International Journal of Heat and Mass Transfer 53 (2010) 3874–3886

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Experimental investigation of impingement heat transfer on a flat and dimpled plate with different crossflow schemes

Yunfei Xing *, Bernhard Weigand

Institut für Thermodynamik der Luft- und Raumfahrt (ITLR), Stuttgart University, Pfaffenwaldring 31, 70569 Stuttgart, Germany

ARTICLE INFO

Article history:
Received 13 October 2009
Received in revised form 13 April 2010
Accepted 27 April 2010
Available online 2 June 2010

Keywords:
Impingement cooling
TLC
Heat transfer
Dimple

ABSTRACT

A nine-by-nine jet array impinging on a flat and dimpled plate at Reynolds numbers from 15,000 to 35,000 has been studied by the transient liquid crystal method. The distance between the impingement plate and target plate is adjusted to be 3, 4 and 5 jet diameters. Three jet-induced crossflow schemes, referred as minimum, medium and maximum crossflow correspondingly, have been measured. The local air jet temperature is measured at several positions on the impingement plate to account for an appropriate reference temperature of the heat transfer coefficient. The heat transfer results of the dimpled plate are compared with those of the flat plate. The best heat transfer performance is obtained with the minimum crossflow and narrow jet-to-plate spacing no matter on a flat or dimpled plate. The presence of dimples on the target plate produce higher heat transfer coefficients than the flat plate for maximum and minimum crossflow.

1. Introduction

The last 20 years have seen a large improvement in gas turbine technology, due to mainly an increase in turbine pressure ratio and turbine inlet temperature. The effect of firing temperature is very important: for every 100 °F (55.5 °C) increase in temperature, the work output increases approximately 10% and gives about 1–1.5% increase in thermal efficiency [1]. In practice, high turbine inlet temperatures have been achieved because of the growth of materials technology, new coatings and new cooling schemes. Film cooling methods are mainly considered in the past for the cooling of the combustor liner. But now the modern dry low emission (DLE) combustor is required to produce low NO_x emissions. The control of the NO_x problem requires to minimize film cooling and dilution air. These combustors are typically cooled by enhanced backside convective heat transfer. The liner has a double-wall structure and impingement cooling is often used to keep the cooling effectiveness high.

Numerous investigations on flow and heat transfer characteristic of multiple jet impingement have been published in order to tailor the impingement hole shape, size and locations to attain both a sufficiently high average heat transfer coefficient and the uniformity in the surface distribution to avoid local hot or cold spots. Han and Goldstein [2] published in 2001 a review of jet impingement heat transfer of a single and multiple jets in gas turbine systems. The variation of local Nusselt number with jet Reynolds number, jet-to-plate spacing and interactions between multiple jets have been discussed.

There have been a number of attempts to complement jet impingement with other enhancing techniques such as crossflow, ribs and turbulators in order to effective heat transfer with low pressure loss. Dimple arrays are firstly an attractive method for internal cooling channels, since they produce time varying multiple vortex pairs which augment local Nusselt number distributions downstream of the dimple. Now dimpled plate has become into the consideration due to its potential in heat transfer augmentation, light weight, low pressure penalty and low maintenance [3].

The impingement on a dimpled plate has not yet been well apprehended because of numerous complications. Gau and Chung [4] measured slot jet impingement on concave and convex surfaces by varying the Reynolds number from 6,000 to 350,000 and the slot to plate spacing from 2 to 16. They found that the Nusselt number increases with increasing surface curvature for impingement on a concave surface. Azad et al. [5] measured an array of inline air jets impinging on dimpled target plates using a transient liquid crystal technique with three different spent air crossflow orientations, jet Reynolds numbers ranging from 4,850 to 18,300 and a jet-to-plate spacing H/d of 3. The dimple diameter is equal to the jet diameter ($D_d/d = 1$) and the dimple depth $t_d/D_d = 0.5$. The results showed that the Nusselt numbers for a dimpled and a flat plate are about the same. Ekkad and Kontribitz [6] investigated the effect of jet impingement on a target plate with a dimple pattern for Reynolds numbers varying from 4,800 to 14,800 and the jet-to-plate spacing $H/d = 3$. The dimples diameter of $D_d/d = 2$ and two different dimple depths of $t_d/D_d = 0.1$ and 0.2 have been

* Corresponding author. Tel.: +49 711 685 62444; fax: +49 711 685 62317.
E-mail address: yunfei.xing@itlr.uni-stuttgart.de (Y. Xing).

