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Heat transfer in a two-inlet rotating wedge-shaped channel with various locations of the second inlet

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ABSTRACT

The heat transfer characteristics of a two-inlet wedge-shaped channel with lateral fluid extraction are experimentally investigated under both rotating and non-rotating conditions. Three different locations of the second inlet are tested. The effective inlet Reynolds number is fixed at 15,000 whereas the mass flow rate ratio (*MR*, second inlet mass flow rate/major inlet mass flow rate) ranges from 0 to 1.0. The highest rotation number is around 0.23. The results show that injecting the secondary coolant at the top of the channel should be the best option in the view of optimizing the bulk temperature and the heat transfer of the high-radius corner of the channel in both rotating and non-rotating cases. Two critical *MR* can be identified in the non-rotating channels. Higher the radius of the coolant injection, lower the critical *MR*. The magnitude of the second critical *MR* is double of the first one in all three cases. However, the local critical mass flow ratio (MR_x) is almost independent of the location of the second inlet. In both rotating and non-rotating cases, operating at high *MR* over the critical point is not recommended because the cooling air is not used efficiently. The influence of the second coolant injection is limited in the vicinity of the major inlet, which brings the convenience to optimize the local heat transfers without disturbing the heat transfer at the upstream locations.

1. Introduction

In modern advanced gas turbines, many sophisticated cooling technologies have been adopted to protect the high pressure turbine blades from damage. In order to satisfy the demands, both external and internal cooling techniques have been employed [1,2]. In the fundamental aspect, the most popular and widely investigated topic was the rotating rectangular channel which was a simplified model of the internal cooling passage in the mid-span of a turbine rotor blade. Rotating generated a heat transfer difference between the leading wall and the trailing wall due to the effects of Coriolis force. The heat transfer on the leading wall was weakened whereas the trailing wall was promoted by rotation in a radial outward channel in the low rotation number regime [3,4]. Once the rotation number was high enough, the heat transfers on both leading and trailing walls were elevated by rotation. A critical rotation number could be observed on the leading wall, which was found to be a function of radius [5,6]. The high buoyancy induced reverse flow broke the thick thermal boundary on the leading wall and finally increased the heat transfer. Unlike the aforementioned cases, the thin turbine rotor blade trailing edge was a region difficult to cool due to the constraints of the shape of the turbine blade. A typical trailing-edge internal cooling passage had a wedge-shaped cross section, a large-angle channel orientation, pin–fins or ribs roughened walls, and lateral fluid extractions [7,8].

Before the studies of the rotating channels, non-rotating cooling channels had been investigated. The lateral fluid ejection brought significant difficulties to determine the local bulk temperature and mass flow rate in the internal cooling channel. Introducing lateral fluid extraction reduced the overall heat transfer in the pin–fin arrayed rectangular cooling channel [9,10]. The length of the ejection hole did not influence the heat transfer notably. The local mass flow rate could be determined by measuring the pressure drops of ejection holes [11]. The sucking effects of the ejection holes promoted the heat/mass transfer in the vicinity of the ejection holes but weakened that in the opposite region [12]. The effects of lateral-to-total flow ratio in a wedge-shaped channel were also investigated, and the worst overall heat transfer occurred as the

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Nomenclature

English symbols

A	area (m^2)
C_p	heat capacity at constant pressure (J/(kg·K))
D_h	hydraulic diameter (m)
h	heat transfer coefficient (W/(m^2·K))
I	current of heater (A)
L	length of the channel (m)
$\dot{m}$	mass flow rate (kg/s)
MR	mass flow rate ratio
Nu	Nusselt number
P	heat transfer coefficient measured point in X-direction
Pr	Prandtl number
r	distance from rotation axis to channel inlet (m)
R	resistance of each heater (Ω)
Re	Reynolds number
Ro	rotation number
T	temperature (°C)
U	mean velocity of coolant (m/s)
WP	wetted perimeter (m)
X	coordinate direction

