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Experimental and numerical investigation of a fully confined impingement round jet

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ABSTRACT

The heat transfer characteristics of a fully confined impingement jet are experimentally and numerically evaluated. Full surface heat transfer coefficient distributions are obtained for the target and impingement plate of the model using the transient liquid crystal technique and a commercial CFD solver. The confined box consists of a single round jet impinging over a flat surface at relatively low jet-to-target plate distances, varied between 0.5 and 1.5 jet diameters. The impingement geometry is blocked from the three sides, and therefore, the air of the jet is forced to exit the model in a single direction resulting in a fully confined configuration. Experiments were carried out over a range of Reynolds varying between 16,500 and 41,800. The experimental data is compared to the numerical simulations aiming to quantify the degree of accuracy to which the heat transfer rates can be predicted.

1. Introduction

Among many convection transport processes, jet impingement is a particularly attractive cooling/heating mechanism due the very high heat transfer rates that can be achieved in the stagnation zones. Therefore, single or multi-jet impingement configurations are highly applicable at critical regions where high heat transfer coefficients are required, such as turbine blade cooling, drying technologies and thermal management of electronic equipment.

The main flow and heat transfer characteristics of baseline geometries with single or multi-jet arrangements impinging onto flat surfaces are well documented in the literature in a variety of configurations, and review studies can be found in Jambunathan et al. [1], Viskanta [2], and more recently, in Weigand and Spring [3]. The results of jet impingement heat transfer research concluded a power-law relating heat transfer rates and flow conditions ($Nu \sim Re^m$), i.e. Goldstein et al. [4], where typical values of exponent m are varied from 0.5 for laminar-stagnation point regions to 0.8 for turbulent-jet wake regions [5]. Additionally, a considerable effect of jet-to-target plate distance (Z/D) on the level and distribution of convection coefficients is also observed, especially for free impinging jets, i.e. Baughn and Shimizu [6]. Stagnation point heat transfer is maximised when the target surface is placed at the end of jet potential core ($Z/D = 5 - 7$) and secondary peaks of heat transfer appear at a certain radial distance ($r/D \sim 2$), when $Z/D \leqslant 2$. The local heat transfer distribution is a direct consequence of the resulting flow field, which is characterised by a spectrum of vortical structures and strong recirculation zones, as shown by the smoke flow visualisation experiments of Popiel [7].

However, in many engineering applications the jet is required to be confined by the presence of a wall at the level of jet exit which results in a significant effect on the flow structure, and hence on the distribution of convection coefficients, due to the interaction between the jet and the top wall [8]. There are a number of experimental studies focused on confinement effects for the target plate heat transfer, which mainly concluded a heat transfer degradation depending on the jet-to-target plate distance. San et al. [9,10] investigated the local Nusselt number distribution of a single confined jet between $Z/D = 1$ and 6 and reported similarities with unconfined configurations regarding the dependancy with Reynolds number. However, Gao and Ewing [11] observed that the presence of a confining plate has a great impact at relatively small jet-to-target plate distances ($Z/D \leqslant 1$), where the local heat transfer rate is reduced by about 50% in regions where $r/D \geqslant 1.5$. Choo and Kim [12] showed that the thermal performance of the confined jet is about 20–30% lower than that of the unconfined one under fixed flow conditions. Some improvements on the thermal performance, by reducing the pressure drop of confined configurations, were reported by Brigoni and Garimella [13] using chamfered nozzle designs, while Fenot et al. [14] showed that confining plates prevent the entrainment of surrounding fluid maintaining relatively high effectiveness values ($\eta \geqslant 0.8$) even if $Z/D = 5$. The main conclusions of the above experimental studies

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Nomenclature

y_1^+	non-dimensional wall distance	θ	non–dimensional temperature
x,y,z	coordinate system	μ	dynamic viscosity (kg/ms)
X	Box length (m)	c	specific heat (J/kgK)
Y	Box width (m)	t	time (s)
Z	Separation distance (m)	ΔP	pressure drop (Pa)
D	jet diameter (m)	U	velocity (m/s)
T	temperature (K)	LC	liquid crystals
h	heat transfer coefficient (W/m^{2}K)	o	initial conditions
Re	Reynolds number	g	hot gas conditions
L	length of impingement hole (m)	D	jet diameter
Nu	Nusselt number	w	wall
Pr	Prandtl number	C	Flow coefficient
ρ	density (kg/m^3)		

were also confirmed by the detailed numerical work of Behnia et al. [15].

