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Thermal measurements in a single axisymmetric jet impinging normal to a flat plate

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Abstract

Measurements on convective heat transfer coefficients between a flat plate and an air jet impinging perpendicular to it are made to account for the influence of some goveming parameters in order to gain some new understandings for a data correlation. Particular attention is focused on the influence of the recirculation effects within the exhaust area and the temperature difference between the jet and the ambient. Measurements of surface temperatures (adiabatic wall and wall temperature) are made by means of an infrared scanning radiometer applied to the *heated thin foil* technique.

Nomenclature

D	exit nozzle diameter
L	nozzle length
M	Mach number
Nu	Nusselt number
R	radial distance
Re	Reynolds number
T	temperature
Z	impingement distance

Greek symbols

ε	emissivity coefficient
ν	kinematic viscosity

Subscripts

a	ambient
aw	adiabatic wall
o	stagnation
w	wall

1. Introduction

Jets of fluids have been and are still now of great interest because of their application in a wide variety of industrial processes including electronic equipment, paper and glass manufacturing, high temperature gas turbine, anti-icing systems, where jets (single or arrays) are used to cool, heat or dry a surface.

Earliest measurements of heat transfer were done by Gardon and Cobonpue [1] who considered the influence of nozzle exit diameter, nozzle to plate distance and flow rate. Later, Gardon and Akfirat [2] accounted for the role of turbulence in heat transfer measurements. Many studies have been made involving either jets of the same temperature as the ambient fluid or not and effects of thermal entrainment have been accounted for. In this context, Striegl and Diller [3, 4] developed an analytical model for the case of a single turbulent plane jet; later Goldstein *et al.* [5] carried out experimental tests and found that the convective heat transfer coefficient is independent of the temperature difference between the jet and the ambient if it is defined with the difference between the heated wall temperature and the adiabatic wall temperature.

Despite a considerable amount of data available in literature, a general correlation remains a challenge to researchers. In fact, the convective heat transfer coefficients between a surface and jets of fluid impinging on it depend upon many factors involving nozzle geometry, jet velocity, jet turbulence level, thermal conditions in the surrounding test section, as well as the experimental arrangement (relative position of the plate, vertical or horizontal with the jet impinging above or below) and particularly the exhaust area.

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It is worth noting that measurements from straight ducts [1] or sharp edge holes [6] show values higher than those relative to an ASME profile [5, 7, 8]. Another point to evidence is concemed with a recent paper by Mohanty and Tawfek [9], who reported heat transfer values surprisingly high compared to literature. In particular, their results are found to be about 30% higher than those of Gardon and Cobonpue [1] for the same conditions. Discrepancies may be associated with the jet displacement after impingement linked in part to the nozzle shape and in part to the exhaust area. Conversely, some studies are lacking in fundamental information making difficult to generalize experimental results.

The aim of the present study is to wrestle with geometrical parameters and initial flow conditions with particular attention to phenomena originating in the transition jet region under certain circumstances and affecting heat transfer measurements.

2. Experimental set-up and technique

The experimental apparatus, sketched in *figure* 1, includes a thin constantan foil (200*mm* wide, 470*mm* long and 0.050*mm* thick) heated by passing an electric current through it and cooled by an air jet directed perpendicular to it. A suitable stiffening tool is employed to assure the surface flatness. The cooling air, supplied by a compressor, goes through a pressure regulating valve, a heat exchanger that keeps the (total) temperature of the jet close to that of the ambient room air, then, after filtering, to a plenum chamber where pressure and temperature are metered and finally spread out by a replaceable nozzle. A truncated cone nozzle is employed with exit diameters of 5*mm* and 10*mm* and different lengths to have L/D=10-20; the dimensionless nozzle-to-plate distance Z/D ranges from 2 up to 30. The nozzle exit velocity is varied to keep the Mach number from very little values up to 0.6 with a corresponding Reynolds number, defined as $Re = VD/\nu$ (where V is the nozzle exit velocity), from 10000 to 100000 (depending on the nozzle diameter).

