

Research Article

Rotational Effects on Heat Transfer for In-Line Jet Impingement Array in a Mid-Span Turbine Channel: An Experimental Study

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Abstract:

This study examined heat transfer for a 3x11 inline impinging jet array within a rotating channel that simulates the turbine blade cooling in mid-span region. The heat transfer on the impingement surface was investigated using Thermochromic Liquid Crystals and a thin-foil heater under steady-state conditions. The influence of impingement distance ($L/D=2, 4, 6$) and Rotation number ($Ro=0.0-0.0083$) was examined at a constant Reynolds number of $Re=10,000$. Results indicate intricate relationships between impingement distance, Coriolis forces, and crossflow. Increased rotation numbers generally diminish heat transfer on both of leading and trailing sides relative to the stationary condition. At a small L/D of 2, crossflow effects predominated over rotational effects, resulting in gradual decrease in downstream region. In contrast to distances $L/D=4$ and 6, the Nusselt number declined rapidly in the upstream region, whereas the influence of rotation lessened in the downstream region.

Keywords: *Impinging jet array, Inline arrangement, Rotating channel, Heat transfer, Coriolis effect, Crossflow, Thermochromic Liquid Crystals (TLC)*

1. Introduction

Increasing the turbine inlet temperature (TIT) is a critical approach in the advancement of contemporary gas turbines to optimize thermal efficiency. This breakthrough improves efficiency and decreases fuel consumption while also being essential for reducing carbon dioxide (CO_2) emissions and facilitating the future shift to low-carbon fuels like hydrogen. Nonetheless, these increased operating temperatures immediately affect components subjected to the hot gas stream, particularly the combustion liner and turbine blades. Thus, efficient cooling methods are crucial for keeping material temperatures within safe operational thresholds, thereby prolonging component longevity and averting thermal breakdown. Numerous internal cooling methods are extensively utilized in both fixed and moving blades, such as pin fins, serpentine channels with rib turbulators, film cooling, and impinging jets [1-3].

Jet impingement is often considered to be the most effective techniques for internal cooling. It generates extremely high heat transfer rate in the region where the jet impinges directly. Because of this major advantage, jet impingement is often used for parts that require substantial cooling, such the leading edge of a turbine blade, which is the first part to be impacted by the hot gas flow. Additionally, it is used in the mid-span area, which includes the pressure and suction sides, to improve local heat transfer in places with very high thermal loads.

A significant body of prior research has investigated heat transfer from jet rows and jet arrays under stationary conditions, simulating their application in stator vanes. Critical geometric and flow parameters affecting thermal performance have been studied extensively, including the jet impingement distance (L/D) and arrangement

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configuration (e.g., in-line or staggered). A primary factor identified in these studies is the detrimental effect of crossflow. This phenomenon occurs when spent air from upstream jets accumulates and obstructs the downstream jets' path. It is well established that this crossflow adversely affects performance by deflecting the jets, reducing their penetration, and, consequently, decreasing heat transfer in the downstream regions [1].

For applications in rotating rotor blades, the flow and heat transfer physics become significantly more complex. The rotation introduces potent inertial forces, namely the Coriolis force and the centrifugal force. These forces have a profound impact on the secondary flow structure, turbulence intensity, and boundary layer stability [2, 3]. A key dimensionless parameter describing this effect is the Rotation Number (Ro), which represents the ratio of Coriolis to inertial forces. Studying the effect of Ro number is therefore essential for understanding and designing cooling systems under realistic operating conditions.

Numerous studies have examined the influence of rotation on jet impingement, although the results differ based on geometry and conditions. Elston and Wright [4] investigated the effects of impingement on a curved leading edge and found that rotation enhanced heat transfer by promoting mixing within the cavity, particularly in the absence of crossflow. Conversely, Chang and Yu [5] found that in a rotating trapezoidal channel, the Coriolis force resulted in an uneven mass flow distribution from the individual jet orifices. A thorough literature review reveals that most research has focused on a single row of jets for cooling the blade in leading-edge region, which has narrow, curved geometry.

Due to the trend of rising TIT, thermal loads are no longer restricted to the leading edge. The mid-span area encompassing both suction and pressure sides of the blade, also requires high-performance cooling techniques such as jet impingement. However, very little research has been published on the application of impinging jet arrays for cooling the mid-span region of a rotating turbine blade. Yang et al. [6] studied impingement in a two-pass channel, observing that rotation diminished heat transfer on both the leading and trailing impingement sides. They attributed this reduction to jet deflection caused by Coriolis forces. Nontula et al. [7] found that the heat transfer of jet impingement for a row jet on leading and trailing sides had different effects. More recently, Singh and Ekkad [8] investigated the detailed heat transfer measurements for inline array impinging jets under rotation, studying the heat transfer improvement using dimples on the jet impingement surface for a specific jet impingement distance and rotation speed. Therefore, a critical research gap still exists in the fundamental understanding of the heat transfer behavior of array jets under different rotation speeds and jet impingement distances, which is essential for the design of modern, high-performance gas turbines.

This experimental study is designed to simulate the jet impingement cooling in the mid-span region of the rotating blade on suction and pressure side. The heat transfer on the impingement surface was investigated using Thermochromic Liquid Crystals and a thin-foil heater under steady-state conditions. The test model consists of a 3×11 array of orifice nozzles ($D = 5$ mm) arranged in an in-line configuration with a pitch-to-diameter ratio of $S/D = 4$. The purpose of this study is to reveal the influences of jet impingement distance ($L/D = 2, 4$, and 6) and rotation speed (100, 200, 300, and 400 rpm) on heat transfer characteristics compared to the stationary case (0 rpm), at a Reynolds number (Re) of 10,000.

