International Journal of Heat and Mass Transfer 53 (2010) 4663–4671

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Heat transfer enhancement in rectangular channels with axial ribs or porous foam under through flow and impinging jet conditions

Akhilesh P. Rallabandi, Dong-Ho Rhee, Zhihong Gao, Je-Chin Han *

Turbine Heat Transfer Lab, Texas A&M University, College Station, TX 77843-3123, United States

A R T I C L E I N F O

Article history:
Received 11 December 2009
Received in revised form 26 May 2010
Accepted 26 May 2010
Available online 17 July 2010

Keywords:
Internal flow
Impingement
Enhancement
Rib
Porous media
Metal foams

A B S T R A C T

Heat transfer and pressure loss characteristics of a high aspect ratio duct are measured under both, jet impingement and channel flow conditions, respectively. For both cases, roughness elements in consideration are staggered and inline axial ribs. The spacing (P) to height (e) ratios studied are $P/e = 2$ and $P/e = 4$; the rib height (e) to channel height (H) ratio is 0.125. Also studied is an aluminum foam roughness with a porosity of 92% and a height to channel height ratio of 0.15. Reynolds numbers considered for the channel flow case (based on the hydraulic diameter) range from 10,000 to 40,000. Reynolds numbers for the jet impingement case (based on the hole diameter) range from 5,000 to 20,000. Tests are performed using the copper plate regional average method. Results show a 50–90% increase in heat transfer due to the use of axial ribs in both, impingement and channel flow cases. The porous foam shows a more significant increase in heat transfer coefficient for both channel flow and impingement cases.

1. Introduction

Heat transfer enhancement is of vital importance to the industry. Limited capacity for the removal of generated heat is a significant bottleneck in the miniaturization of electronic circuitry. It also places a restriction on the turbine inlet temperature of a gas turbine engine – and thus its efficiency. Aggressive internal turbine blade cooling is therefore a very active current research pursuit. The focus of the current study is on the usage of roughness elements in two common internal cooling configurations: jet impingement cooling and turbulated channel cooling.

In gas turbine engines, jet impingement cooling is used in regions of very high thermal loading – such as combustor liners, blade leading edge [1] and stator mid-sections [2]. It is also used in the refrigeration and food engineering industry [3], electronics cooling [4] as well as in the textile industry. In general, jet impingement provides very high heat transfer, though at the cost of a high pressure loss and some compromise in structural integrity due to the use of perforated regions. It is for this reason that it is not used in rotor blade mid-sections.

Jet impingement cooling has been extensively studied over the past few decades. Local measurements of single jet smooth surface impingement were reported by Viskanta [5]. A primary peak at the stagnation point and a secondary peak radially outward from the stagnation point are observed. The location of the secondary peak varies with the diameter to target plate distance ratio. Goldstein et al. [6] developed a correlation for single jet impingement heat transfer on a smooth surface. Goldstein and Seol [7] compared circular jets to slot-shaped jets, and concluded that circular jets offer a higher heat transfer coefficient for a given mass flow rate of the coolant.

In jet impingement cooling, it is conventional to use an array of jets rather than a single jet. A study of multiple jets, parameterizing jet spacing and arrangement was conducted by Florschuetz et al. [8]. In some cases, design constraints disallow fluid from being discharged from the impingement duct into the mainstream immediately after heat has been absorbed at the target region. This results in the spent fluid creating a significant cross-flow within the channel. Such a scenario exists in stator blade mid-chord cooling, as documented by Florscheutz and Metzger [9]. The cross-flow effect was studied using the transient liquid crystal method by Huang et al. [10] and naphthalene sublimation by Rhee et al. [11].

Since the leading edge of the turbine blade is a very heavily thermally loaded zone, some designs call for the provision of shower-head film cooling holes. It is common in these designs to make the air impinge on the internal surface of the leading edge, remove heat by impingement cooling and then discharge as film coolant through the shower-head holes. The curved target surface effect (simulating the blade leading edge) was studied by Chupp et al. [1] and Hrycak [12]. Results showed that Nusselt numbers tended to increase with increasing strength of curvature. Advanced cooling calls for very high velocities (in the compressible regime) within the jet hole itself. The Mach number effect was studied by

* Corresponding author.
E-mail address: jc-han@tamu.edu (J.-C. Han).

0017-9310/$ - see front matter © 2010 Elsevier Ltd. All rights reserved.
doi:10.1016/j.ijheatmasstransfer.2010.06.027

Nomenclature

D_H	hydraulic diameter of channel, $\frac{2HW}{H+W} = 42.16$ mm(1.67 in)
d	jet hole diameter, $= 6.35$ mm(0.25 in)
e	height of rib ($= 3.17$ mm(0.125 in)), thickness of metal foam layer ($= 3.81$ mm(0.15 in))
f	friction factor
h	heat transfer coefficient, W/m^2 K
H	height of channel, $= 25.4$ mm(1 in), H = Z
L	length of rib, $= 20$ e, 65.5 mm(2.5 in)
$\dot{m}$	mass flow rate, kg/s
Nu	channel flow Nusselt number, based on hydraulic diameter
Nu_d	jet impingement Nusselt number, based on jet diameter
P	spacing between ribs, $= 2$ e, 4 e
q''	heat flux, W/m^2
Re	channel flow Reynolds number, based on D_H
Re_d	jet Reynolds Number, based on d
S_x	streamwise spacing between jet holes $= 5$ d, 31.75 mm (1.25 in)
S_y	spanwise spacing between jet holes, $= 5$ d, 31.75 mm (1.25 in)
t	thickness of impingement plate, $= 12.7$ mm(0.5 in)
T	temperature, K
V	velocity in through flow configuration, m/s
V_j	velocity in jet hole, m/s
W	width of channel, $= 127$ mm(5 in)
Z	jet to target plate spacing, $= 25.4$ mm(1 in), Z = H