Greek symbols

α	wall-to-environment heat-loss coefficient (W/K)
β	angle from channel symmetrical plane to rotating plane (°)
μ	viscosity of coolant (Pa·s)
Ω	rotational speed (rad/s)
λ	heat conductivity coefficient (W/(m·K))
ρ	density of the coolant (kg/m^3)
ϕ	a standard quantity of mass flow rate (kg/s)

Subscripts

1	inlet at the bottom of the channel (main stream)
2	inlet at the top of the channel (second stream)
ave	averaged parameter
b	bulk temperature
eff	effective parameter based on total inlet mass flow rate
i	number of the measured point in the X direction
in	inlet
$loss$	loss
net	net
s	stationary
w	wall
x	local parameter
0	fully-developed turbulent flow in circular pipe with smooth wall

Superscript

$*$	normalized with major inlet parameter

ratio was around 0.3–0.4 depended on the shape of pin–fins [13–15].

In the investigations of rotating channels, the rectangular channel with a high aspect ratio (AR) and a large angle of channel orientation (AR = 4:1, β = 135°) was considered as the model of the trailing edge cooling channel at the first beginning. It was found that the heat transfer on both the leading surface and the trailing surfaces were enhanced by rotation [16]. The effects of the aspect ratio and channel orientations were also investigated [17–19]. The aspect ratio was a critical parameter in the formation of the Coriolis force induced secondary flow, and in turn influenced the channel heat transfer [17]. Compared with high and low AR channels, the square channel (AR = 1) presented the most significant leading-to-trailing heat transfer difference. The channel with an aspect ratio of 1:4 had the best thermal performance when taking the effects of friction factor into account [18]. However, the difference was weakened when the channel orientation changed from 90° to 135° [19].

The passages with wedge-shaped cross section were employed to mimic the blade trailing edge cooling channels. It was found that the heat transfer on all of the surfaces were enhanced by rotation in a smooth channel without lateral fluid extraction [20]. The lateral ejection weakened the rotation induced heat transfer enhancement in the regions next to the slots compared with the non-ejection cases [21]. Adding the pin–fins in the channel promoted the overall heat transfer and reduced the effects of rotation [22]. The rotating rib-roughened channel was also investigated. The ejections had pronounced effects on heat transfer distribution [23]. The heat transfers were reduced by the fluid extraction in both rotating and non-rotating cases [24]. The channel with mixed rib and pin–fin turbulators was also investigated [25]. What is more, the channel orientation was found to be a crucial parameter in influencing the heat transfer, and the most significant rotational effects were observed in the case of β = 90° [8,26]. Once the Coriolis force of the main flow directed to the outer region, the heat transfer of the channel was abated, and vice versa. Coriolis force may drag the main flow to the inner side or outer side depended on the channel orientation. Therefore, a critical channel orientation was observed [8]. The downstream local mass flow rate reduced significantly due to the side-wall fluid extraction, which rendered a low heat transfer region located at the inner top of the passage. Therefore, a second inlet was introduced at the location with minimum heat transfer coefficient. The rotational effects on flow and heat transfer in the configuration were investigated. The second inlet notably improved the heat transfer in that region and compensated the negative effects induced by rotation. Moreover, the overall averaged heat transfer was enhanced by the second coolant injection with a major-to-second inlet mass flow ratio of around 0.3 in the rotating conditions [7].

As mentioned above, the effects of second coolant injection are phenomenal, however, it is unknown whether the location of the second inlet would play an important role. Therefore, the objective of the current work is to experimentally study the effects of second-inlet location on flow and heat transfer at different rotation number and MR in a smooth wedge-shaped rotating channel with lateral fluid extractions.

2. Experimental facility

Fig. 1 shows the schematic of the rotating test platform, which consists of four major modules: electric motor, rotating arm with support, temperature measurement module and slip rings. The test section is assembled on the rotating arm, and the channel orientation is illustrated in the figure as well. The symmetrical plane of the channel orientates at an angle of 135° with respect to the rotating plane. Two channels of compressed cooling air are supplied through a two-pass rotary union and the tubes in the hollow shaft to the test section. The mass flow rate of each stream is measured with a FCI-SC98 thermal flowmeter before the rotary union. The T-type thermocouples are connected to the temperature measurement module which includes an isothermal cavity and the