2. Experimental Setup

2.1 Experimental model

The experimental model was simulated using impingement cooling at the mid-span area of the turbine blade. The air jets from the array of orifices on the jet chamber impinge normally on the internal wall and flow in the cooling channel between the jet chamber wall and the internal wall to the outlet at the blade tip. The impingement wall was divided into the leading side wall (suction side of the rotor blade) and the trailing side wall (pressure side of the rotor blade). Fig. 1 shows the details of the array of orifice jets used in this study. The orifice wall was made of a 3 mm-thick aluminum plate. Each orifice hole was circular, with a diameter $D = 5$ mm. The orifice plate was arranged in an inline configuration with 3×11 holes, with a pitch distance $S/D = 4$ in both the streamwise and spanwise directions. The channel width was $W/D = 12$. Radial distance (average radius) from the rotation axis to the center of the orifice array was 315 mm. In this study, the jet impingement distance (distance from orifice to impingement surface) was varied at $L/D = 2, 4$, and 6 . The pitch distance and jet impingement distance are selected in this study based on the literature

review in previous research [9]. The upstream of the channel was closed with a wall, and the air jet flow from orifices after impinging on the wall will be forced to flow radially outward to the downstream outlet.

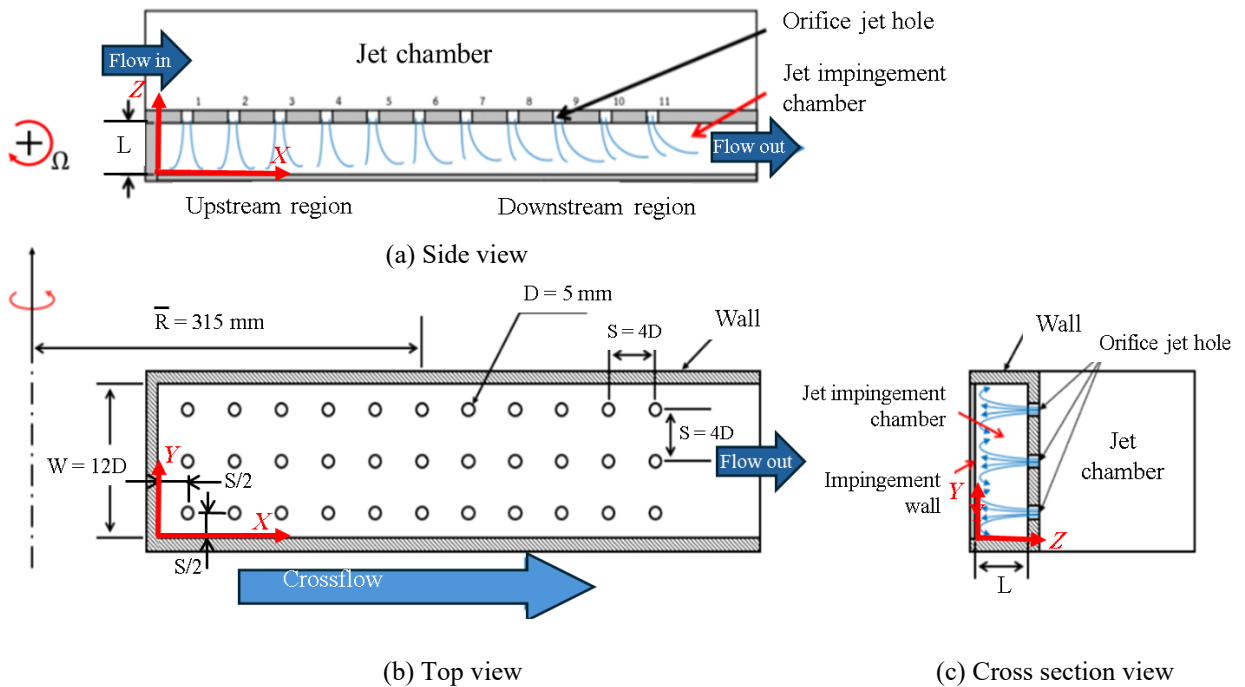


Fig. 1. Details of orifice arrangement in the rotating channel.

2.2 Experimental Apparatus

A schematic of the complete experimental test rig designed to investigate the heat transfer of array impinging jets in rotating channel is depicted in Fig. 2. The apparatus is constructed on a robust steel frame and integrates three parts: the air supply, the rotation system, and the test section. The air supply is driven by a high-pressure blower, with the air flow rate precisely controlled by an inverter. After exiting the blower, the main airflow passes through an orifice flow meter for accurate measurement. The air is then conditioned in a temperature-controlled chamber (heater) to ensure a stable, uniform inlet temperature before it enters the rotating components. The conditioned air flows into a stationary rotary union which seals against a specific hollow rotating shaft, facilitating the passage of air from the stationary supply to the rotating test section. The rotation system is responsible for inducing the Coriolis and centrifugal forces. The test section, which contains the jet chamber and the impingement flow channel, is mounted on a rotating arm, while a corresponding counterweight is fixed on the opposite side to ensure the assembly is dynamically balanced and to minimize vibration. This entire rotating arm assembly is connected to the hollow rotating shaft, which is supported by two high-speed bearings. The shaft is driven by an electric motor via a V-belt, and its rotational speed is governed by an inverter, with the precise rotation speed continuously monitored by a digital encoder. To supply thermal energy, a stationary DC power supply delivers a constant electrical current to a slip ring assembly. This slip ring features stationary carbon brushes that maintain continuous contact with the conductive copper contacts on the rotating shaft, thereby energizing the heater within the test section to produce a constant heat flux.