In all the aforementioned investigations, confinement effects were considered by the addition of a solid boundary at the level of nozzle exit resulting actually in a partially confined configuration where the spent air of the jet was spread out in the radial direction, while no heat transfer information were provided for the top wall (jet–plate) of the geometry which experience considerable heat transfer at low separation distances. Therefore, a better understanding of the local heat transfer distributions for the bottom and top wall are essential in order to ensure a reliable design and optimisation of a fully confined impingement configuration.

The objective of this study is to investigate the flow and heat transfer characteristics of a fully confined impingement jet at relatively small separation distances, varying between $Z/D = 0.5$ and 1.5. The confined geometry consists of a single round straight nozzle and a single exit for the spent air of the jet resulting in an overall asymmetry of the flow domain. Full surface heat transfer coefficient distributions are obtained for the target (bottom wall) and the jet–plate (top wall) using experimental (transient liquid crystal technique) and numerical tools (CFD). The CFD simulations are compared to experiments aiming to quantify the degree of accuracy to which the local heat transfer rates can be predicted. The results are analysed using various post-processing procedures aiming to determine fully confinement effects on the local heat transfer rate.

2. Experimental arrangement

2.1. Test facility

Fig. 1 shows a schematic representation of the test facility. The open circuit low speed wind tunnel is operated in blowing mode and consists of a two–stage axial flow fan, a bellmouth transition piece with honeycombs, a straight section of square ducting that contains an air heater and the test model at the exit of the tunnel.

In a typical transient heat transfer experiment, the flow temperature is subjected to a sudden temperature change and the optical response of a liquid crystal surface coating is optically monitored. The temperature change in the main flow was obtained with a heater mesh, which was able to increase the temperature of the air within 0.1 s using Joule heating. Power to the metallic mesh was supplied by a 5-kW (40 V, 128 A) DC-Power supply. The flow conditions were determined considering the jet–plate as an orifice plate and acquiring the overall pressure drop using various static pressure tappings around the whole flow path. The Reynolds number, based on jet diameter (D), was calculated as follows:

$$Re_D = C\frac{D\sqrt{(2\rho\Delta P)}}{\mu} \tag{1}$$

where C is the jet flow coefficient determined by independent velocity measurements. Typical C values were about 0.75. Note that air properties were calculated at hot conditions (heater mesh in operation). The uncertainty in the determination of Reynolds number was always below 4%.

2.2. Test model

A schematic representation of the confined configuration is illustrated in Fig. 2. The impingement hole (D) is drilled on the jet–plate resulting in a straight round nozzle of equal length–to–diameter ratio ($L/D = 1$). A square target plate is chosen so that $X/D = Y/D = 5$. Three different separation distances (Z/D) were investigated: $Z/D = 0.5$, 1 and 1.5. The walls of the confined box are manufactured of transparent acrylic material so that the evolution of the liquid crystals is recorded on the backside of the target plate with a CCD camera (AVT Pike–F210C) at 20 Hz. Note that two different experiments were carried out at the same flow conditions in order to capture the liquid crystal evolution for both surfaces. For the impingement plate experimentation the painted target plate was replaced with a transparent one. Several Reynolds numbers were investigated in the range of Re_D=16,500–41,800.

3. Measurement technique

3.1. Transient liquid crystal technique

In the majority of previous studies, a steady-state heat transfer approach using a uniformly heated target surface with an iso-flux boundary condition was applied for the evaluation of the local heat transfer rates. Convection coefficients were then calculated by measuring the wall temperature distribution using a number of embedded thermocouples [16,17], liquid crystal surface coatings [18,19,20] or infrared thermal imaging techniques [21,22]. In the present study, the transient liquid crystal technique, which finds great applicability in multi-array heat transfer experiments, i.e. Xing et al. [23], Esposito et al. [24], Wang et al. [25], is applied in a single jet configuration. The main advantages are the high spatial resolution on the examined surfaces and the minimum disturbance of the flow. Detailed reviews of the current method and its applications can be found in the studies of Baughn [26] and Ireland et al. [27].

In a typical transient liquid crystal experiment, the flow initial temperature (T_o) is subjected to a sudden temperature change and the evolution of a liquid crystal isotherm (T_{LC}) is optically