visualisations. However, it has to be pointed out that, by neglecting the radiative losses and the continuous increase of T_b along the channel, the temperature difference is inversely proportional to the distribution of the convective heat tranfer coefficient (see eq. 1), i.e. an higher temperature results in a lower h and viceversa.

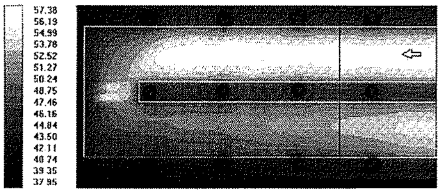

Fig. 3a . Temperature map; *Ro*=0

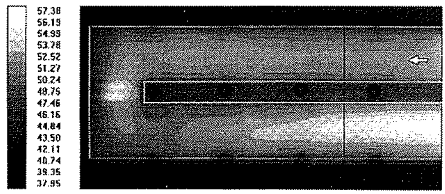

Fig. 3b Temperature map; *Ro*=0.35

In these preliminary results the effect of buoyancy will not be considered. In this case, the two dimensionless numbers that rule the physical phenomena are the Reynolds number *Re* and the rotation number *Ro* (which is the inverse of the Rossby number):

$$Re = \rho\, V D / \mu \qquad (3)$$

$$Ro = \omega\, D / V \qquad (4)$$

where D is the hydraulic diameter of the channel and V is the average inlet velocity in it. *Re* governs the *static* behavior of the flow field while *Ro* is a dimensionless measure of the rotational effects. The efficacy of thermography to perform this type of measurements is demonstrated by the two thermograms of *figure 3* which show the temperature maps of the heated trailing side for *Re*=5000 and for the two cases of static channel (*figure 3a*) and channel rotating at 552 *rpm* (*figure 3b*). The Joule heating fluxes and the temperature at the channel inlet are practically equal for the two cases. The two white spots that appear just on the left side of the channel partition wall are due to the presence of the two pass copper plate that electrically connects the two stainless steel foils and produces a discrete heating in that zone.

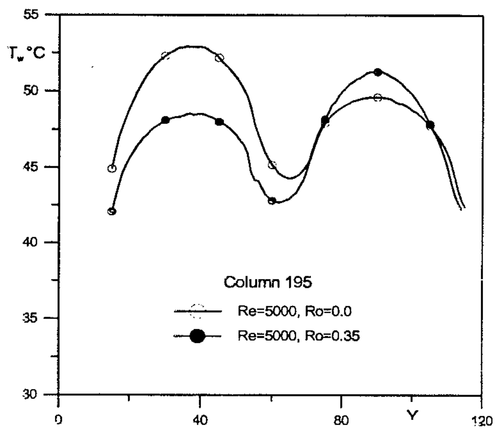

Fig. 4 . Temperature profile

http://dx.doi.org/10.21611/qirt.1996.024

In the rotating channel case, the shifting of the darker zones towards left (i.e. towards the end plate of the channel) and the more uniform temperature at the entrance of the exhaust are due to the effect of the centrifugal force on the flow that pushes the flow towards the end plate. No recirculation zone in the non-rotating channel is found in the vicinity of the first outer corner probably because of the relatively high thickness of the partition wall with respect to the channel width.

The spanwise temperature profiles along the two vertical dark lines marked in *figures 3a* and *3b* are reported in *figure 4* for the two cases of rotating (*Ro*=0.35) and non-rotating (*Ro*=0) channels. For the rotating case, the profile is relative to the trailing side of the channel. The left hand side of the profiles is relative to the situation of radially outward flow, while the right hand side to a radially inward flow.

By recalling eqn.(1) and assuming that T_b is the same in the two cases, which is true if the two radiation losses are practically equal, the two profiles seem to agree with the assertions presented in the introduction. Indeed, the lower value of temperature reached for the rotating channel (with respect to the non-rotating one) in the radially outward flow is a witness to a higher heat transfer coefficient; the reverse is true for the inward flow. It has also to be noticed that, at the considered location, in the non rotating channel, the inward flow attains a spanwise averaged heat transfer coefficient higher than that corresponding to the outward flow; Instead, in the rotating channel, the higher heat transfer coefficient is found for the outward flow.

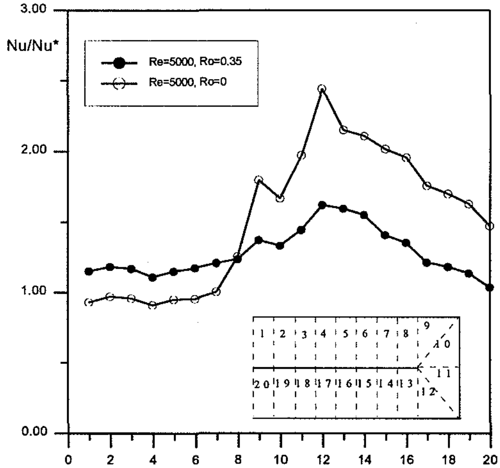

Fig. 5 . *Nu* distribution.along the channel

Figure 5 shows the segment-by-segment distribution of the average Nusselt number on each segment *Nu* normalized by means of the Nusselt number *Nu** as predicted by the Dittus-Boelter correlation. The test conditions are the same as in the previous figures but the radially outward flow corresponds, in this figure, to the first nine segments and the radially inward to the last nine ones. Each segment is 0.5*D* long.