The details of the test section are illustrated in Fig. 3. The assembly consists of a jet chamber and an impingement flow channel, which are constructed with acrylic side walls. Conditioned air enters the jet chamber and is then forced through an array of orifice holes, forming the impinging jets. The impingement wall comprised of 0.03 mm stainless foil (SUS304) as impingement surface affixed to an acrylic plate with a central window for heat transfer measurement. The foil was connected to copper bars at the lateral edges. Electrical current was supplied from power supply passed the slip ring and directly through the foil via copper bars for generating constant heat flux on the impingement surface. The array of jets then impinges on the heated surface for cooling.

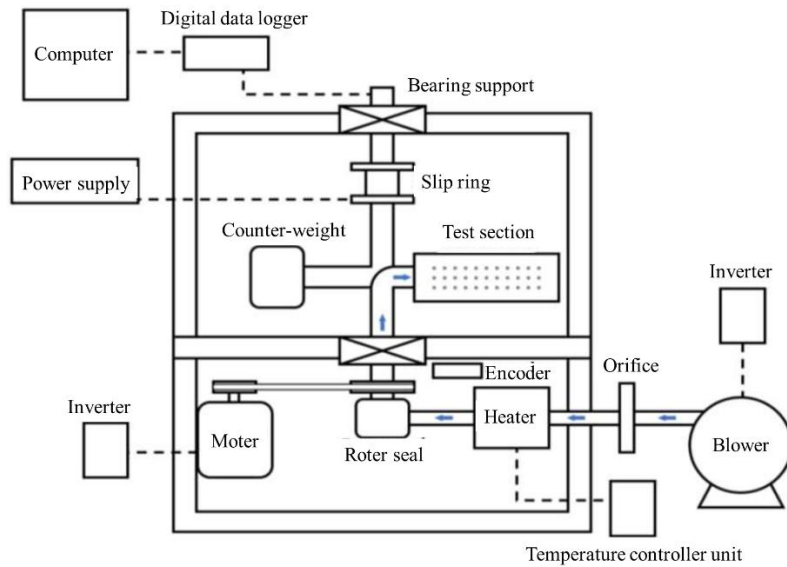


Fig. 2. Diagram of experimental apparatus.

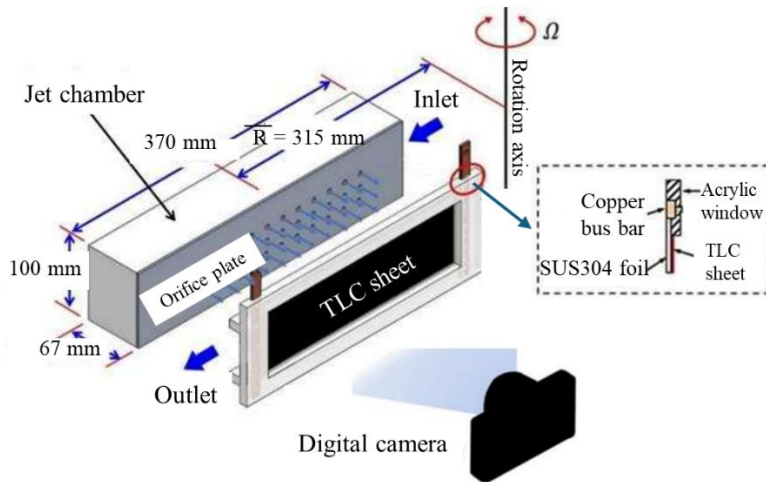


Fig. 3. Details of the test section.

For temperature measurement, the rear side of the stainless-steel foil is attached with a Thermochromic Liquid Crystal (TLC) sheet. This TLC sheet is applied in the temperature range of 30.0°C to 35.0°C to display a color changing from red to green to blue. A digital camera is fixed relative to the impingement wall and rotates with the test section, continuously recording the color patterns displayed on the TLC sheet. During each experimental run, the ambient and inlet air temperatures are monitored and recorded by a datalogger. To examine the jet impingement on the leading and trailing side of a rotating blade, heat transfer measurements are conducted by reversing the direction of the rotation arm.

2.3 Image Processing and Temperature Conversion

The recorded images are subsequently processed to map the local surface temperature distribution according to details in previous study [9]. An in-situ calibration was performed to establish the precise relationship between the color and the temperature which was recorded by a datalogger connected to T-type thermocouples placed between TLC sheet and adjustable temperature surface. During the experiment, once the temperature reached a steady state, the digital camera recorded the color patterns displayed by the TLCs. This image data was then post-processed by converting the RGB (Red, Green, Blue) pixel values into their corresponding Hue values. Using the established calibration curve,

these Hue values were then converted directly into a high-resolution temperature map. This detailed temperature data was subsequently used for calculating the local convective heat transfer coefficients and Nusselt numbers across the impingement surface.

