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Heat transfer characteristics in a rotating trailing edge internal cooling channel with two coolant inlets

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ABSTRACT

Heat transfer performances in two-inlet wedge-shaped channel with side-wall coolant ejection are experimentally studied at both rotating and stationary conditions. The test section is used to model a novel turbine blade trailing edge internal cooling structure. The coolant mass flow rate ratio (*MR*, top-to-total coolant mass flow rate) ranges from 0 to 1.0. The major Reynolds number and rotation number are varied from 10,000 to 25,000 and 0 to 1.06, respectively.

At non-rotating condition, two-inlet inflow promotes heat transfer of top-half outer region but reduces that of bottom-half channel. The *MR* and overall averaged *Nu* ratios (based on effective Reynolds number) are approximate parabolic relation, and the lowest heat transfer case is corresponding for the evenest channel heat transfer distribution. In rotating cases, the increasing *MR* decreases the span-wise heat transfer differences for both trailing and leading walls, and so does the trailing wall heat transfer enhancement. Also, the effect of rotation reduces and trailing-to-leading surface heat transfer difference is relatively small due to two-inlet inflow. The rotational heat transfer at low-radius half channel is dominated by bottom coolant generated radially outward flow, but high-radius half is dominated by two-inlet inflow mixing.

1. Introduction

In order to achieve higher power output and thermal efficiency, modern advanced gas turbine operates at extreme high turbine inlet temperature which is far beyond the allowable temperature of blade material. Efficient cooling systems, both internal and external cooling schemes, are required to protect the turbine blades from damage [1,2]. In the fundamental aspect, the most widely investigated topic was the rotating rectangular channel which was a simplified model for the internal cooling passage in turbine blade middle region. Due to the rotation induced Coriolis force effect, heat transfer on the trailing wall was promoted whereas the leading wall was weakened in a radial outward channel at low rotation numbers [3,4]. Rotational effect reduced in radial inward channel compared to radial outward one [5]. As the rotation number increased high enough, heat transfer on both leading and trailing wall were elevated by rotation [6]. A critical rotation number, corresponding to the lowest heat transfer level, was observed at different locations of the leading wall [7]. This critical rotation number was found to be a function of non-dimensionless location, channel orientation and wall-temperature ratio [8,9]. Also, at the downstream of the rotating inward flow passage, heat transfer on the suction side inversely exceeded the pressure side [9]. The strong combined effect of Coriolis and buoyancy forces from rotation was accounted for this phenomenon. The measurement of flow field and the development of related technology significantly helped to understand the heat transfer characteristics inside the channel [10,11]. The measured secondary flow results inside the turn were successively used to explain previous heat transfer trends. And the flow characteristics in a parallelogram channel with and without ribs were also reported [12,13].

However, unlike the regular rectangular cross-section channel, the wedge-shaped convergence channel was selected for the blade trailing edge due to the constraint of the turbine blade. A typical trailing edge internal cooling channel had a wedge-shaped cross-section, pin–fins or ribs roughened walls and lateral fluid extractions [14,15]. The side-wall fluid ejection caused reducing mass flow rate along the channel. The ejections enhanced heat transfer

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Nomenclature

English symbols

A	area (m^2)
Buo	buoyancy parameter
D_h	hydraulic diameter (m)
h	heat transfer coefficient (W/(m^2·K))
I	the current of each heater (A)
$\dot{m}$	mass flow rate (kg/s)
MR	mass flow rate ratio
Nu	Nusselt number
n	rotating speed (rev/min)
Pr	Prandtl number
Q	heat energy (W)
r	rotating radius (m)
R	resistance of heater (Ω)
Re	inlet Reynolds number
Ro	inlet rotation number
T	temperature (K)
U	mean velocity (m/s)
X	coordinate direction (m)

Greek symbols

α	heat loss coefficient (W/K)
β	channel orientation
Ω	rotation rate (rad/s)
μ	viscosity of the coolant (Pa·s)
ρ	density (kg/m^3)
λ	heat conductivity coefficient (W/(m·K))

