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Time and phase average heat transfer in single and twin circular synthetic impinging air jets

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ABSTRACT

This work presents an experimental investigation of impingement heat transfer in single circular synthetic jets and twin circular synthetic jets in phase opposition. All experiments have been performed at Reynolds number equal to 5100 and Strouhal number equal to 0.024 varying the jet axes distance and nozzle to plate distance. An IR camera is used as temperature transducer for both time average and phase average heat transfer measurements. Time average heat transfer maps show that single synthetic impinging jets have a behavior similar to that of continuous jets: at low nozzle to plate distance (up to 4 diameters) the heat transfer distribution shows an inner and an outer ring shaped region of maximum while for higher nozzle to plate distance such a feature disappears.

While obviously the twin configurations produce an heat transfer enhancement due to the fact that two jets instead of one are impinging, the interaction is found in general to have a beneficial effect. The physical behavior is in common between single synthetic jets and twin configurations at jet axes distance equal to 3 and 5 diameters. The twin circular synthetic air jets, with jet axes distance equal to 1.1 diameter, shows a different behavior with respect to single synthetic jet for H/D equal to 2 but for values of H/D higher than 4 it starts acting like a single synthetic jet differently from the other twin configurations which behave as two separated synthetic jets. Phase averaged measurements allow for an accurate description of the heat transfer mechanism: at low nozzle to plate distances (2 and 4 diameters) the heat transfer is dominated by the unsteady impinging flow produced by the ring vortex that sweeps the wall and causes the formation of the inner and outer ring shaped regions. At higher nozzle to plate distance the heat transfer is due also to a steady and less coherent turbulent flow since the impingement occurs after the potential core and the saddle point.

1. Introduction

The high heat transfer rate obtainable with impinging jets is widely recognized and explained in scientific literature [1,2] and the use of jets is very popular in many industrial applications such as paper drying, glass tempering and turbine blades cooling. A huge quantity of data is available for single, rows and multiple jets with also correlations for heat and mass transfer [3,4]. The average and instantaneous flow topology of circular impinging jets is well known as well as its effect on heat transfer [5,6].

Recent literature is focusing on the design and optimization of advanced impinging jets devices in order to apply them in particular fields such as electronic cooling. In particular several recent literature works (see for instance [7–9]) focus on the study of synthetic impinging jets. Synthetic jets are jets with zero-net-mass flux synthesized directly from the fluid in the system in which the jet device is embedded [10]. Such a feature obviates the need for an external input piping, making them ideal for low cost and low space applications. A synthetic jet is generated by a membrane oscillation in a cavity which produces a periodic volume change and thus pressure variation. As the membrane oscillates, fluid is periodically entrained into and expelled out from the orifice. During the injection portion of the cycle the flow field could be considered as one inducted by a sink, which coincides with the orifice, while during the expulsion portion of the cycle, a vortex ring can form near the orifice and, under certain operating conditions [11], convects away to form a time averaged jet [10]. In synthetic jets literature the stroke length L_0 is the integral of the average velocity at the nozzle exit during the ejection part of the cycle:

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Nomenclature

Bi	Biot number
c_p	stainless steel specific heat (J/(kg K))
D	nozzle diameter (m)
dT_w/dt	time derivative of wall temperature (K/s)
f	phenomenon frequency (Hz)
f_1	actuation frequency (Hz)
Fo_{f1}	modified Fourier number
H	nozzle to plate distance (m)
H/D	dimensionless nozzle to plate distance
h	convective heat transfer coefficient (W/(m^2 K))
$\overline{h}$	time average convective heat transfer coefficient (W/(m^2 K))
k	air thermal conductivity (W/(m K))
k_{loss}	head loss
l	jet-to-jet spacing (m)
L	nozzle length (m)
L_0	stroke length (m)
$\overline{Nu}$	time average Nusselt number
Nu_φ	phase average Nusselt number
Nu'	standard deviation of Nusselt number phase average
p_c	cavity pressure (Pa)
p_{amb}	ambient pressure (Pa)
q''_j	Joule heat flux (W/m^2)
q''_r	radiation heat flux (W/m^2)
$\overline{q''_r}$	time average radiation heat flux (W/m^2)
q''_n	natural convection heat flux (W/m^2)
q''_k	tangential conduction heat flux (W/m^2)
$\overline{q''_k}$	time average tangential conduction heat flux (W/m^2)
Re	Reynolds number
s	foil thickness (m)
Sr	Strouhal number
T_a	ambient temperature (K)
T_w	wall temperature (K)
$\overline{T_w}$	time average wall temperature (K)
T_{aw}	adiabatic wall temperature (K)
$\overline{T_{aw}}$	time average adiabatic wall temperature (K)
U	axial velocity (m/s)
U_a	exit velocity on the jet axis (m/s)
U_0	reference velocity
V	sub-cavity volume (m^3)
x	abscissa in the foil plane (m)
y	ordinate in the foil plane (m)

