International Journal of Heat and Mass Transfer 88 (2015) 470–480

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

The effect of jet location and duty cycle on the fluid mechanics of an unconfined free jet and its heat transfer on an impinging plate

W.R. Marcum [a,*], S.R. Cadell [a], C. Ward [b]

[a] Oregon State University, Department of Nuclear Engineering and Radiation Health Physics, 116 Radiation Center, Oregon State University, Corvallis, OR 97330, United States
[b] Oregon State University, Department of Mechanical, Industrial, and Manufacturing Engineering, 204 Rogers Hall, Oregon State University, Corvallis, OR 97331, United States

ARTICLE INFO

Article history:
Received 24 February 2015
Received in revised form 13 April 2015
Accepted 13 April 2015
Available online 15 May 2015

Keywords:
Air jet
Impingement
Pulse
Duty cycle

ABSTRACT

Unsteady heat transfer caused by an unconfined impinging air jet is experimentally studied. Four test cases are considered, two pulse frequencies and two duty cycles. The outcome of this experimental study is compared to that of steady jet impinging heat transfer relations and data. Characterization of the pulsed air jet is conducted using a hot wire anemometer at twelve discrete locations for each pulse case considered. The local instantaneous velocities collected were then used to calculate time averaged mean velocity, and turbulent intensity. A power spectral distribution was produced for each of the jet flows to acquire insight toward the most effective jet flow characteristics for heat removal on an impingement surface. An infrared (IR) camera was used to determine a full field instantaneous heat transfer coefficient for each of the pulse cases and a nozzle to plate setback distances. All data was collected over a Reynolds number range of 866 and 3776. The jet characterization suggested that a setback distance of three hydraulic diameters (as opposed to one-half or six) produces the most effective power spectral density given a pulse frequency of 5 Hz and a duty cycle of 50%. The infrared camera data collected argues against the conclusions drawn from the hot wire data suggesting that the most effective setback distance is six hydraulic diameters and with a frequency of 10 Hz and a duty cycle of 50%. All data provides similar evidence that the duty cycle over a pulse period has a larger impact on the heat transfer of an impinging air jet than that of pulse frequency.

Published by Elsevier Ltd.

1. Introduction

The study of hydrodynamics and heat transfer resulting from impinging gas jets has become a canonical area of study. Developments in the further understanding of this phenomenon have recently been made for a wide variety of industrial applications including electronics, paper drying, and turbine blades. Thermal characterization of steady jets has been studied extensively over several decades and correlations have been developed to describe heat transfer characteristics at a surface with steady jet flow parameters [1–6]. Prior studies have shown, improvements can be made to the heat transfer characteristics of an impinging jet by providing a temporal rather than steady velocity profile to the heated surface under consideration [7–12]. These improvements allow industrial operations to reduce the volume of cooling fluid which must be supplied to a process [13–16].

In 2005 Poh et al. [17] predicted through computational methods that the enhancement of heat transfer associated with pulsating flows in the turbulent regime is also present in the laminar regime. Recently, Herwig and Middelberg [18] reported a dependence of the heat transfer performance of a pulsating jet on the turbulence structure of the jet flow. Azevedo et al. [19] determined that heat transfer degradation occurs at the stagnation point of an impinging jet with a temporal velocity profile. Research conducted by O'Donovan and Murray [20,21] concluded that the magnitude of fluctuating velocity (u'), directionally normal to the impingement surface, is a single attribute of the flow field which most significantly influences the heat transfer enhancement from said surface. Additionally, Knowles and Myszko [22] experimentally measured the instantaneous velocity components of an air jet with varying distances from a plate. They found that as the plate-to-jet nozzle distance decreased, the magnitude of fluctuating velocity increased. The combined work of O'Donovan et al. and Knowles and Myszko clearly show that the relative position of a jet relative to a plate influences the heat transfer off of the plate within the turbulent flow regime. Bhattacharya and Ahmed [23] produced heat transfer enhancement measurements by placing a body in the jet stream causing a flow periodicity in the wake. Baydar found a link between local heat transfer coefficient peaks

* Corresponding author. Tel.: +1 (541) 737 3018; fax: +1 (541) 737 0480.
E-mail address: marcumw@engr.orst.edu (W.R. Marcum).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2015.04.041
0017-9310/Published by Elsevier Ltd.

