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AN EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER CHARACTERISTICS IN STEAM-COOLED SQUARE CHANNEL WITH RIB TURBULATORS

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ABSTRACT

In recent years, steam is used in vane internal cooling ducts as a new coolant to replace compressed air. Much research has been carried out into the closed circuit steam cooling for vanes. But most of the existing literatures only provide information on the overall cooling effect of vanes. The effects of steam parameters on the cooling effectiveness of the vane internal channel are still not very clear. A steam heat transfer enhancement test platform of turbine blade(vane) internal channel at Equipment Cooling Technology Laboratory, Xi'an Jiaotong University was built to study the steam cooling characteristics in blade(vane) internal cooling channel. In this paper, heat transfer data of a steam-cooled duct have been experimentally obtained under different operating conditions. The range of key governing parameters is presented as follows: Reynolds numbers based on the channel hydraulic diameter(10000-60000), entry absolute pressures(0.3Mpa-0.6Mpa),heat flux of heat transfer surface area(5.5kWm^{-2}-22kWm^{-2}), steam superheat(0K-40K). The results show that the average Nusselt numbers along the centerline of the ribbed wall of the steam-cooled square duct are about 15 percent higher than that of air, but the overall heat transfer characteristics of steam in the square duct are similar to that of air under the same conditions.

NOMENCLATURE

A	[m^2]	heat transfer surface area (not include the rib surface area)
D	[mm]	hydraulic diameter of test channel
e	[mm]	rib height
H	[mm]	flow channel height
h	[W/(m^2·K)]	heat transfer coefficient
Q	[Wm^{-2}]	heat flux
L	[mm]	channel length
Nu	[-]	Nusselt number
Nu_{steam}	[-]	Nusselt number for steam flow in a channel
Nu_{air}	[-]	Nusselt number for air flow in a channel
$\overline{Nu_{steam}}$	[-]	average Nusselt number for steam flow in a channel along centerline of the ribbed wall
$\overline{Nu_{air}}$	[-]	average Nusselt number for steam flow in a channel along centerline of the ribbed wall
P	[Mpa]	inlet pressure of steam
p	[mm]	rib pitch
q	[W/m^2]	heat flux of heat transfer surface
q_{loss}	[W/m^2]	heat loss
Re	[-]	Reyonld number
T_w	[K]	local wall temperature
T_b	[K]	bulk mean temperature of steam
$T_{superheat}$	[K]	superheat of steam
W	[mm]	flow channel width
x	[mm]	axial distance from channel entrance
α	[°]	rib angle-of-attack

INTRODUCTION

The development of advanced cooling technology is the primary method to improve the performance of gas turbine. Currently, complex air cooling technology was used in the gas turbine blade(vane). However, as the turbine inlet temperature increases, the traditional air cooling technology is increasingly hard to deal with the requirement of high efficiency, high reliability of gas turbine. It has become the key bottleneck in the development of gas turbine. Steam has better heat transfer performance than air, so if steam was used as the new coolant in internal channel of gas turbine blade(vane), the cooling efficiency will be significantly improved. Much research has been carried out into steam cooling technology[1-6]. Some companies of USA and Japan such as GE, Westinghouse etc have successfully developed the most advanced heavy-duty gas turbines using steam cooling technology, whose simple cycle efficiency is as high as 39% and the combined-cycle efficiency based on those gas turbines is about 60%[7].