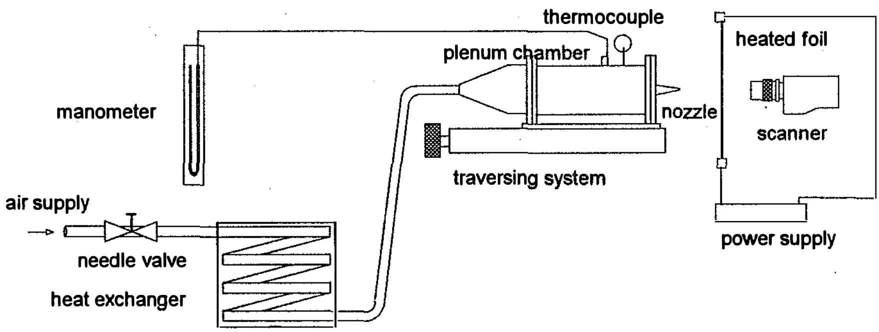

Fig. 1. Experimental apparatus

A thermographic system (IRSR), based on an Agema Thermovision 880LW scanner connected via a TIC 8000 A/D converter board to an IBM AT computer, is employed to visualize surface temperatures. The object radiation is detected by means of a mercury cadmium telluride (MCT) element covering the 8-12μm band. In the present case IRSR is applied to the *heated thin foil* technique, which consists in measuring the convective heat transfer coefficient h between a thin metallic foil, heated by Joule effect, and an air jet impinging on it. Surface temperature distribution is measured by viewing the rear face of the foil (i.e. the opposite side of jet impingement). In fact, since the Biot number $Bi=hs/k$ (h heat transfer coefficient, s and k thickness and thermal conductivity of the foil, respectively) is small respect to unity, the temperature can be considered practically uniform across the foil thickness. So h is defined as $h=Q/(Tw-Taw)$ where Q is the Joule heating per unit area,

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corrected for conduction and radiation losses (when not negligible). So that, each test run consists of two parts; firstly, electric current off, *Taw* is measured and the so-called "cold image" is recorded, secondly, electric current on, *Tw* is measured and the "hot image" is recorded. In particular, each image is averaged over 16 images, *Tw-Taw* is obtained by subtracting two thermal images, the cold from the hot one. The high sensitivity of IRSR allows keeping joule heating quite low avoiding high raising of foil temperature. As a result on one hand the electrical resistivity of the foil can be assumed constant, on the other hand the heat losses by natural convection and radiation on the back can be practically neglected. According to the desired spatial resolution, a lens of 7° or 20° is used, the spatial resolution, at minimum focus distance, is of order of one pixel per millimeter. Higher spatial resolution up to 10 pixels per millimeter can be obtained by using extension rings. In order to make the emissivity factor ε of the surface viewed by IRSR close to unity, the surface itself is coated with a thin film of opaque black paint. The emissivity factor is measured, still by the IR system, by comparing the detected radiation from a specimen heated by means of a bath/circulator with its real temperature measured by means of a mercury thermometer. In particular, for the employed paint, ε is found to be equal to 0.95.

In order to quantify the influence of the relative position of the jet respect to the plate, the plate is positioned horizontally with the impinging jet placed vertically below. In this case being far-fetched the vertical position of the scanner (the dewar for the liquid nitrogen cannot stand an inclination angle greater than 70°) the image is reflected from a mirror placed at 45°.

3. Results

The air jet, after impingement, is displaced in circumferential isotherms as shown in *figure* 2.

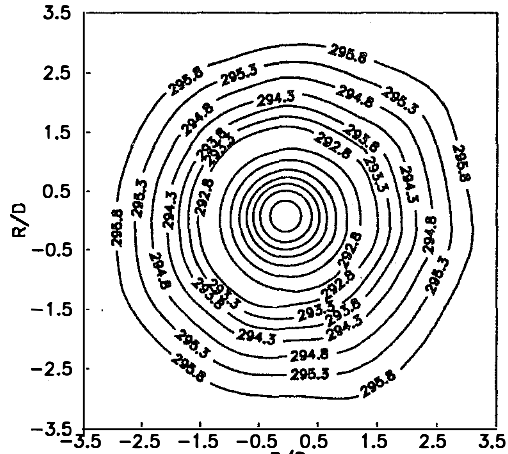

Fig.2. Map of adiabatic wall temperature D=10mm, Z/D=3, M=0.56

Experimental data, averaged over each circumference of given radius are reduced in dimensionless form in terms of the Nusselt number Nu defined as:$Nu = hD/k$ where h is the local heat transfer coefficient and k is the thermal conductivity of air evaluated at the film temperature.

In general, the radial Nusselt number distribution follows the description given by Gardon and Cobonpue [1]. In details, as shown in *figure* 3 (truncated cone nozzle D=10mm, L/D=10, Re=60000) for large Z/D (Z/D>6) the radial distribution takes the form of a characteristic bell shape, at Z/D=6 an annular "hump" begins to develop around the central peak and grows with

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decreasing *Z/D*, at *Z/D*=4 it becomes a well defined secondary peak that grows further and reaches a maximum at *Z/D*=2. Simultaneously, the central peak decreases turning itself into a central minimum. So at short impingement distances (*Z/D*<6) a minimum value occurs at the stagnation point, the inner peak at *R/D*=0.5 and the outer peak (more pronounced than the former one) at about *R/D*=2. It has to be pointed out that the above described distribution is sensitive to many factors including geometrical parameters as well as temperature and turbulence of the jet.

In particular cases related to flow displacement and flow conditions (low Reynolds numbers for instance) phenomena as recirculation and/or buoyancy can originate in the wall jet region with subsequent increasing of the jet temperature. To this last, it can be observed a time-dependent heat transfer decreasing, which, in particular, is stressed upon as the joule heating is increased. Buoyancy effects, originating at relatively high *R/D* when a jet impinges on a vertical plate at moderate velocity, are investigated by means of the Schlieren technique and by means of the same thermography by viewing the lateral side of the nozzle (for relatively short impingement distances). It is found that the reversal flow crosses the nozzle (or the horizontal flow) at a distance of 30*mm* from the plate. The former drawback is overcome by positioning the plate horizontally with the nozzle impinging from below. However, a short nozzle length associated with a large diameter of the flange supporting the nozzle can give rise to recirculation effects, which affect the lateral Nusselt distribution as shown in *figure* 4.