2.4 Data Reduction and Uncertainty

The electrical power supplied to the stainless-steel foil was used to calculate the total generated heat flux $\dot{q}_{gen}$ from the measured voltage V and current I applied over the area of heat transfer surface A_s , as shown in Eq. (1).

$$\dot{q}_{gen} = \frac{IV}{A_s} \quad (1)$$

For this study, the generated heat flux on impingement surface was fixed at 1050 W/m² for all cases. The net convective heat flux $\dot{q}_{conv}$ was determined by subtracting the heat losses from the generated heat flux. These losses included free convection $\dot{q}_{loss,conv}$ and radiation $\dot{q}_{loss,rad}$ from the TLC sheet to the ambient surroundings, as well as conduction losses $\dot{q}_{loss,cond}$ on the impingement surface. The local heat transfer coefficient h was calculated with the local surface temperature T_s obtained from the TLC sheet images and a reference air temperature T_a , as defined in Eq. (2).

$$h = \frac{\dot{q}_{conv} - \dot{q}_{loss,conv} - \dot{q}_{loss,rad} - \dot{q}_{loss,cond}}{T_s - T_a} \quad (2)$$

The air temperature T_a was defined as the average of the air temperatures measured at inlet T_{in} and the outlet T_{out} of test section

$$T_a = \frac{T_{in} + T_{out}}{2} \quad (3)$$

The local Nusselt number Nu was then calculated using the thermal conductivity of air k_a and the jet hole diameter D :

$$Nu = \frac{hD}{k_a} \quad (4)$$

The Reynolds number was evaluated using the center velocity at the jet orifice exit. U_j , the jet hole diameter D , and the kinematic viscosity of air ν :

$$Re = \frac{U_j D}{\nu} \quad (5)$$

The Rotation number Ro was calculated using the angular velocity of rotation Ω :

$$Ro = \frac{\Omega D}{U_j} \quad (6)$$

Table 1 describes the experimental conditions. Experiments were performed at a Reynolds number of 10,000, with impingement distances (L/D) of 2, 4, and 6, and Rotation numbers (Ro) ranging from 0.0 to 0.0083. The inlet air temperature was maintained at $26.0 \pm 0.1^\circ\text{C}$, while the laboratory ambient temperature was regulated at $25.0 \pm 0.5^\circ\text{C}$. Following the method proposed by Moffat [10], the experimental uncertainty analysis yielded maximum values of 4.2% for the Reynolds number, 3.8% for the Nusselt number, and 2.9% for the Rotation number.

Table 1: Experimental condition

Jet hole diameter (D)	5 mm
Impingement distance (L/D)	2, 4 and 6
Rotation number (Ro)	0, 0.0017, 0.0033, 0.0050, 0.0066 and 0.0083
Rotational speed	0, 100, 200, 300, 400 and 500 rpm
Reynolds number (Re)	10,000

3. Experimental Results and Discussion

In the stationary condition, the heat transfer characteristics are strongly influenced by crossflow which the accumulation of spent air from upstream impinging jets. When rotation is introduced, the flow field is further subjected to centrifugal and Coriolis forces, as illustrated in Fig. 4. This investigation simulates jet impingement cooling for both the pressure and suction sides of a rotating turbine blade. Two distinct flow orientations are modeled: (1) a co-rotating configuration, representing leading-side impingement, where the jets discharge in the same direction as the blade's rotation; and (2) a counter-rotating configuration, representing trailing-side impingement, where the jets discharge in the direction opposite to the blade's rotation. As shown in Fig. 4, centrifugal force acts radially outward and increases in magnitude with increasing distance from the axis of rotation. The Coriolis force is more complex, as it acts on both the jet flow and crossflow. As this crossflow velocity increases along the channel path toward the outlet, the magnitude of the Coriolis force acting upon it also intensifies. Consequently, the rotational forces alter jet flow and the crossflow. Because the jet flow in leading and trailing sides experience Coriolis forces in different directions, this interaction ultimately affects the heat transfer characteristics.

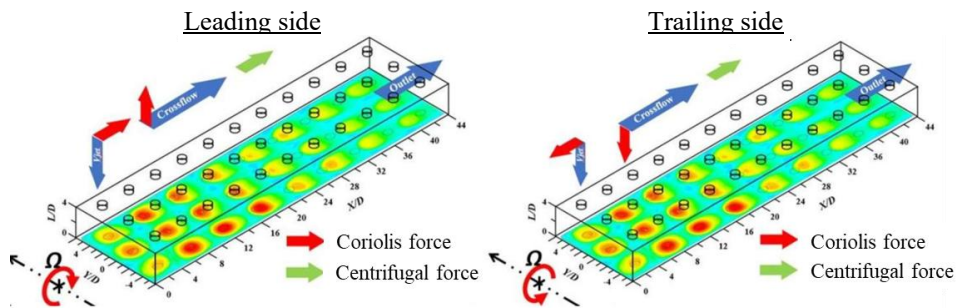


Fig. 4. Coriolis force and centrifugal forces on crossflow and jet flow in impingement channel during rotation, comparing the leading (left) and trailing (right) sidewalls.

4. Results and Discussion

4.1 Nusselt Number Contour

The Nusselt number contour is compared at different Ro numbers at distinct impingement distances (L/D) of 2, 4, and 6 as shown in Figs. 5, 6, and 7, respectively. The black circle shows the location of each orifice hole.

For a stationary channel at impingement distance $L/D = 2$, the contour in the impingement region is mostly in circular shape for the first two columns of jets. The Nusselt number then becomes on a semicircular shape due to the crossflow generated by the accumulating jet flow. As it goes downstream, The Nusselt number decreases and extends more downstream. In the upstream impingement region ($0 \leq X/D \leq 20$), the Nusselt number reduces further as the impingement distance L/D increases to 4 and 6, but in the downstream impingement region ($24 \leq X/D \leq 44$), it decreases rapidly. This is due to the lower momentum of jet impingement, and most downstream jets cannot impinge on the wall due to the stronger crossflow.