Steam cannot be ejected into the gas path relative to air, or the temperature of gas will drop sharply, and causing the lower cycle efficiency, so steam cooling of turbine(vane) was designed as a closed-loop. Steam cooling technology is proposed in 90s of the last century, so far, many research institutions have done a lot of work on the steam cooling technology. Corman[8] and Fukue[9] discussed the advantages of steam cooling and cycle efficiency etc. H.Nomoto et al.[10] did an experimental investigation of steam cooling technology in vane internal channel with a closed-loop design using a hot wind tunnel which simulated the actual conditions of gas turbine. The results of the hot wind tunnel tests show that steam has excellent cooling effectiveness for vanes, these also show that the steam is a promising coolant for the combined cycle of the next generation. Dieter bohn et al.[5] has numerically analyzed a steam-cooled test vane in a cascade with external hot steam flow using the in-house code CHTflow, The results from two investigated cases show that the steam cooling technique has a good application potential in the steam and gas turbines. In addition TingWang et al.[4] did some experimental research on steam impingement cooling technology, the results show that cooling effects of steam impingement cooling technology is very good. Steam cooling technology has been extensively evaluated and used as a new coolant, and the cooling effect of steam cooling technology on blade(vane) has been widely studied. However, most of the existing literatures only provide information on the overall cooling effect of vanes. The effects of steam parameters on the cooling effectiveness of the vane internal channel are still not very clear.

In this paper, the steam cooling characteristics in blade(vane) internal cooling channels have been experimentally studied. Internal cooling channel of blade(vane) is usually modeled as square or rectangular ribbed duct with different channel aspect ratios[11]. The main parameters impacting steam heat transfer effects in blade(vane) internal channel are as follows: Reynolds number (Re), channel aspect ratio (W/H), rib angle (α), rib pitch (p/e), channel blockage ratio (e/D), pressure (P), heat flux （Q）. This research object is a square ribbed channel with a length of 1000mm, e/D = 0.048, p/e=10, α=60° and the square channel has a cross-section of 40×40mm.The range of key governing parameters is presented as follows: Reynolds numbers based on the channel hydraulic diameter (10000-60000), entry absolute pressures(0.3Mpa-0.6Mpa), heat flux of heat transfer surface area ($5.5kWm^{-2}$-$22kWm^{-2}$), steam superheat (0K-40K).

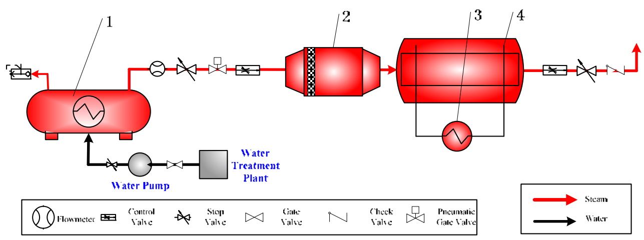
EXPERIMENTAL APPARATUS AND DATA REDUCTION

Experimental Apparatus

The schematic of the experimental apparatus is shown in Fig.1. The test system consists of a steam generator, a plenum, test channel, heating device of the test channel, control system and data acquisition system.

The high temperature steam supplied by the steam generator flows through the plenum to the test channel, and then turn on the heating device, heating the steam flowing through the test channel on a certain heat flux, finally, the steam was exhausted into the atmosphere. The key parameters of the steam generator are as follows: steam temperature: 373-503K; pressure: 0.1-0.8 Mpa; maximum steam flow: 600kg/h.

The steam generator consists of a common electric boiler and a steam superheater, so we can adjust the evaporation of the boiler and the temperature of the superheater according to different test conditions. The steam generator forced steam through a 5 cm diameter pipe to the test channel, equipped with a flowmeter, a stop valve, a pneumatic gate valve, a control valve to measure and control the flow rate. A stainless steel plenum was connected between the pipe and the test section to ensure that the steam entering the test channel had a sharp contraction entrance condition (hydrodynamically developing) to simulate the actual condition of turbine cooling channels. At the end of the test channel, the steam was exhausted into the atmosphere through a 5 cm diameter pipe, equipped with a control valve, a stop valve and a check valve to control the pressure of the steam flowing through the test channel.



1—Steam Generator 2—Plenum 3—Heating device of test channel 4—Test channel

Fig.1.System diagram of steam heat transfer enhancement test platform of turbine blade(vane) internal channel

The sketch of an internally cooled turbine airfoil is shown in Fig.2(a). The internal-cooling passages can be approximately modeled as rectangular channels with a pair of opposite rib-roughened walls, as shown in Fig.2(b). Coolants flow through the ribbed channels to cool the high temperature blade(vane). So the test section is designed as a ribbed rectangular channel in order to simulate the actual configurations of blade(vane) internal passages. Considering the safety and the constrains of materials, the wall temperature of the test section is 473-673K.