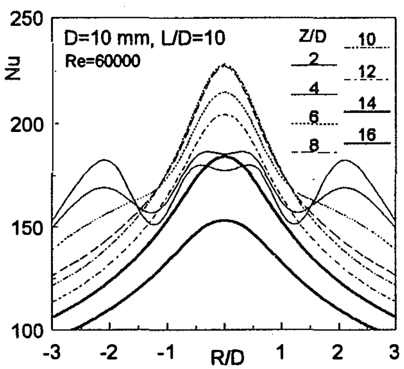

Fig.3. Radial Nusselt number distribution influence of impingement distance

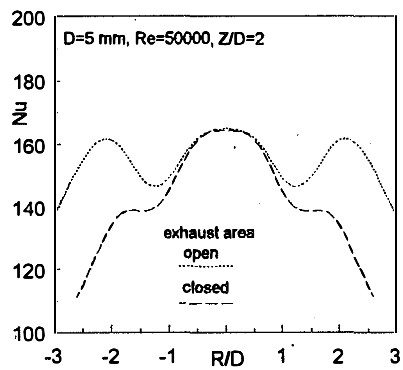

Fig.4. Radial Nusselt number distribution influence of exhaust area

At last, the behaviour of the flow field, under a difference of the above cited temperatures, is analysed; some tests are carried out with *To* about 10°C or more above the ambient temperature. The Nusselt axial distribution in the stagnation point Nu_o, as one can see from *figure* 5 that refers to a truncated-cone nozzle of *D*=10*mm* for *Re*=89000 and *To-Ta*=0°C and 10°C respectively, shows no significant influence of *To-Ta* and confirms the finding of Goldstein *et al.* [5]. It has to be pointed out that, to enhance resolution, in these measurements thermographic images visualize only a narrow zone close to the stagnation point. It has also to be said that a relatively low difference between the jet total temperature and the ambient one affects in the same way both the adiabatic wall temperature and the wall one, so no effects can be noticed in terms of heat transfer, whilst, warming the jet to a greater temperature involves variations in the impingement flow pattern.

However, it is known [10] that the potential core length tends to shorten as the jet temperature increases. As shown in *figure* 6 for D=10*mm*, and *To-Ta*=40°C the outer peaks relative to *Z/D*=4 become much milder while the value in the stagnation point grows to

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approach the configuration relative to *Z/D*=6 for *To-Ta*=0°*C*; otherwise *Z/D*=6 assumes the form of a characteristic bell shape in accordance with the configuration relative to higher *Z/D*.

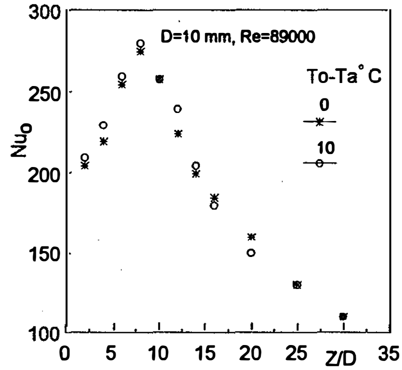

Fig.5. Stagnation point Nusselt number influence of To-Ta

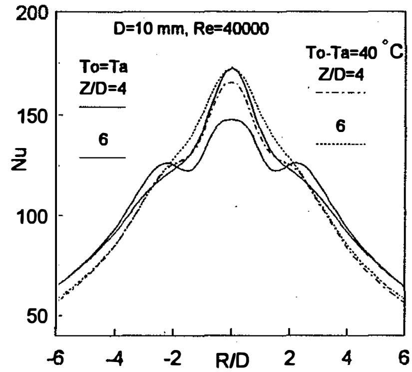

Fig.6. Radial Nusselt number distribution influence of To-Ta

4. Conclusions

Investigations concerned with convective heat transfer coefficients, between a plate and an air jet impinging on it, show, apart from the flow conditions (difference between the total jet temperature and the ambient one), a strong influence of the exhaust area. So that, from a practical standpoint, the supporting tool of the nozzle and the relative position of the nozzle to the plate should be a prime consideration. In fact, from arguments of the present work, a large supporting flange underneath the nozzle (horizontal plate with impingement from below) prevents upcasts and favours recirculation of spreaded air entailing warming of the jet air and time-dependend heat transfer lowering. This might result in devastating effects in certain industrial processes.

In addition, the difference between the jet total temperature and the ambient one has to be taken into account because it can entrain large amount of fresh air giving rise to upper movement of the stagnation point and chaotic temperature distribution over the surface under treatment.

Acknowledgments

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