



Effect of Discrete Ribs on Heat Transfer and Friction Inside Narrow Rectangular Cross Section Cooling Passage (AR=1:5) of Gas Turbine Blade

Karthik Krishnaswamy, Srikanth Salyan

Abstract: The performance of a gas turbine during the service life can be enhanced by cooling the turbine blades efficiently. The objective of this study is to achieve high thermohydraulic performance (THP) inside a cooling passage of a turbine blade having aspect ratio (AR) 1:5 by using discrete W and V-shaped ribs. Hydraulic diameter (D_h) of the cooling passage is 50 mm. Ribs are positioned facing downstream with angle-of-attack (α) of 30° and 45° for discrete W-ribs and discrete V-ribs respectively. The rib profiles with rib height to hydraulic diameter ratio (e/D_h) or blockage ratio 0.06 and pitch (P) 36 mm are tested for Reynolds number (Re) range 30000-75000. Analysis reveals that, area averaged Nusselt numbers of the rib profiles are comparable, with maximum difference of 6% at Re 30000, which is within the limits of uncertainty. Variation of local heat transfer coefficients along the stream exhibited a saw tooth profile, with discrete W-ribs exhibiting higher variations. Along spanwise direction, discrete V-ribs showed larger variations. Maximum variation in local heat transfer coefficients is estimated to be 25%. For experimented Re range, friction loss for discrete W-ribs is higher than discrete V-ribs. Rib profiles exhibited superior heat transfer capabilities. The best Nu/Nu_0 achieved for discrete V-ribs is 3.4 and discrete W-ribs is 3.6. In view of superior heat transfer capabilities, ribs can be deployed in cooling passages near the leading edge, where the temperatures are very high. The best THP₀ achieved is 3.2 for discrete V-ribs and 3 for discrete W-ribs at Re 30000. The ribs can also enhance the power-to-weight ratio as they can produce high thermohydraulic performances for low blockage ratios.

Keywords: Reynolds number, Discrete W and V-ribs, Nusselt number ratio, Heat transfer coefficient, Friction factor ratio, Thermohydraulic Performance

1. INTRODUCTION

Gas turbine engine performance improves with turbine inlet temperatures (TIT). In general, the operating temperatures of current generation gas turbines are very high. In many cases, temperatures are close to melting point, and therefore increase in TIT is possible only by cooling the turbine blades efficiently. Turbine blade surfaces are cooled internally and externally by extracting air from the compressor.

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Drawing out excess air from the compressor for cooling will reduce the power output and hence the cooling process is required to be optimized to achieve maximum heat transfer. Inside a cooling passage, ribs or so called turbulators are positioned on the inner wall surfaces to improve the heat transfer. As the coolant flows over the ribs, turbulence is generated which improves the heat transfer. The magnitude of turbulence is dependent on the physical parameters of the rib such as shape, height, length, orientation, pitch, and number of walls fixed with ribs. The research output obtained by various researchers are discussed in detail further. Han et al. [1] performed experimental studies with nine rib profiles and established that, friction was highest for inverted V-ribs and heat transfer augmentation was highest for in-line V-ribs. Lau et al. [2] assessed the rib performances in a channel of aspect ratio 1:1 and concluded that, heat transfer enhancement was greatest for V-ribs and friction was greatest for inverted V-ribs. It was also concluded that, increase in pitch reduced the heat transfer improvement and friction loss. Han and Zhang [3] studied the performance of different rib profiles and presented that, for Re range 15000-90000 in a cooling channel having aspect ratio 1:1, broken ribs exhibited best performance and ribs oriented at 60° showed better heat transfer capabilities. Ekkad and Han [4] performed experiments with different ribs in square cross section channel having two passes. The authors presented that, for Re range 6000-60000, V-broken ribs produced better enhancements in first pass and the second pass enhancements were 200-300% greater than first pass view intensification of secondary flows by 180° turn. Tzeng and Mao [5] performed experimental studies to comprehend flow and thermal parameters for staggered half V-ribs in four pass serpentine channel for Re 20000 and 40000. They indicated that, effects of ribs on heat extraction were present in all passes. Besides, they also reported that, coolant when pressurized performed better. Al-Hadhrani et al. [6] using a channel of aspect ratio 2:1 tested different V-shaped ribs ($P/e=10$) and concluded that, augmentation was highest for downstream facing parallel V-shaped ribs oriented at 45° to coolant flow. Wright et al. [7] experimented different ribs ($P/e=10$) in a cooling channel (AR=4). They reported that, for Re up to 40000, W-shaped ribs produced high enhancement and friction whereas discrete V and W-ribs produced best thermal performance.