Nomenclature

d	jet diameter [m]	$\bar{U}$	time average local velocity [m/s]
f	pulsation frequency [Hz]	St	Strouhal number
h	time averaged heat transfer coefficient [W/m^2-K]	λ	Taylor micro-scale [ms]
k	thermal conductivity [W/m-K]	ν	fluid kinematic viscosity [m^2/s]
r, d	Local position normal to jet flow direction [m]	ω	discrete frequency
t	current time [ms]	ψ	turbulent intensity
t'	alternate time [ms]	ρ	auto-correlation value
u	local instantaneous mean velocity [m/s]	σ	uncertainty
u'	local rms velocity fluctuation [m/s]	τ	time difference [ms]
z	nozzle to foil spacing [m]	θ	normalized local heat transfer coefficient
Nu	time average Nusselt number	Γ	normalized enhancement factor
Re	time averaged Reynolds number	Λ	enhancement ratio
S	power spectral density	$\top$	integral turbulent timescale [ms]

and low pressure regions of the flow [24]. The enhancement of heat transfer on non-flat surfaces has been shown to be significant for steady and unsteady flows relative to that of flat plates [25–33]. Persoon et al. provides a detailed overview of the conditions for which previous studies have considered pulsating impinging jets [11]. Most recently, Alimohammadi et al. performed a similar scope of work to that presented herein. The study centered on the augmentation (enhancement) of heat transfer due to a pulsing, unconfined air jet as compared to a steady air jet [34].

These previous pulsating jet studies have focused on the heat transfer dependence on frequency, temporal flow profile, and geometric effects of which only two have utilized circular nozzles and did not address the combined influence of duty cycle and pulsation frequency. Few researchers, however, have investigated the effect of duty cycle and its enhancement of heat transfer upon a heated surface relative to that of steady flow. The purpose of this study is to investigate the effect of a change in duty cycle of an impinging air jet on the convective heat transfer from an isoflux surface. Three duty cycles and two frequencies are considered and the results are compared to that of a steady jet. These comparisons are made through the synthesis of a time averaged Reynolds number (Re) for all cases to support the underlying assumption of enhanced efficiency relative to the volume of flow provided by a jet.

2. Facility description

The pulsed jet was generated by an ASCO 8238 two-way solenoid valve digitally controlled with an OPTO 12 series relay through a National Instruments USB-6009 A/D Converter. Note that the use of a two-way valve during the conduct of this study does yield a limitation to the reliability of measurement when compared to a three-way valve. Previous studies have shown a much more smooth temporal response during transients through application of a three-way valve [34]. Approximately 32 hydraulic diameters downstream of the valve the jet was delivered via a 1.55 mm inner diameter (ID) copper pipe. Upstream of the valve the bulk flow was controlled with a manual flow control valve and was immediately followed by an inline check-valve (not shown in Fig. 1) which was connected to the solenoid valve with 2.5 mm ID heavy wall silicon tubing. The check-valve was incorporated into the design so as to mitigate influences of back-pressure on the bulk flow rate control due to compressibility of the fluid. Heavy-wall silicon tubing was also selected in order to mitigate the impact of compressibility influences of the fluid as it exited the delivery system. Attached to the flow control valve was a Gilmont rotameter used to measure the flow rate of the steady jet stream. Flow was driven by a volume of air kept at 80 psig and stored in an 80 gallon compressed air tank connected to the flow control valve via the same heavy-walled silicon tubing which had an effective tubing length of approximately 100 tube diameters; efforts were made to minimize the lineal tube length to mitigate time-lag of fluid transport between the compressor tank and the delivery nozzle.

Two experimental configurations were employed during this study; one to characterize the free jet flow and the second to measure the time averaged heat transfer coefficients of the impinging flow. For characterization of the free jet, a TSI 1201 single film anemometer was used to capture local air speed measurements. The jet nozzle was located using a Parker–Hannifin two degree of freedom (DOF) traverse which provided positioning accuracy of 0.0005 mm in both DOFs. This configuration allowed nozzle-to-Hot Wire Anemometer (HWA) alignment in both the axial and traverse directions. The positioning was measured using a set of Whitworth calipers which digital read out to 0.001 mm, resulting in a positioning uncertainty of 0.0005 mm for all measurements collected during the study.

Heat transfer coefficients were captured using the apparatus schematic shown in Fig. 1. As in the free jet flow configuration the nozzle position was controlled using the Parker–Hannifin stage. The jet was directed towards a stainless steel 304 alloy foil having a 71.64 mm by 71.06 mm cross section and of 0.0127 mm thickness. This foil was held in place through compression between a copper bus bar and insulative base (i.e. silicon and ceramic). A constant DC voltage from a BK 1692 DC power supply was applied across the foil by a bus bar in order to achieve an isoflux condition on the foil surface. The electrical resistivity of the stainless foil which was utilized is approximately 72 μohm-cm at ambient temperature (20 °C) and does not begin to change until under thermal conditions until its temperature exceed 200 °C which was never reached in any experimental conditions considered herein. Therefore it is an appropriate assumption that given a known constant power (for each respective test) and constant electrical resistance (for all tests), the foil operates under an isoflux condition. To determine the power flowing through the foil a voltage measurement was taken across the copper bus, using a Tektronix DM 50 multimeter, and the current flowing through the supply lines was measured using a Fluke 337 clamp meter. Full field temperature measurements were captured with a TVS-8500 infrared thermal imaging camera positioned on the opposite side of the foil from the jet. While the camera has a maximum acquisition rate of 120 frames per second, all tests performed in support of this study were performed at a rate of 60 frames per second so as to improve