Greek symbols

ρ	density, kg/m^3
ν	kinematic viscosity, m^2/s

Goodro et al. [13]. For a given hole Reynolds number, it was found that increasing the Mach number increased the heat transfer on the target surface. A Mach number of 0.6 resulted in an increase of around 30%. Parsons et al. [14] studied impingement in a rotating frame of reference, to simulate rotor conditions. The effect of the Coriolis force in deflecting the coolant jet was also discussed.

Jet impingement on a rib-roughened surface was studied by Haiping et al. [15]. Placement of the ribs aligned with the jet hole resulted in low heat transfer; optimum heat transfer resulted when the jet hole was located between two ribs. Trabold and Obot [16] compared ribbed impingement with smooth impingement, and showed that ribs have beneficial effect in region dominated by cross flow. Rhee et al. [11] studied impingement heat transfer with effusion cooling holes, and concluded that effusion holes actually increase the heat transfer coefficient.

Internal channel heat transfer is used extensively in the rotor blade passages owing to its relatively low pressure drop. This configuration has been under investigation for most of the previous century. Seminal works by Dittus and Boelter [17] characterized the heat transfer coefficient due to turbulent flow in smooth channels. Kays et al. [18] offers a comprehensive summary of various related correlations for flow inside channels with various roughness elements.

In the past few decades, there has been interest in using skewed rib turbulators to generate secondary swirls in the bulk fluid flow, increasing the turbulence and heat transfer. A massive database pertinent to aircraft engines exists in the literature, e.g. Han and Zhang [19] and Taslim et al. [20]. More recently, these results have been extended to land based power generation turbines [21].

Newer manufacturing technologies have now made possible the fabrication of metal foams – porous materials with low metal to void ratio ($\approx 8\%$). Metal foams can be used in electronic heat sinks to improve the performance of impingement and channel flow cooling. Turbine blade internal serpentine passages (conventionally roughened by ribs) can also be roughened with metal foam to enhance heat transfer in through flow configurations; internal passages within leading edges can be roughened with foam to enhance heat transfer under jet impingement. The internal body of a gas turbine vane can also be cooled by jet impingement on a metal foam.

Enhanced convection due to metal foams has been modeled and studied by Calmidi and Mahajan [22]. Tzeng and Tzeng [23] discuss impingement on Al foam and predict using numerical methods, significant enhancements. More recently, experimental studies into using graphite foam to enhance impingement heat transfer coefficient by up to 60% have been documented by Sultan et al. [24].

Heat transfer coefficient under impinging jet configurations depends on several parameters, such as jet hole configuration and jet hole-target spacing. In this study, a typical jet hole configuration ($S_x/d = S_y/d = 5,\ t/d = 2$), with a typical jet hole-target spacing ($Z/d = 4$) is studied. The jet hole configuration is not varied. All velocities considered in this test are in the low Mach number incompressible regime.

Ribs considered in this study are axial rather than skewed or perpendicular to the bulk flow. It is anticipated that the friction factor offered by axial ribs in channel flow will be lower than skewed or perpendicular ribs, since axial ribs do not trip the flow field. An aluminum foam with 92% porosity is used as a porous roughness layer. The focus of the current study is not on parametric variations of the porous foam configuration. The foam in consideration is typical and easily available, and the height of the porous layer is similar to that of the ribbed cases.

The focus of the current study is to provide a comprehensive comparison of the heat transfer performance of axial ribs and a porous roughness layer under different cooling configurations. Another area of emphasis of the current study is the comparison of heat transfer and pressure penalty characteristics of two different commonly used cooling techniques – channel flow and impinging jet cooling.

2. Experimental setup and data reduction

High Pressure air, available at 120 psi (8 bar, abs) is ducted into the test section (Fig. 1). The flow rate of the air is regulated by a valve. A *T*-valve is used to duct the air to the desired test section – either the impingement test section or the channel flow test section. The air also passes through a heater which is utilized while running the transient liquid crystal (TLC) tests. An ASME 38.1 mm (1.5″) dia. square-edged orifice plate flow-meter is used to measure the mass flow rate. Pressure taps are provided both, upstream and downstream of the orifice plate, in order to measure upstream and differential pressures to calculate the flow rate in accordance with the ASME standard.

The test section is operated in two configurations, the channel flow configuration (Figs. 1 and 3(a)) and the jet impingement configuration as shown in (Figs. 1 and 3(b)). Aerodynamic entry into the channel flow configuration is fully developed; the height of the channel, H, is 25.4 mm (1″) and the width is 127 mm (5″), yielding a hydraulic diameter, $D_H = 42.4$ mm (1.67″). The length