International Journal of Heat and Mass Transfer 126 (2018) 969–979

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Convective heat transfer in circular and chevron impinging synthetic jets

Cuono Massimo Crispo *, Carlo Salvatore Greco, Gennaro Cardone

Dipartimento di Ingegneria Industriale – Sezione Aerospaziale, Università degli Studi di Napoli, "Federico II", 80125, Piazzale Tecchio 80, Napoli, Italy

A R T I C L E I N F O

Article history:
Received 2 February 2018
Received in revised form 10 June 2018
Accepted 10 June 2018

Keywords:
Chevron nozzle
Synthetic jets
Heat transfer enhancement
Impingement heat transfer
Infrared thermography

A B S T R A C T

An experimental investigation on the heat transfer enhancement achieved by impinging synthetic jets with vortex generators, in the form of chevron elements at the nozzle exit, is carried out. The heated thin foil heat transfer sensor is used in conjunction with the infrared thermography to measure the spatial distribution of the Nusselt number on the target plate. The heat transfer rates of impinging circular and chevron synthetic jets are compared under the same condition of cavity pressure. A parametric study on the effect of the dimensionless stroke length and the nozzle-to-plate distance on the heat transfer rates is carried out. For increasing dimensionless stroke length, at short nozzle-to-plate distances, the chevron synthetic jet reveals a star-shaped heat transfer pattern similar to that observed for an impinging continuous one. The results show that a chevron exit geometry can provide a significant heat transfer enhancement for relatively small nozzle-to-plate distances, up to a 20% increase with respect to the circular synthetic jet. At small dimensionless stroke lengths, such an enhancement is observed for a wide range of nozzle-to-plate distances.

1. Introduction

Thermal management is critical to many fields of application, such as aerospace, automotive, electronics and medicine. Nowadays, the improvement of the thermal performance of existing solutions and the development of innovative cooling systems are of paramount importance from both an economical and a technical viewpoint. Conventionally, impinging jets can offer very high heat transfer rates since they exhibit amongst the highest known levels of transfer capabilities for single phase flows, especially at low nozzle-to-plate distances. Thus, their properties are exploited in a wide range of processes. The heat transfer enhancement, in addition to more practical requirements such as high reliability, easy design and implementation and low cost and low space applications, have pushed a lot of researchers to focus on synthetic jet devices.

Basically, synthetic jet devices, also called zero-net-mass-flux devices, are made by a cavity bounded by an oscillating membrane and by an opening. In the most common applications, the oscillating membrane consists of a loudspeaker, a piezoelectric diaphragm or a piston; while the opening consists of an orifice, a slot or a nozzle. The jet is generated by the oscillating motion of the membrane, which causes periodic variation of the cavity volume and pressure leading to the periodic ejection and suction of fluid through the opening. In such a way, a net momentum is transferred to the working fluid outside of the cavity, without the need of active mass injection and piping systems. During the ejection stroke, the flow rolls up at the exit edges and separates, and if the momentum transferred to the fluid is large enough to overcome the suction forces associated with the ingestion stroke, a vortex ring forms and convects away. During the suction stroke, the fluid is entrained into the cavity from all directions. This time-periodic cycle leads to the formation of a train of vortex rings. Each vortex convects away, under its own self-induced velocity, generating the jet. The vortex ring may be followed by a column of fluid called trailing jet. The existence and extension of the trailing jet depend on the Strouhal number [1,2]. The Strouhal number and the Reynolds number are the dimensionless parameters governing the synthetic jet flow field. They are defined as follows:

$$St = D/L_0 = fD/U_0 \tag{1}$$

$$Re = \rho U_0 D/\mu \tag{2}$$

where U_0 is the characteristic velocity of the jet, D is the characteristic length, f is the oscillation frequency of the membrane, ρ is the fluid density, μ is the fluid dynamic viscosity and L_0 is the stroke length (defined as U_0/f). According to Smith and Glezer [3], the reference velocity U_0 is defined as

$$U_0 = \frac{1}{\tau}\int_0^{\frac{\tau}{2}} u_e(t)dt \tag{3}$$

* Corresponding author.
E-mail address: cuonomassimo.crispo@unina.it (C.M. Crispo).

