International Journal of Heat and Mass Transfer 83 (2015) 294–306

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Film cooling effectiveness in the region of the blade-endwall corner junction with the injection assisted by the recirculating vortex flow

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ARTICLE INFO

Article history:
Received 1 July 2014
Received in revised form 7 October 2014
Accepted 19 November 2014
Available online 19 December 2014

Keywords:
Film cooling
Blade endwall
Junction heat transfer
Secondary flows

ABSTRACT

The region around the blade-endwall junction point in inlet guide vane blades of modern gas turbines constitutes one of the hardest "hot spots", requiring special attention for its cooling. The present study investigates the merits of a film cooling application where the coolant injection is implemented through a small rectangular slot on the blade (bluff body) symmetry plane (that incorporates the blade leading edge line) in such a way that the coolant flow is assisted by the recirculating flow which results from interaction of the phenomena generating the horseshoe vortex. The experimental results showed a very encouraging cooling effectiveness, especially given the tiny amount of the coolant flux.

1. Introduction

The blade-endwall junction presents one of the most difficult hot spots that must be cooled within the blade channels of the inlet guide vanes (IGV) of the turbine in any modern gas turbine. While several techniques have been proposed for blade cooling, the endwall region is the most complicated region largely due to the presence of strong three dimensional secondary flows, as where indicated by York et al. [1]. Three models of the flow field near the endwall of a turbine vane were proposed by Langston [2], Sharma and Butter [3] and Goldstein and Spores [4], each model exhibiting its own modulations. The latest most comprehensive and detailed secondary flow model was proposed by Wang et al. [5]. The significance of the endwall secondary flow for the turbine blade channel loss and heat transfer processes became known by the works of Blair [6], Georgiou et al. [7] and Graziani et al. [8]. Since then the main attention has been focused on to the heat transfer aspects, since the contribution of the two horseshoe vortex legs on the total pressure losses inside a cascade is rather small when compared to the contributions of the passage and tip leakage vortices. In order to address the problems associated with the enhanced wall heat transfer imposed by the secondary flows, several concepts have been proposed involving endwall film cooling techniques and blade leading edge modifications. Sauer et al. [9], Zess and Thole [10], Shih and Lin [11], incorporated various leading edge/endwall junction geometries so as to modify the relevant flowfield. The nature of the mechanisms by which a junction filet reduces aerodynamic losses and heat transfer for turbine vanes were investigated by Lethander [12], Becz et al. [13] and Han and Goldstein [14]. These studies showed that adding a filet to the airfoil-endwall junction marginally changes the endwall heat transfer distribution while it increases the wall shear and reduces the endwall flow turning at the passage exit. Additional filet geometries have been investigated by Mahmood and Acharya [15] and Zhang et al. [16].

As far as the endwall film cooling is concerned, a number of studies have been published (e.g. by Blair [17], Burd et al. [18], Oke et al. [19] etc), showing that the film cooling effectiveness depends on various factors like shape, size, location and orientation of the holes, the freestream relative speed and the temperature of the injected cool air, etc. Colban et al. [20] used a backward facing slot with several different coolant exit conditions while Kost and Nicklas [21] combined an upstream slot with film cooling holes within the passage of the vane. One of the most comprehensive and detailed study regarding endwall film cooling was performed by Friedrichs [22], in which, among others, he identifies the leading edge junction region as one of the most thermally distressed areas. Even today, the region presents significant difficulties in achieving an acceptable film cooling cover due to the fact that the endwall film cooling configurations employed so far utilize coolant ejections upstream of the blade leading edge juncture (e.g. Zhang and Jaiswal [23], Kost and Nicklas [24], Nicklas [25], Knost