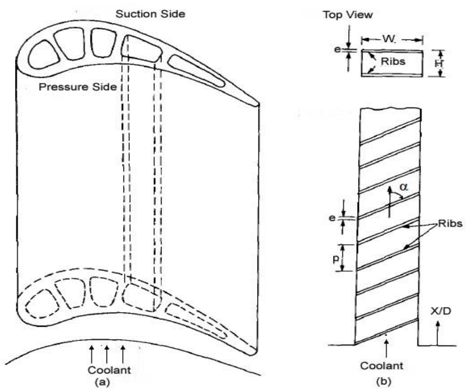
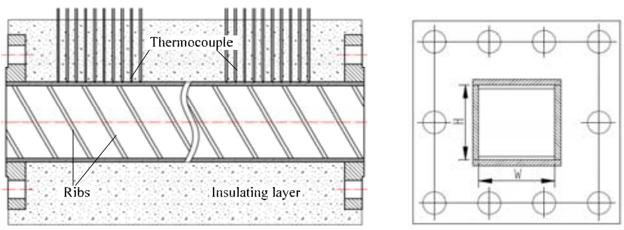


Fig.2.(a) Sketch of an internally-cooled turbine airfoil. (b) Rectangular channel with a pair of opposite rib-roughened walls.

Most of the heating temperatures of the existing test section of similar test platforms are lower than 393K[12-16], even at room temperature. The materials used in the similar platforms are Teflon, resin, heat-resistant glass and wood etc. The wall temperature of the test section is as high as 673K, and the pressure of steam is 0.6Mpa. These materials are not available according to the high temperature of the test section and the high pressure of the steam. The material can be used in high temperature and resistant to pressure is needed. Considering reasons mentioned above, the test section is manufactured by the heat-resistant stainless steel 1Cr18Ni9Ti.



(a) (b)

Fig.3. (a) Structure of the test channel. (b) Cross-section of the test channel

Usually, the stainless steel foil heater was cemented to the inner surface of each plate to provide electric heating to the test channel. This approach is unsuitable for high heat flux situations. The test channel is heated to 473-673K, and the heat flux is about $5.5kWm^{-2}$-$22kWm^{-2}$ under different test conditions. The foil heater is not available because of the high heat flux, in addition, the test channel is welded together by four stainless steel plate, and the foil heaters cannot be cemented to the inner surface of the test channel. After lots of investigations, the low-voltage, high current heating method is used in the test system. The heating device consists of a controller and a transformer. The quantity of heat can be controlled via regulating the current through the test section according to the test conditions. The maximum current of the heating device is 4500A, and the maximum voltage is 7.5V. It is more secure to heat the test section by this way, and the arrangement of the measuring points of temperature is relatively simple. The thermocouple can be welded to the outside surface of the test channel directly.

For the square channels, the research of Han[11] indicates that the best heat transfer performance occurs at 60° ,45° angled ribs, and the Nusselt numbers of 60° angled ribs are slightly higher than that of 45° angled ribs. The study object is a square channel with 60° angled ribs on the opposite walls. The square channel has a cross-section of 40mm×40mm, when the rib pitch(p/e) is 10, the channel blockage(e/D) is 0.048. The test channel is welded by four stainless steel plate, of which the ribbed walls are milling by 5mm thick plates.

The detail of thermocouple locations on the test channel is shown in Fig.4. The test channel has 64E-type thermocouples in strategic locations to measure the local surface temperature. 40 of these thermocouples are placed on the region of $0 \le x/D \le 4.3$, which is the developing region of flow, 20 are placed on the region of $8.1 \le x/D \le 10$. There are 5 thermocouples between the distance of each rib and the thermocouple spacing is 3.8mm on the above two regions. The other 4 thermocouples are placed between the two regions, in 8 rib spacing, and the thermocouple spacing was 19mm. All the thermocouples are placed along the centerline of the ribbed wall. There are no thermocouples on the smooth wall. The test channel is insulated with 50mm thickness aluminum silicate and the heat loss is very small. A Yokogawa MX100 data acquisition system is used for temperature readings and recordings.