https://doi.org/10.1016/j.ijheatmasstransfer.2018.06.062
0017-9310/

Nomenclature

Roman letters

A	surface area of the foil, m^2
Bi	Biot number
D	nozzle diameter, m
f	oscillation frequency of the membrane, Hz
Fo	modified Fourier number
H	nozzle-to-plate distance, m
h	convective heat transfer coefficient, $W\,m^{-2}\,K^{-1}$
I	electrical current across the foil, A
IR	infrared
k	fluid thermal conductivity, $W\,m^{-1}\,K^{-1}$
k_f	foil thermal conductivity, $W\,m^{-1}\,K^{-1}$
L_0	stroke length, m
Nu	Nusselt number
Nu_0	Nusselt number at the stagnation point
Nu_r	azimuthally averaged Nusselt number
Nu_{circ}	Nusselt number related to the circular case
Nu_{chev}	Nusselt number related to the chevron case
NETD	noise equivalent temperature difference
PIV	particle image velocimetry
P_{cavity}	cavity pressure, Pa
$\dot{q}_j$	Joule heat flux, $W\,m^{-2}$
$\dot{q}_r$	radiation heat flux, $W\,m^{-2}$
$\dot{q}_{nc}$	natural convection heat flux, $W\,m^{-2}$
$\dot{q}_{tc}$	tangential conduction heat flux, $W\,m^{-2}$
r	radial distance from the stagnation point, m
Re	Reynolds number
s	foil thickness, m
St	Strouhal number
T_{amb}	ambient temperature, K
T_{aw}	adiabatic wall temperature, K
T_{cavity}	cavity temperature, K
T_{film}	film temperature, K
T_w	wall temperature, K
U_0	reference velocity, $m\,s^{-1}$
u_e	velocity along the jet centreline at the exit section, $m\,s^{-1}$
V	voltage applied to the foil, V
x-, y-, z-	spatial coordinates, m

Greek letters

α	foil thermal diffusivity, $m^2 s^{-1}$
γ_0	enhancement ratio at the stagnation point
ε	coating paint emissivity
μ	fluid dynamic viscosity, Pa s
ρ	fluid density, $kg\,m^{-3}$
ρ_f	foil density, $kg\,m^{-3}$
σ	Stefan-Boltzmann's constant, $W\,m^{-2}\,K^{-4}$
τ	actuation period, s
φ	phase angle, °

Other symbols

∇^2	two-dimensional Laplacian operator, m^{-2}

where τ is the actuation period and u_e is the velocity along the jet centreline at the exit section.

In the recent years, the flow topology of synthetic jets has been widely investigated [4]. Smith and Glezer [3] and Smith and Swift [5] compared the flow fields of synthetic and continuous jets. They have showed that synthetic jets exhibit high rates of growth in both the jet width and entrainment when compared to continuous jets. Such an effect is more evident in the near field, where the flow field is dominated by the vortex ring that entrains more fluid than continuous jets. It is widely recognised that the enhancement of jet entrainment is one of the relevant parameters to increase the heat transfer performance of impinging jets [6]. Hence, the impinging synthetic jet could be a possible solution to achieve high heat transfer rates. For heat transfer purposes, Mahalingam and Glezer [7] used a synthetic-air-jet-based heat sink for high-power dissipation electronics (forced convection with the synthetic jets enabled a power dissipation of 59.2 W at a case temperature of 70 °C) showing that the synthetic jet heat sink dissipates approximately 40% more heat compared to a steady flow from a ducted fan blowing air through the heat sink. The heat transfer performance of an impinging synthetic jet at a Reynolds number ranging between 1500–4200 and for nozzle-to-plate distances varying between 0–25 exit diameters was investigated by Chaudhari et al. [8]. They carried out a comparison with a continuous jet observing comparable performance between the two jets at Reynolds number equal to 4000 and envisaging a better performance of the synthetic jet at higher Reynolds numbers. A recent work of Tan et al. [9] shows the heat transfer characteristics of a synthetic jet driven by a piston actuator. Attention is paid to the comparison between the local and laterally-averaged behaviours of synthetic and continuous jets. It is shown that continuous jet exhibits stronger local heat transfer than the synthetic jet near the stagnation point. However, the synthetic jet is characterised by a more flat and uniform distribution of the local heat transfer coefficient over the surface, offering better performance above all at large jet-to-surface spacings. These investigations lead to the conclusion that these devices have the potential to achieve a significant heat transfer improvement.

Besides the entrainment, another parameter of paramount importance for the heat transfer enhancement is the flow turbulent mixing [6]. Passive strategies are implemented to enhance the turbulent mixing of continuous jets. The literature reveals that several solutions have been proposed over the last decades to increase the turbulent mixing of jet and, consequently, to improve their heat transfer rate or uniformity. Some examples of application are swirlers [10], mesh screens within the nozzle [11] and streamwise vortices generators [12]. Among the passive strategies, the exit shape modification is a fast and efficient solution. Several studies deal with the effect of the exit geometry on the heat transfer and flow field of impinging steady jets [6,13–15]. Nevertheless, few studies on the effect of exit geometry of impinging synthetic jets have been undertaken.

To the authors' knowledge, the effect of the orifice shape on synthetic jet based impingement cooling was experimentally investigated for the first time by Chaudhari et al. [16]. They considered different configurations consisting in square, circular and rectangular orifice shapes. Experiments were undertaken at a Reynolds number ranging between 950 and 4000 while the dimensionless nozzle-to-plate distance varied between 1 and 25. A square orifice is found to perform better at axial distances larger than 5 diameters, while a rectangular orifice (with small aspect ratio 1–5.25) is found to be more effective at lower axial distances. Subsequently, Bhapkar et al. [17] carried out an acoustic and heat transfer study on the effect of orifice dimensions and operating parameters on the noise level and heat transfer capabilities of syn-