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2014.11.067
0017-9310/

Nomenclature

S_t	Stanton number ($h/\rho C_p V_\infty = N_u/R_e P_r$)
T_∞	surrounding air temperature
d_T	injection air metering tubes inside diameter
L_T	injection air metering tubes length
V_T	velocity inside air metering tubes
ΔP	pressure difference on air metering device
f	laminar flow friction coefficient
$\dot{m}_c$	injection air mass flux
R_e	Reynolds number ($= V_\infty D_h/v$)
D_h	hydraulic diameter of the test section rectangular channel
P_r	Prandtl number
N_u	Nusselt number ($= hx/k$)
R	heated plate resistance
$\dot{q}_w$	heat flux on heated plate
$\dot{Q}_{el}$	electric power input
A_{heated}	exposed area of heated plate
$T_{B\infty}$	local bulk temperature of the mainstream
T_w	local wall temperature
C_p	specific heat capacity
I_r	blowing ratio (V_{INJ}/V_∞)
V_{INJ}	injection air velocity
V_∞	mainstream velocity
A_T	injection slot exit area
r	radius of bluff body
r^*	non-dimensional radius (r/R_{LE})
h	heat transfer coefficient
k	thermal conductivity
T_∞	mainstream temperature upstream of the test section
T_{ex}	mainstream air temperature downstream of bluff body at the exit plane
T_c	coolant air temperature
T_{ND}	non dimensional temperature (T_w/T_w^{UP})
T_w^{UP}	endwall temperature unaffected by the presence of the bluff body
X_{UP}	non dimensional upstream distance along endwall centerline
D	bluff body leading edge diameter
R_{LE}	bluff body leading edge radius
V_{PL}	voltage at the heated plate
S_{to}	Stanton number without injection
C	blade true chord or total length of bluff body
y^+	dimensionless wall distance
R_e^D	Reynolds number based on inlet velocity and leading edge diameter
R_e^C	Reynolds number based on inlet velocity and true chord
Tu	turbulence Intensity level

Greek

ρ	air density
v	kinematic viscosity
θ	degrees measured from the bluff body's leading edge centerline
Θ_c	non dimensional cooling parameter $(T_\infty - T_w)/(T_\infty - T_c)$
$\Theta_c^{N.I}$	non dimensional cooling parameter for the standard non injected case
δ	inlet boundary layer thickness
δ^*	inlet boundary layer displacement thickness
θ	inlet boundary layer momentum thickness
H	inlet boundary layer shape factor

and Thole [26,27]). In this type of injection the streamlines enclosing the injected coolant become separated and are displaced away from the endwall due to the action of the local junction flows. The strong influence of secondary flows on the coolant trajectory was established early by the works of Coldstein and Chen [28] and Jabbari et al. [29]. A recent study by Thrift and Thole [30] investigating the influence of the flow injection angle on the leading-edge horseshoe vortex revealed that injecting high momentum flow upstream of the junction led to an increase in local turbulence levels and endwall heat transfer when compared to the non injected case.

As it is indicated by the majority of researchers working in relevant configurations, the most prominent feature on the symmetry plane upstream the junction is the horseshoe vortex. It has been perceived and understood that the endwall heating and film cooling processes are significantly affected by these secondary flows. As a result it is difficult to achieve a high level of film cooling effectiveness within this region.

The present study provides experimental data in order to investigate the effectiveness of air injection as a coolant, when it is injected through a thin rectangular slot located on the symmetry plane near the junction (leading edge centerline) of the blade with the endwall and in a direction opposite to that of the mainstream. Thus the coolant jet follows the same direction of rotation as the horseshoe vortex and as a result it is expected to be assisted by the pumping action of the latter. This kind of injection is deemed applicable to the turbine inlet guide vanes, just after the combustion chamber, since the stream of the flue gases maintains a constant direction impacting upon the fixed stator blade leading edges. The resulting coolant mass flux constitutes a tiny fraction of the mainstream entering the blade channel. The experiments were conducted in the inverse mode (hot wall – cool mainstream – hotter "coolant") but the generated heat flux coefficients (Stanton numbers) are expected to represent the actual engine distributions quite accurately. The steady state heat flux distribution on the endwall near the junction point was measured by employing infrared thermography and this was collaborated by thermocouples embedded just below the heated wall. In the presence of the above injection configuration, the results indicated a sharp reduction of the non dimensional Stanton number distribution around the leading edge junction and for an upstream distance at least half the bluff body's radius. The measured Stanton number reduction percentage reaches the 50% levels on the leading edge symmetry plane and near the junction point, decreasing away from the bluff body. A significant symmetric reduction of Stanton number was recorded over an arc of at least $\pm 15^\circ$ around the leading edge centerline.

2. The experimental facility and inlet flow conditions

The tests were conducted in an open loop, subsonic low speed wind tunnel whose straight duct was 2 m long, with a constant rectangular cross sectional area of $0.30 \times 0.20\ \mathrm{m}^2$. The duct was attached to the wind tunnel's exit nozzle. The maximum speed at the exit of this nozzle was about 16 m/s, provided by a centrifugal blower as it is illustrated in Fig. 1. The exit nozzle was preceded by a 1,5 m long settling chamber integrating an array of 8 damping screens and a honeycomb grid at the end. At the exit plane of the nozzle, the flow was measured with a hot wire anemometer and was found to exhibit a maximum turbulence intensity of less than 0.25% and a mean velocity non–uniformity of less than 0.5% against the span-wise average value. A long 2,75 m diffuser with