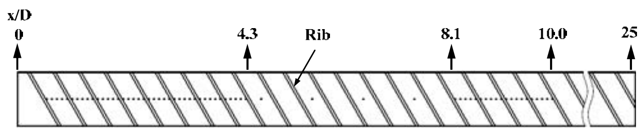


Fig.4. Detailed thermocouple locations

Data reduction

The average heat transfer coefficient is calculated from the average net heat transfer rate per unit surface area from the wall to the cooling steam, the local wall temperature, and the local bulk mean steam temperature as

$$h = \left(q - q_{loss}\right) \big/ \left[A\left(T_w - T_b\right)\right] \qquad (1)$$

Equation (1) is used for the ribbed wall heat transfer coefficient calculations. The average net heat transfer rate is the electric power from the heating device, minus the heat loss to the outside of the test channel. There is a wattful power meter in the heating device, the power transfer to the test channel can be calculated and shown automatically, and the loaded power can be adjusted 0-100% according to the test conditions. The heating device provides a nearly uniform heat flux on each wall of the test channel. The heat loss from the test channel is estimated to be natural convection when there was no steam flow in the test channel. The net heat flux level is varied from about 5.5kWm^{-2} to 22kWm^{-2} depending on the test conditions.

The local wall temperature used in equation (1) is obtained from the output of the thermocouples. The bulk mean steam temperatures of inlet and outlet in the test channel are measured by thermocouples. The local bulk mean steam temperature used in equation (1) is calculated, assuming a linear steam temperature rise along the flow channel. It is checked that the net heat transfer from the test channel to the cooling steam agreed well with the cooling steam enthalpy rise along the test channel. The difference between the two measurements is estimated to be less than 5% for Reynolds numbers greater than 10000. The inlet bulk steam temperature is about 432-463K depending on the test conditions.

The maximum uncertainty in the Nusselt number is estimated to be less than 8% for Reynolds numbers larger than 10000 by using the uncertainties estimation method of Robert J.Moffat[17].

EXPERIMENTAL RESULTS AND DISCUSSION

Experimental results for different Reynolds numbers

Typical results for different Reynolds numbers are shown in Fig.5. In general, the average Nusselt numbers of steam along the center line of the ribbed wall exhibit nearly the same trend, and the average Nusselt numbers increase with increasing Reynolds numbers. It can be seen that at the entrance of the test channel, the average Nusselt numbers are much higher than that of later, and the average Nusselt numbers decrease rapidly until x/D=3. The sharp contraction of the test channel produces the high heat transfer coefficient because the steam is not fully developed in the region. At about 7≤x/D≤10, for the test channel the average heat transfer results are consistent comparatively; at 3≤x/D≤7, the average Nusselt numbers rise slowly along the ribbed wall. These are similar to the results of air from Han's test channel[11].

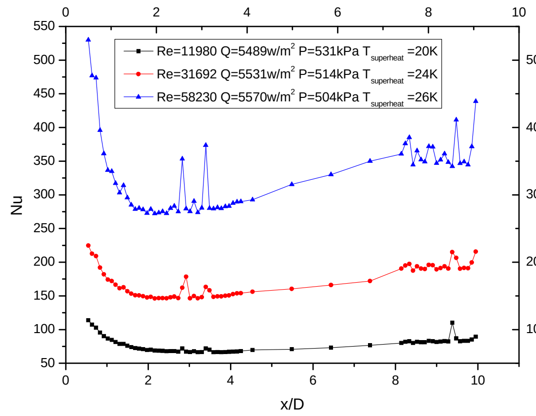


Fig.5. The effect of Reynolds numbers on the centerline of the ribbed wall heat transfer distribution