Figure 5 illustrates the Nusselt number contour for the case of $L/D = 2$. In the upstream region ($0 \leq X/D \leq 20$), the Nusselt number is high in the impingement region for each jet and subsequently decreases as going downstream. In the downstream region ($24 \leq X/D \leq 44$), the local Nusselt number gradually diminishes. This degradation is attributed to the influence of the accumulating channel crossflow and the increase of crossflow velocity, which deflects more the downstream jets. This reduces their impingement momentum on the wall and thus diminishes their heat transfer. A rise in the Ro number causes a substantial reduction in Nusselt number. Nusselt number is clearly greater on the leading side than on the trailing side. This phenomenon is induced by the combination of strong crossflow for $L/D = 2$ and rotation effect acting on the crossflow.

Figure 6 presents the results for the case of $L/D = 4$. The significant reduction in Nusselt number observed in the downstream region, compared to the $L/D = 2$, is attributed to the crossflow-induced dissipation of jet momentum over the extended impingement distance. For the case of small Ro numbers ($Ro = 0.0017$ and 0.0033), Nusselt number is marginally smaller on leading side than the trailing side in upstream region. This may be due to on the trailing side,

the direction of Coriolis force to the upstream direction. And the crossflow influenced by the Coriolis force upon attachment to the impingement wall. However, the Nusselt number is measurably lower as going downstream because of the crossflow. The degradation of heat transfer at higher Ro numbers is due to the intensified centrifugal force acting on the crossflow, which suppresses the Nusselt number on both sides.

Figure 7 shows the results for case of $L/D = 6$. The trends of Nusselt number are again qualitatively similar to the $L/D = 4$ case. The local Nusselt number is high in the upstream region. And the Nusselt number decreases rapidly downstream because the jet does not impinge on the wall. The weakest jet impingement momentum results in a diminishing Nusselt number. It should be noted that the case of $L/D = 6$ at $Ro = 0.0083$ does not show due to the high vibration of the experimental apparatus.

4.2 Spanwise-Average Nusselt Number

Figures 8, 9, and 10 illustrate the variation of the spanwise-average Nusselt number along the streamwise direction for impingement distances $L/D = 2, 4$, and 6 , respectively. The average Nusselt number are obtained by averaging the local Nusselt number from the contour plots along the spanwise direction presented in Figs. 5, 6, and 7. The dashed line shows the centerline of the orifice of each column.

A common trend is evident for all jet impingement distances. The average Nusselt number is large in the upstream impingement region ($0 \leq X/D \leq 20$) and gradually drops in the downstream impingement region ($20 \leq X/D \leq 44$), as indicated by the contour Nusselt number. This degradation is the characteristic effect of the accumulating crossflow, which deflects downstream jets and reduces their impingement momentum.

The results for case of $L/D = 2$ are shown in Fig. 8. The average Nusselt number peaks are nearly identical in the upstream region and decrease gradually as going to downstream region. Despite a strong crossflow, the jet in the downstream region remains to impinge on the wall due to the significant impingement momentum of its potential core. The average Nusselt number over the wall decreases with increasing Ro number for the leading and trailing side. The average Nusselt number is nearly the same on both side walls at each Ro number in upstream region. On the other hand, in the downstream region, a high Ro number is associated with an average Nusselt number that is somewhat higher on the trailing side. One possible explanation is that the crossflow is subject to a different direction of Coriolis force. The crossflow on the leading side tends to separate from the impingement wall, increasing the turbulence intensity of jet flow before impingement.

At the greater impingement distance of $L/D = 4$ (as shown in Fig. 9), the average Nusselt number follows the same pattern as the $L/D = 2$ case. It diminishes with increasing the Ro number. The crossflow significantly reduces effect on the Nusselt number in the downstream region. In the downstream region, the sensitivity of the Nusselt number to the Ro number was markedly higher on the trailing surface than on the leading surface. However, the trailing side exhibited lower overall heat transfer, suggesting that the detrimental effects of Coriolis force on accumulating crossflow.

For the largest impingement distance ($L/D = 6$; Fig. 10), the Nusselt number peaks start to decrease in the upstream region due to the low momentum of jet impingement and then decrease rapidly as going downstream. The effect of rotation on the Nusselt number is evident in the upstream region. In the downstream region, the crossflow effect is stronger than that of rotation, particularly on the leading side.

4.3 Area-Average Nusselt Number

Figure 11 compares the area-average Nusselt numbers on the heat transfer surface, evaluated from averaging the local Nusselt numbers overall the area from the contour plots shown in Figs. 8, 9, and 10. The average Nusselt numbers at different Ro numbers are compared on the leading and trailing side relative to the stationary case at impingement distances $L/D = 2, 4$, and 6 . A consistent trend emerged, showing that As the Ro number increases, the average Nusselt values for the leading and following surfaces both reduce. The average Nusselt number is consistently lower when impingement occurs on the trailing side whereas it is always greater when impingement occurs on the leading side. Although the area-average Nusselt number is higher for small impingement distances ($L/D=2$) compared to larger ones ($L/D=4$ and 6), the results demonstrate a strong sensitivity to rotation. Elevating the Ro number resulted in a substantial suppression of the average Nusselt number across both the leading and trailing sides.