Subscripts

1	inlet at the bottom of the channel
2	inlet at the top of the channel
ave	average
b	bulk
cu	copper plate
e	environmental parameter
eff	effective parameter based on total inlet mass flow rate
i	the number of the measured point in X direction
in	inlet for heated channel
loss	loss
major	parameters based on major inlet
net	net
s	stationary
w	wall
0	fully-developed turbulent flow in non-rotating smooth circular pipe

in the region close to the holes but presented negative effect in the opposite inner region [16,17]. But the overall heat transfer was reduced compared to the non-ejection case [18]. The effect of lateral-to-total flow ratio on heat transfer in a trapezoidal pin–fin arrayed channel was also concentrated [19,20]. The worst overall heat transfer was observed when this ratio approached to around 0.3 or 0.4 depended on the shape of pin–fins.

The wedge-shaped cross-section with rotation also played an important role in rotational heat transfer distribution. Significant rotational heat transfer enhancement was observed in a smooth wedge-shaped channel without lateral coolant extraction [21]. The lateral ejection weakened the rotational effect in the region next to the slots [22,23]. But the lateral coolant extraction was greatly affected by rotation [24]. More mass flow split at the slots of high-radius regions when the channel rotated. As the pin–fins or ribs were inserted, the effects of rotation were further reduced. But the overall heat transfer was effectively elevated [25,26]. The similar influence was also observed in a wedge-shaped channel with mixed rib and pin–fins [14]. Due to the span-wise convergence and lateral ejection, a wide range of low heat transfer area was observed at the inner section of the downstream region [15,22]. The significant stagnant flow vortex was considered to be attributed to the low heat transfer performance [27]. Besides, the outlet boundary condition was found to be more influential at non-rotating conditions than rotating ones [28]. And higher rotation number significantly weakened the heat transfer differences induced by the second outlet passage. Recently, a rotating two-inlet wedge-shaped cooling concept was reported [29]. The second coolant was introduced to enhance heat transfer of the high-radius region. Compared with the non-rotating case, the overall averaged heat transfer could be enhanced by the second inlet once the mass flow ratio was around 0.3.

The current study extends study from Li et al. [29] by considering a full range of mass flow rate ratio and the rectangular top inlet structure. The current work aims to provide additional information for heat transfer performance in such rotational two-inlet cooling channel and the coupling effect of rotation and mass flow rate ratios on heat transfer distribution.

2. Experimental apparatus

2.1. Rotating facility

The rotating test rig is shown in Fig. 1. Four independent parts are included: electric motor, rotating arm with the support, temperature measurement module, slip rings. A two-pass rotary union is used for providing two passes of individually controlled coolants into the rotating test section. Two separated FCI-98 thermal flowmeter are installed in front of each branch of coolant to measure the mass flow rate, which is controlled by an electromagnetic valve. The total mass flow rate of these two branches is also measured by an additional flow-meter for double-check. A rotating back pressure control valve is fixed near the rotating axis that ensures high air pressure inside the channel.

The test section is assembled on the rotating arm that has a maximum diameter of 1.3 m. Before transmitted to the non-rotating facilities through slip rings, the analog signals of the thermocouples are converted into digital ones by AD-chips that inside the temperature measurement module. The reference parts of all the thermocouples are placed in a rotating insulated cavity that the temperatures of the thermocouple cold ends are stable and uniform. The temperature of this cavity is measured by several DS18B20 chips. Thus, only four slip rings (two for chips power supply and two for data transmission) are required for hundreds of temperatures measurement. More detailed facility details and validations can be found in our previous works [7,9].

2.2. Test section

The schematic of test section with two inlets is shown in Fig. 2. The cross-section of the channel is a wedge with a height of 54 mm. The 20 mm wide edge of this trapezoid narrows to 6 mm and gives the apex an angle of 15°. The hydraulic diameter of the cross section is 20.9 mm. The length of heated section is 230 mm ($L/D_h = 11$). Before the bottom inlet, a 63 mm length unheated entrance locates with an expanded cross-section. The cross-section shape is an isosceles trapezoid with a height of 34.5 mm.