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Gao and Sunden [8] assessed the flow patterns of six rib profiles, which included V and inverted V-ribs in a 1:8 aspect ratio channel. Results for Re range 1000-5800 revealed that, V-ribs facing upstream resulted in highest heat transfer augmentation. Fu et al. [9] assessed the capabilities of 45° angled ribs in cooling passages having aspect ratios 1:4 and 1:2. The authors noticed that, Nusselt numbers of ribbed wall under stationary condition were 2.5-3 times greater than that predicted using Dittus-Boelter correlation for the Re ranging from 5000-40000. Akhanda M.A.R et al. [10] conducted experiments in a square duct for Re ranging from 46000 to 52000 with square ribs and concluded that, heat transfer and pressure drop due to ribs were 15% and 6% greater than smooth wall duct. Maurer and Wolfersdorf [11] by utilizing thermochromic liquid crystal technique analyzed the performances of V-ribs in a channel ($AR=2:1$) and established that, for Re between 95000-500000 and $P/e=10$, ribs with lesser height fixed on one wall delivered the highest performances. Gupta et al. [12] concluded that, amongst orthogonal, saw tooth and 60° V-broken ribs, augmentation was highest for V-broken ribs for Re range 10000 to 30000. Sriharsha et al. [13] tested orthogonal ribs and 60° broken ribs for Re up to 30000 by using a square cross section channel. Authors brought out that, broken V-ribs with $P/e=10$ generated highest augmentation. Alkhamis et al. [14] evaluated angled and V-shaped ribs for their performances up to Re 400000 and confirmed that, 45° V-shaped ribs produced highest performance. Baraskar et al. [15] examined the performance characteristics of 60° V-ribs in aspect ratio 1:8 channel for Re up to 14000. The authors reported that, heat transfer and friction for 60° V-ribs were 2.57 and 2.85 times greater than smooth wall case at Re 14000. Lamont and Kaminski [16] with the help of transient liquid crystal technique compared three rib profiles in a diverging channel and confirmed that, V-ribs at 45° to coolant flow provided uniform and higher levels of heat transfer than orthogonal and 60° ribs at Re 28000. Kumar and Amano [17] studied the effects of V-ribs in a channel ($AR=1:1$) and concluded that, broken ribs augmented heat transfer better than continuous ribs and performances of ribs pointing upstream was best. Xie et al. [18] investigated 45° mid truncated and V-shaped ribs inside aspect ratio 1:1 channel for Re up to 50000. Their investigation revealed that, mid-truncated angled ribs exhibited best performance and mid-truncated V-ribs facing downstream provided best augmentation. Smith et al. [19] assessed the flow patterns for 45° angled ribs ($P/e=10$) in cooling channels with different cross section. For the experimented Re range 4000-130000, they noticed that, secondary flows were well developed in the channel with 1:6 aspect ratio. Ghodake et al. [20] examined the augmentation capabilities of different rib profiles in a rectangular duct. The authors concluded that, rectangular tape ribs and 45° broken V-ribs ($P/e=8.3$) delivered the best heat transfer enhancement. Abraham and Vedula [21] used a converging channel to establish the friction loss and heat transfer for 45° continuous W and V-shaped rib turbulators. The authors reported large differences in heat transfer along the span for V-ribs and also concluded that, performances of the rib profiles were within the range of uncertainty for optimum pitch to height