Greek symbols

α	thermal diffusivity (m^2/s)
ε	total hemispherical emissivity coefficient
φ	phase (°)
λ_f	foil thermal conductivity (W/(m K))
μ	air dynamic viscosity (kg/(m s))
ρ	air density (kg/m^3)
ρ_{foil}	stainless steel density (kg/m^3)
$\sum$	dimensionless jet-to-jet spacing
σ	Stefan Boltzmann's constant (W/(m^2 K^4))
τ	actuation period (s)

Acronym

NETD	Noise equivalent temperature difference
SSJ	Single synthetic jet
TSJ	Twin synthetic jet

$$L_0 = \int_0^{\tau/2} U_a(t)dt \tag{1}$$

and the reference velocity is defined as:

$$U_0 = L_0/\tau \tag{2}$$

where τ is the actuation period and U_a is the exit velocity on the jet axis.

Following Smith and Glezer [10] synthetic jets are characterized by Reynolds number and dimensionless stoke length, which essentially is the inverse of the Strouhal number [11]:

$$Re = \rho \cdot U_0 \cdot D/\mu \tag{3}$$

$$\frac{1}{Sr} = \frac{L_0}{D} \tag{4}$$

where ρ is air density, μ is air dynamic viscosity and D is the nozzle diameter.

The first literature work on synthetic jets used as cooling devices was published in 1982 by Gutmark et al. [12] that presented heat transfer data on synthetic jet used to enhance both natural and forced convection. The results revealed that the acoustically excited airflow can increase the overall heat-transfer coefficient by a factor of four. Later [13] studied the design and thermal performance of a heat sink for high power dissipation in electronics enhanced with synthetic jet impingement. The results revealed a case temperature decrease from 71.5 to 36 °C with synthetic jets operation and a power dissipation of 20–40% higher with respect to the same heat sink with a fan in the flow rate range of 3–5 cubic feet per minute.

Chaudhari et al. [7] carried out experiments on the cooling of a flat plate by using a synthetic jet generated through a circular orifice. Such experiments for Reynolds number in the range 1500–4200 and nozzle to plate distance in the range 0–25 D show that the Nusselt number is comparable with that of continuous axisymmetric jets at low Reynolds number (up to 4000), expecting it to be higher at greater values of Reynolds number.

Valiorgue et al. [8] identified two different flow regimes through defining a critical stoke length versus nozzle to plate distance L_0/H equal to 2.5. The heat transfer rate (that obviously increases with Reynolds number increasing) is found to be linearly proportional with L_0/H up to $L_0/H = 2.5$ than constant for increasing L_0/H values.

As for steady jets, heat transfer correlations have been developed also for synthetic jets [14,15]. Arik and Icoz [14] established a closed form empirical correlation to predict the heat transfer coefficient as a function of Reynolds number, axial distance, orifice size and jet driving frequency. They observed that the heat transfer coefficient on a vertical surface increases with the driving voltage; it has a peak at the resonance frequency and the effect of the axial distance on the heat transfer becomes stronger as the jet driving frequency increases. The empirical correlation proposed by Arik and Icoz [14] is valid for $Re < 2900$, $5 < H/D < 20$ and actuation frequency between 0.16 times the resonance frequency and the resonance frequency. Persoons et al. [15] compared the stagnation point heat transfer performance of an axisymmetric synthetic jet versus established steady jet correlations. Such a research led to a general correlation for the stagnation point Nusselt number including the effect of all appropriate scaling parameters: Reynolds number ($500 < Re < 1500$), jet to surface spacing ($2 < H/D < 16$) and stroke length ($2 < L_0/D < 40$). Based on such correlation, Persoons et al. [15] defined four heat transfer regimes, each one identified by a different range of values acquired by the ratio L_0/H. The fourth regime, which is attained for a value of L_0/H greater than 2.5, shows