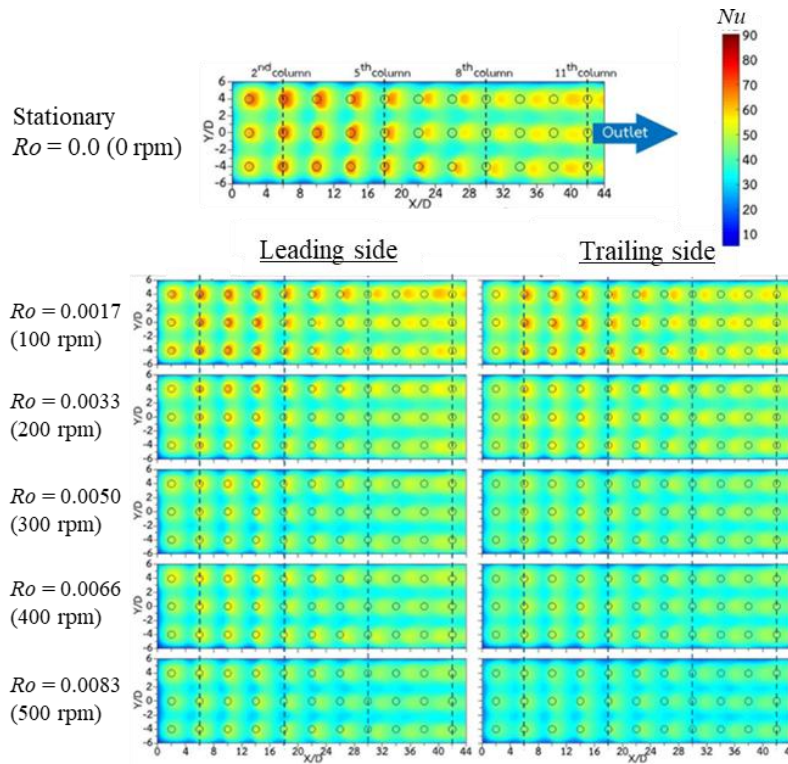


Fig. 5. Effect of Rotation number (Ro) on Nusselt number distribution at $L/D = 2$, comparing the leading (left) and trailing (right) sides.

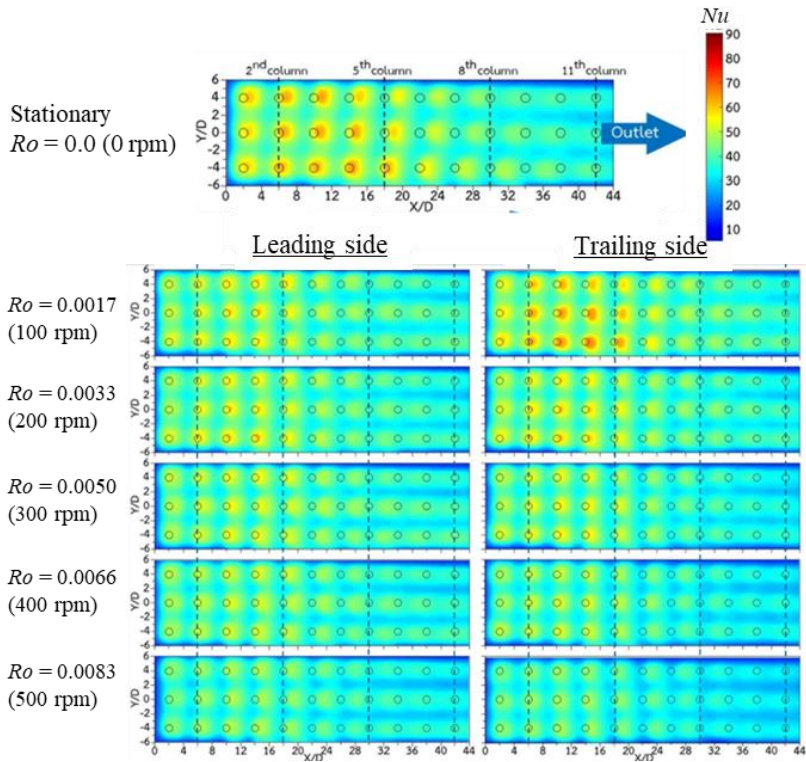


Fig. 6. Effect of Rotation number (Ro) on Nusselt number distribution at $L/D = 4$, comparing the leading (left) and trailing (right) sides.

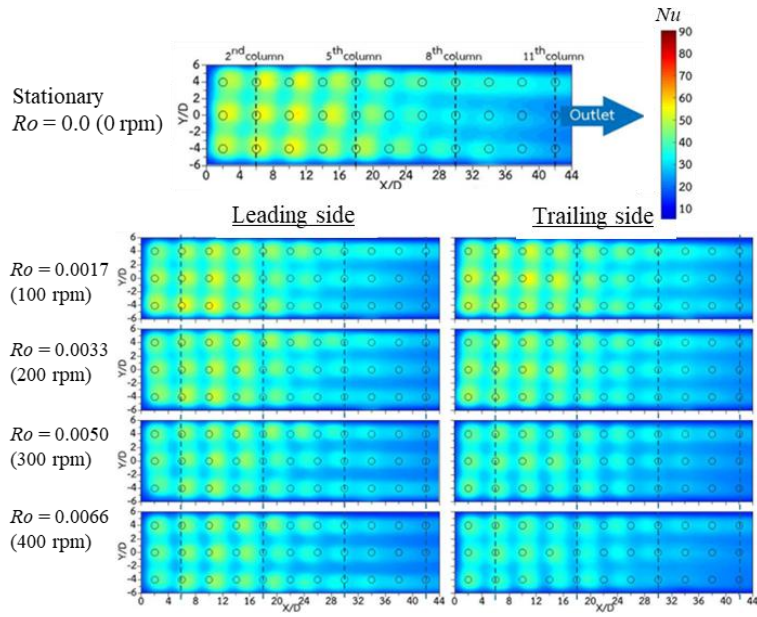


Fig. 7. Effect of Rotation number (Ro) on Nusselt number distribution at $L/D = 6$, comparing the leading (left) and trailing (right) sides.