0017-9310/$ - see front matter © 2010 Elsevier Ltd. All rights reserved.
doi:10.1016/j.ijheatmasstransfer.2010.05.006

Nomenclature

A	open area of exit rims (m^2)
c	specific heat (J/(kg K))
C_d	discharge coefficient (−)
d	impingement jet diameter, target plate thickness (m)
D_d	dimple diameter (m)
H	heat transfer coefficient (W/(m^2 K))
h	jet-to-plate spacing (m)
k	thermal conductivity (W/(mK))
L	target plate length (m)
$\dot{m}$	mass flow (kg/s)
n	number of impinging jets (−)
N	discrete interval (−)
Nu	Nusselt number, based on nozzle diameter (−)
$\overline{Nu}$	spanwise averaged Nusselt number in x direction (−)
$\overline{\overline{Nu}}$	area averaged Nusselt number (−)
p	pressure loss (Pa)
Pr	Prandtl number (−)
Re	Reynolds number, based on jet diameter (−)
R	dimple radius (m)
T	temperature (°C)
t	time (s)
t_d	dimple depth (m)
x, y	coordinate (m)
X	jet hole spacing in x direction (m)
Y	jet hole spacing in y direction (m)

Greek symbols

ρ	density (kg/m^3)
Θ	temperature ratio (−)

Subscripts

0	initial condition
B	bulk
curv	curved surface
d	dimple
dimpled	dimpled plate
flat	flat plate
i	index
I	impingement plate
O	exit rim
W	wall

investigated. The results showed that the presence of dimples on the target plate produces lower heat transfer coefficients than the flat plate. Kanokjaruvijit and Martinez-Botas [3,7–10] made a series of investigations on the impingement heat transfer on dimpled plate with different crossflow. The jet Reynolds number was in the range of 5000–15,000, and jet-to-plate spacings H/d from 1 to 12 were used. The effect of dimple geometry was considered with two different dimple configurations: hemispherical and cusped elliptical with the same wetted area and they found that both dimples showed similarity in heat transfer results. The investigated dimples parameters are dimple depths (t_d/D_d) of 0.15, 0.25 and 0.29, and dimple diameter (D_d/d) of 0.866, 1.732, 2 and 4. The shallow dimples (t_d/D_d = 0.15) improved heat transfer significantly by 70% at H/d = 2 compared to that of the flat plate, while this value was 30% for the deep ones (t_d/D_d = 0.25). The improvement also occurred for the moderate and lower D_d/d. They found that the total pressure is independent of target plate geometry when $H/d \geqslant 2$. The levels of the total pressure loss of the dimpled plates are not different from those of the flat plate under the same setup conditions. Woei et al. [11,12] made investigations on the heat transfer for an impinging jet array onto two enhanced plates using concave and convex dimples with effusion for the Reynolds number varying from 5,000 to 15,000 and a jet-to-plate spacing H/d from 0.5 to 11. The investigated dimple parameters are dimple depths (t_d/D_d) of 0.3, and dimple diameter (D_d/d) of 3.5. The results showed that the heat transfer performance with convex dimples is better than their counterparts with concave dimples without effusion. An increase of H/d reduces heat transfer differences between the effusion and non-effusion results for both concave and convex-dimpled surfaces.

The objective of the current study is to investigate the heat transfer and pressure loss values for the impingement on a flat and dimpled plate. Although the results in the literature mentioned above provide many inside into the heat transfer performance on the dimpled plate, their jet arrays and flow arrangements are quite different from the present case. The two most important of these unexplored areas are higher jet Reynolds numbers and calculation method of heat transfer value inside the dimples. Here various geometric paramedics such as Reynolds number, jet-to-plate spacing (H/d) and crossflow schemes are chosen to explore the possibility to enhance the heat transfer. All results from the dimpled plate are also compared to those from the flat plate.

2. Experimental setup

2.1. Test section

Fig. 1 shows a sketch of the experimental setup. A vacuum pump system is used to generate the desired air flow in the test channel. The air enters the channel under atmospheric conditions via a filter and a heater. The heater consisted of several meshes made out of stainless steel and is able to heat the air within less than 0.3 s from ambient temperature up to 100 °C. Downstream of the heater the air enters the inlet plenum and then the impingement model. This model is equipped with thermocouples and pressure taps for the measurement of the heat transfer and pressure loss. It consists of an impingement plate, a target plate and side rims with effusion outlet holes, as shown in Fig. 2. The spent air flows through the outlet holes on the exit rim to the outlet plenum. The target plate is made out of perspex, because it has low thermal conductivity and allows optical access, needed for the heat transfer measurements. The target plate is observed from the outside of the outlet plenum with two CCD video cameras. The model is symmetrical, therefore the target plate is only observed for half of the model by two cameras from left to right side.

There are a total of 81 impingement holes for the inline impingement plate. The ratios of jet-to-jet spacing in both directions on the impingement plate are the same ($X/d = Y/d = 5$). Because of temperature gradient of the inlet flow from center to corner, it is quite necessary to install many thermocouples on the impingement plate to certain the local reference temperature which is needed for the heat transfer evaluation. Fig. 3 shows the inline impingement pattern used in the scope of the present work and the positions of the thermocouples used for the data evaluation. Because the impingement plate is symmetrical, only half of it is presented. These thermocouples are placed directly in the center of the impinging hole at the jet exits. The thermocouples are placed directly at the jet exits. Grooves are milled into the wall of the impingement plate and the thermocouples along with their wires are glued into these grooves.