Experimental results for different superheat of steam

The average heat transfer coefficient of different superheat is shown in Fig.6. The results show that the average Nusselt numbers of steam along the center line of the ribbed wall have the same trend at different steam superheat and the steam superheat has nearly no effects on the heat transfer coefficient at the test conditions. The thermal properties of steam with different pressures and temperatures are nearly the same at the test conditions. Prandtl number is the most important parameter impacting the heat transfer coefficient of steam except Reynolds number. When the steam superheat is 6K(412KPa) ,the Prandtl number is 1.0398 and when the steam superheat is 38K(407KPa), the Prandtl number is 0.994. Prandtl number changes about 5%, the thickness of boundary layer and thermal boundary layer and the thermal resistance are almost the same; the maximum uncertainty in the Nusselt number is estimated to less than 8%, so we cannot see the change of the Nusselt numbers.

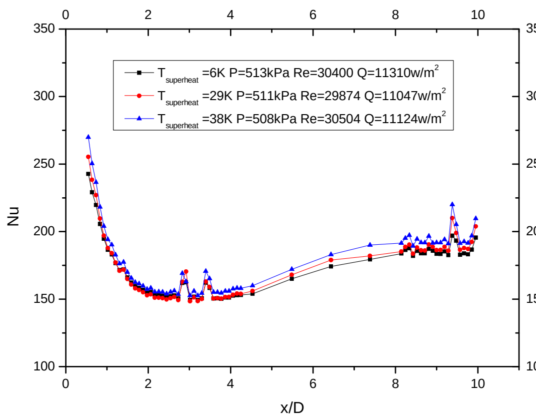


Fig.6. The effect of superheat of steam on the centerline of the ribbed wall heat transfer distribution

Experimental results for different heat flux of heat transfer surface

The average Nusselt numbers of different heat flux of heat transfer surface are shown in Fig.7. The average heat transfer coefficient nearly has no changes with different heat flux of heat transfer surface of the test channel at the test conditions. When heat flux becomes higher, the temperature rise of steam flow is higher too, and the net heat transfer from the test channel to the cooling steam agrees well with the cooling steam enthalpy rise. So increasing the heat flux has nothing to do with heat transfer coefficient.

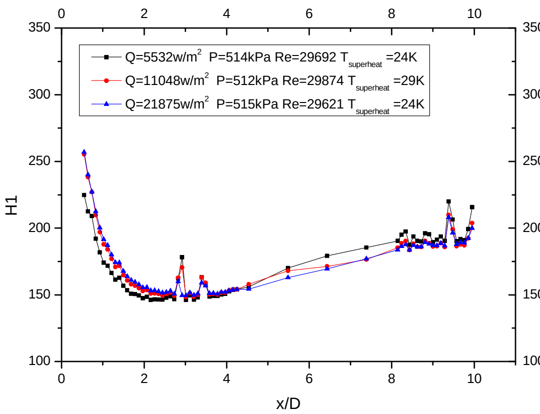


Fig.7. The effect of heat flux of heat transfer surface on the centerline of the ribbed wall heat transfer distribution

Experimental results for different pressures

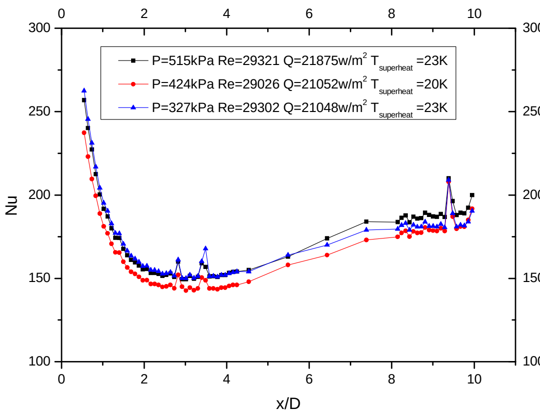


Fig.8. The effect of entry pressures on the centerline of the ribbed wall heat transfer distribution

The distribution of Nusselt numbers for different pressures of steam is shown in Fig.8. At the same superheat, heat flux and Reynolds number, the average Nusselt numbers along the centerline of the ribbed wall at different inlet pressures (226kPa-414kPa) are almost the same. It is similar to the results of different steam superheat. When the inlet pressure is 226kPa, the Prandtl number is 0.999 and when the inlet pressure is 414kPa, the Prandtl number is 1.008, Prandtl number is almost the same at the test conditions, so the Nusselt numbers also have nearly no change with different inlet pressures. In addition, when the total pressure of steam increases, the velocity of steam near the wall also increases, and the thermal boundary layer becomes thinner, so the heat transfer between the ribs can be enhanced theoretically, in fact, the effect is small compare to Prandtl number and Reynolds number.