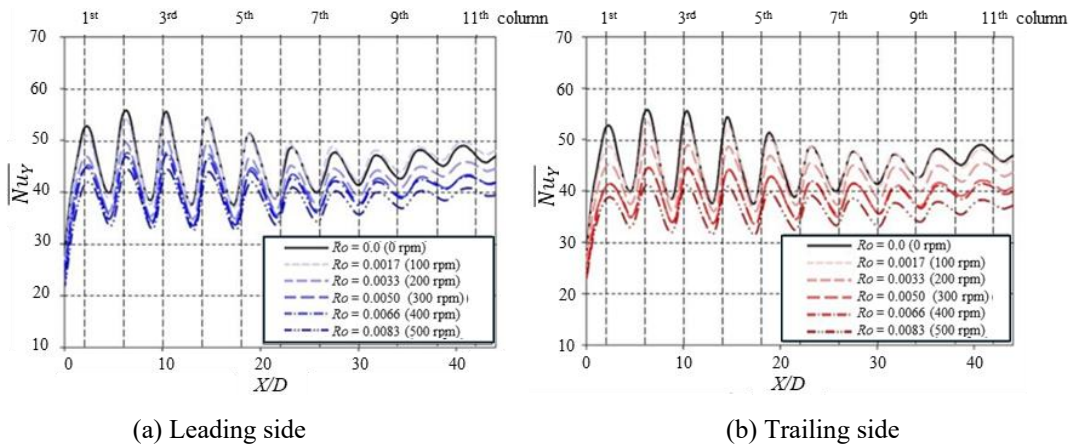


Fig. 8. Spanwise-average Nusselt number distribution along the X/D at $L/D = 2$ for various Rotation numbers (Ro).

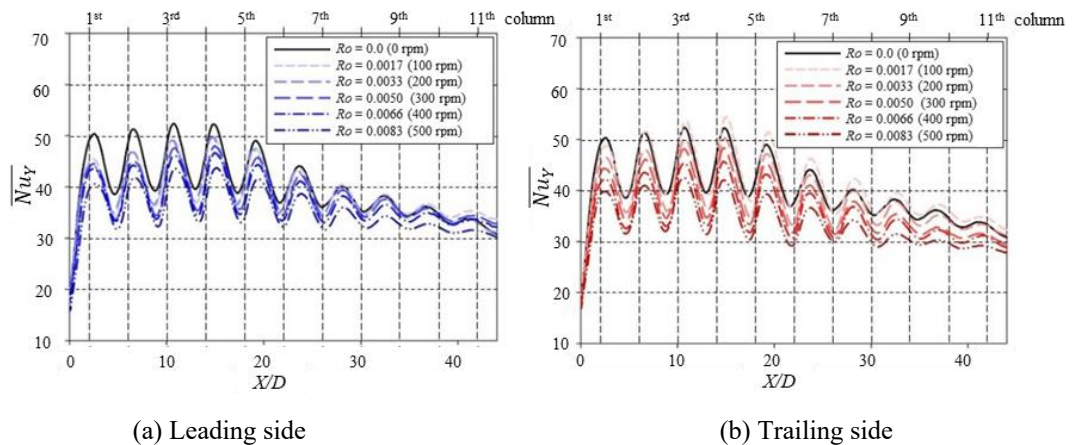


Fig. 9. Spanwise-average Nusselt number distribution along the X/D at $L/D = 4$ for various Rotation numbers (Ro).

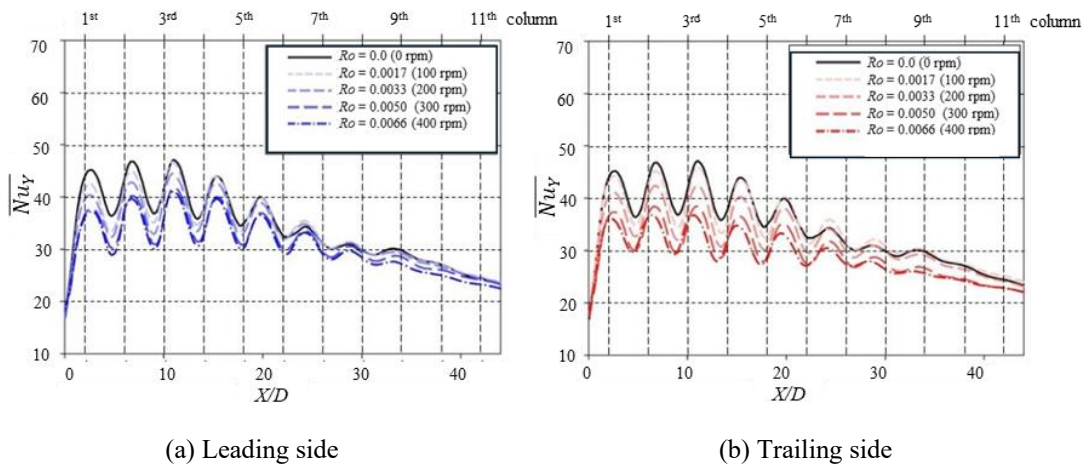


Fig. 10. Spanwise-average Nusselt number distribution along the X/D at $L/D = 6$ for various Rotation numbers (Ro).