For the non-rotating case (*Ro*=0), it can be to notice that the behaviour of a fully developed flow is practically recovered up to segment 7. Subsequently, the average *Nu* increases almost continuously downstream of segment 7 (at the entrance of the turn) up to segment 12 and, afterwards (starting from the entrance of the outlet duct), continuously decreases towards the value corresponding to a redeveloped flow. The decrease of *Nu* which is found in correspondence of segment 10 (and presumably 11) is most probably due to the presence in that zone of the two-pass copper plate connecting the two stainless steel foils.

For the rotating case (*Ro*=0.35): the *Nu* distribution is more flat; the outward flow exhibits a higher Nusselt number, while the contrary is true for the inward flow (the *Nu* towards the exit is lower than that at the entrance); the peak at the exit of the turn is much lower than that for the non-rotating case.

http://dx.doi.org/10.21611/qirt.1996.024

5. Conclusions

The aim of the present study was to develop a new experimental methodology in order to perform accurate measurements of the local heat transfer distribution nearby a 180deg sharp turn in a rotating square channel by means of infrared thermography. To perform heat transfer measurements, the *heated-thin-foil* technique was used and the channel was put in rotation in a vacuum tank so as to minimise the convective heat transfer losses at the surface of the foil on the channel outside.

The preliminary tests have shown that, at the trailing side of the rotating channel, the flow that is moving radially outward exhibits a higher heat transfer coefficient with respect to the non-rotating case, while the contrary is true for a radially inward flow. Furthermore, the value of the peak of the heat transfer coefficient, that is usually present at the exit of the turn for non-rotating channels, is much lower for the rotating one. As a result the total heat transfer distribution along the trailing side of the channel exhibits a more uniform behaviour for the case of the rotating channel with respect to the non-rotating one.

6. Acknowledgment

The authors wish to acknowledge the financial and technical support of Alfa Romeo Avio s.p.a., Pomigliano in manufacturing the test rig.

REFERENCES

[1] HARASGAMA (S. P.) and MORRIS (W. D.) - *The Influence of Rotation on the Heat Transfer Characteristics of Circular, triangular, and Square-Sectioned Coolant Passages of Gas turbine Rotor Blades, ASME Journal of Turbomachinery, Vol 110, pp 44-50, 1988.*

[2] MORRIS (W. D.) and GHAVAMI-NASR (G.) - *Heat transfer Measurements in Rectangular Channels With Orthogonal Mode Rotation, ASME Journal of Turbomachinery, Vol 113, pp 339-345, 1991.*

[3] MORRIS (W. D.) and SALEMI (R.) - *An Attempt to uncouple the Effect of Coriolis and Buoyancy Forces Experimentally on Heat Transfer in Smooth Circular Tubes That Rotate in the orthogonal Mode, ASME Journal of Turbomachinery, Vol 114, pp 858-864, 1992.*

[4] WAGNER (J. H.), JOHNSON (B. V.) and HAJEK (T. J.) - *Heat Transfer in Rotating Passages With Smooth Walls and Radial Outward Flow, ASME Journal of Turbomachinery, Vol 113, pp 42-41, 1991.*

[5] WAGNER (J. H.), JOHNSON (B. V.) and KOPPER (F. C.) - *Heat Transfer in Rotating Serpentine Passages With Smooth Walls, ASME Journal of Turbomachinery, Vol 113, pp 321-330, 1991.*

[6] IACOVIDES (H.) and LAUNDER (B. E.) - *Parametric and Numerical Study of Fully Developed Flow and Heat Transfer in Rotating rectangular Ducts, ASME Journal of Turbomachinery, Vol 113, pp 331-338, 1991.*

[7] HAN (J. C.), ZHANG ((Y. M.) and KALKUELHLER (K.) - *Uneven Wall Temperature Effect on local Heat Transfer in A Rotating Two-Pass Square Channel with Smooth Walls, ASME Journal of Heat Transfer,* Vol. 114, pp. 850-858, 1993.

[8] MOCHIZUKI (S.), MURATA (A.) and YANG (W. J.) - *Heat Transfer Characteristics in a Rotating Square Channel with Radially outward/inward Throughflow, Fifth International Symposium on Transport Phenomena and Dynamics of Rotating Machinery*, Vol. A, may 1994, p. 398-408.

[9] CARLOMAGNO (G. M.) and DE LUCA (L.) - *Heat Transfer Measurements by means of Infrared Thermography,* in Handbook of Flow Visualization, W.J. Yang ed., pp. 531-553 Hemisphere, 1989.

[10] CARLOMAGNO (G. M.) - *Heat Transfer Measurements by Means of Infrared Thermography,* in Measurement Techniques, Von Karman Institute for Fluid Mechanics Lect. Series 1993-05, 1-114, Rhode-Saint-Genese, 1993.

[11] LEZIUS (D. K.) and JOHNSTON (J. P.) - *Roll Cell Instabilities in Rotating Laminar and Turbulent Channel* Flow, *Journal of Fluid Mechanics*, Vol. 77 ,pp. 153-175, 1976.