The heat transfer comparison of steam and air

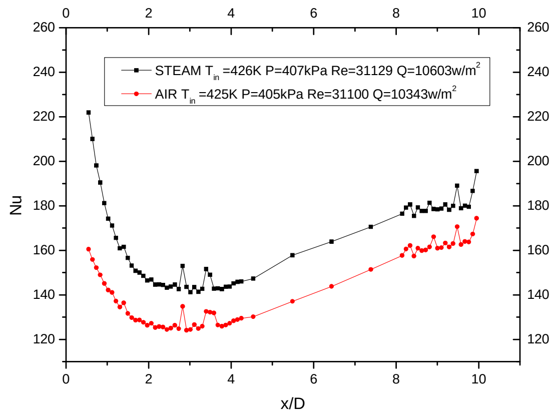


Fig.9. The distribution of Nusselt numbers of the steam and air at the same square channel along the centerline of the ribbed wall.

The experimental results of steam and air at the same test condition are included in Fig.9 for comparison. It can be seen that the local Nusselt numbers of steam are obviously higher than that of air at the same rib angle, blockage ratio and rib pitch. The average Nusselt number is about 15% higher than that of air at the test condition. Otherwise, at the entrance and the fully developed region, the heat transfer coefficient distribution of steam is similar to air. At the same temperature and pressure, the Prandtl number of steam is much larger than air when the conductivity is smaller a little. At the test condition, Prandtl number of steam is 1.06 and Prandtl number of air is 0.705.

$$Nu_{\mathrm{steam}} / Nu_{air} = (\mathrm{Pr}_{\mathrm{steam}} / \mathrm{Pr}_{air})^{0.4}$$

$$Nu_{\mathrm{steam}} / Nu_{air} = (1.06/0.705)^{0.4} = 1.18$$

$$\overline{Nu_{\mathrm{steam}}} / \overline{Nu_{air}} = 163.5/142.2 = 1.15$$

Tt is shown that the steam flow Nusselt numbers everywhere are about 15% higher than those of airflow, and it proves that the Prandtl number is the key parameter impacting the heat transfer between the flow and the wall except Reynolds number, when the conductivity of coolant also has a little effect on the Nusselt numbers. It can be seen that steam is a better coolant than air for those gas turbines used in the combined-cycle.

CONCLUSION

The average Nusselt numbers along the centerline of the ribbed wall increase rapidly with increasing Reynolds numbers in the steam cooled channel. At the region of $x/D \leqslant 3$, the Nusselt numbers decrease rapidly because of the sharp contraction of the test channel. At the fully developed region ($8.1 \le x/D \le 10$), the average Nusselt numbers are constant comparatively when the Reyonld numbers increase from 10000 to 60000.

A detailed analysis has been carried out to show the influence of steam superheat, heat flux of heat transfer surface and inlet pressures. They have nearly no effect on the heat transfer coefficient along the centerline of the ribbed wall. When the steam superheat range from 6K to 38K, the heat flux range from $5.5kWm^{-2}$ to $22kWm^{-2}$, the inlet pressures range from 0.3Mpa-0.6Mpa, the average Nusselt numbers are almost the same. The Prandtl number of steam is the key parameters impacting the heat transfer at these tests except Reyonld number.

The experimental results of steam and air at the same test condition are included in this paper for comparison. The results show that steam is a promising coolant whose average Nusselt number is about 15 percent higher than air because of the effects of high Prandtl number. However, at the entrance and the fully developed region, the heat transfer coefficient distribution of steam is similar to air.

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