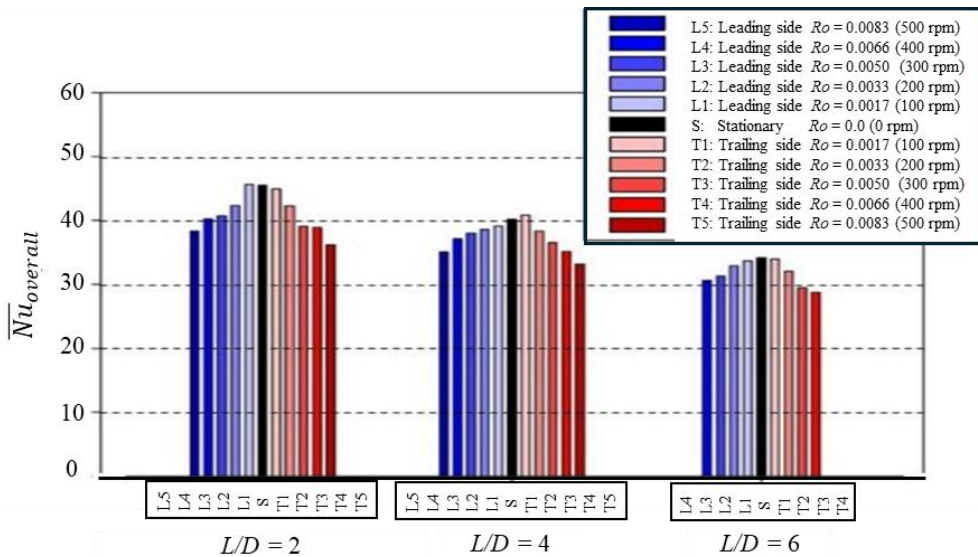


Fig. 11. Effect of Rotation number (Ro) on the area-average Nusselt number for leading and trailing side impingement at $L/D = 2, 4$, and 6

5. Conclusion

This experimental study investigated the heat transfer characteristics of an array impinging jets with inline arrangement in a rotating channel, simulating mid-span cooling of a rotating turbine blade. The effects of impingement distance ($L/D = 2, 4$, and 6) and Rotation number ($Ro = 0.0$ to 0.0083) were examined at a constant $Re = 10,000$. The primary conclusions are as follows:

1. The influence of rotation on heat transfer was found to depend strongly on the impingement distance. For the shortest distance ($L/D = 2$), The Nusselt number on the leading side of impingement was consistently greater than that on the trailing side. Conversely, at larger distances ($L/D = 4$ and 6), the trailing side impingement exhibited slightly higher heat transfer in the upstream region, suggesting a different interaction mechanism of Coriolis forces between jet and crossflow.
2. In comparison to the stationary condition, the Nusselt number on the leading and trailing side impingement decreased as the Rotation number (Ro) increased for all impingement distances L/D .

3. A critical design trade-off was identified. While the $L/D = 2$ configuration provided the highest area-averaged heat transfer, it was also the most sensitive to rotation, showing a significant and detrimental reduction in Nusselt number on both surfaces as Ro number increased.
4. The crossflow performed a key impact. It caused a progressive degradation of Nusselt number in the downstream direction, particularly at larger $L/D=4$ and 6, where the jet momentum was lower than in the case $L/D = 2$

To achieve greater compatibility and effectiveness in advanced gas turbine cooling, future work must concurrently pursue enhanced heat transfer and improved thermal uniformity. This dual objective necessitates investigating the synergistic effect of combining heat transfer enhancement techniques, such as integrating rib-walls and dimple walls. Crucially, optimizing the orifice design and placement is required to ensure a uniform Nusselt number distribution across the impingement surface, which is vital for mitigating localized thermal stresses and extending blade life.

Nomenclature

A_s	Surface area for heat transfer, m^2
D	Jet orifice diameter, m
h	Local heat transfer coefficient, $W/m^2 \cdot K$
I	Electrical current, A
k_a	Thermal conductivity of air, $W/m \cdot K$
L	Jet impingement distance, m
Nu	Local Nusselt number
$\overline{Nu}_y$	Spanwise average Nusselt number
$\overline{Nu}_{xy}$	Overall average Nusselt number
$\dot{q}_{gen}$	Generated heat flux, W/m^2
$\dot{q}_{loss,cond}$	Heat flux loss from conduction, W/m^2
$\dot{q}_{loss,conv}$	Heat flux loss from convection, W/m^2
$\dot{q}_{loss,rad}$	Heat loss from radiation, W/m^2
Re	Jet Reynolds number
Ro	Rotation number
$\bar{R}$	Mean rotating radius, m
S	Pitch distance between orifice center, m
T_a	Average air temperature between test section at inlet and outlet, $^{\circ}C$
T_i	Air temperature at test section inlet, $^{\circ}C$
T_o	Air temperature at test section outlet, $^{\circ}C$
T_s	Impingement surface temperature, $^{\circ}C$
U_j	Jet velocity, m/s
V	Electrical voltage, V
W	Width of impingement channel, m
X	Streamwise coordinate
Y	Spanwise coordinate
Z	Coordinate in direction normal to impingement wall

Greek symbols

ν	Air kinematic viscosity, m^2/s
Ω	Angular velocity, rad/s

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