ratio 10. Prashant et al. [22] studied the performances of four dissimilar profiled ribs and confirmed that, for Re range 19500-69000, performances exhibited by 45° V-ribs and angled ribs were best when compared to other ribs. Ravi et al. [23] evaluated four distinct rib profiles that were inserted at 45 degrees to the flow direction inside a square cross section channel. According to the authors, 45° V-shaped and angled ribs performed best in the Re range of 20000 to 70000. By using 90° ribs, Honghu et al. [24] investigated the effect of three different e/Dh ratios on thermohydraulic performance inside a rotating square channel. For Re up to 40000, the authors found that an e/Dh ratio of 0.1 yielded the best results. V and W-ribs were tested in a channel ($AR=1:4$) by Krishnaswamy and Sivan [25], and found that 45° V and W-ribs with e/Dh values of 0.0729 and 0.0833 produced high thermal performances. Furthermore, the authors stated that W-ribs had the most non-uniformity in heat transport..

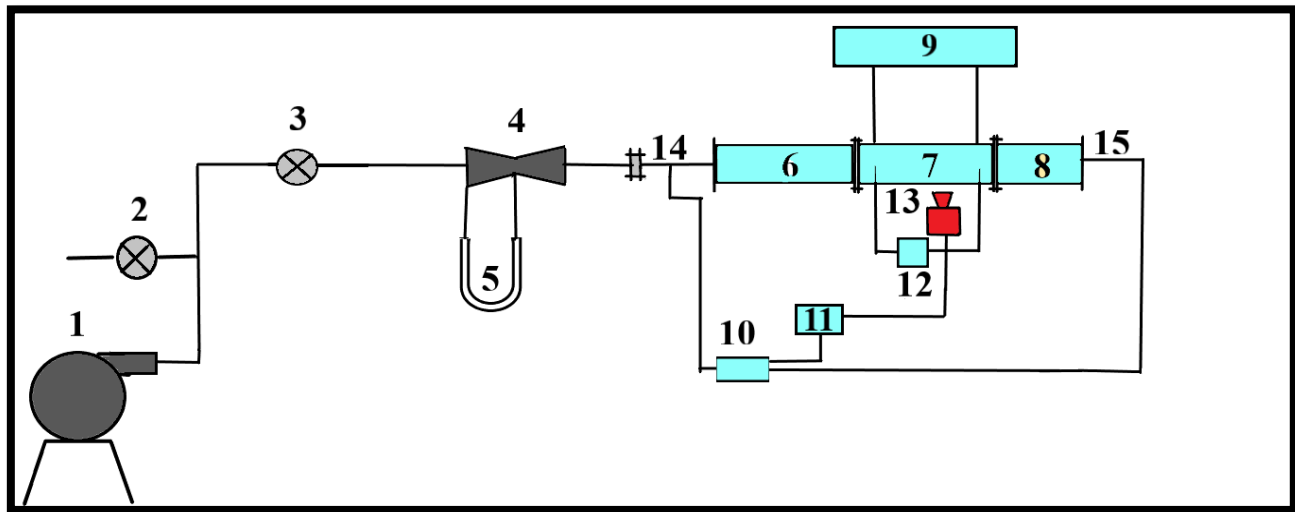
Novelty. The difference in height between the trailing and leading walls of a turbine blade increases towards the leading edge and decreases towards the trailing edge from the mid-chord due to the airfoil contour of the turbine blade. As the height difference grows, very tiny ribbed cooling passageways adjacent to the leading edge appear, with aspect ratios as high as 1:6 [19]. Following a thorough review of the literature, it was discovered that the vast majority of cooling studies conducted to date have been limited to square and narrow ribbed cooling passages with aspect ratios up to 1:2, indicating that there is a need to assess friction and heat transfer in narrow cooling passages adjacent to the leading edge with aspect ratios less than 1:2. Another essential consideration is the efficient management of heat load in the leading edge of an airfoil. Thermal stresses in the regions adjacent to the leading edge are quite high due to high operating temperatures, therefore maximising heat transfer while minimising friction losses and coolant consumption is critical. Because the coolant is used for impingement and film cooling after leaving the ribbed channels, attempting to boost heat transfer by raising the rib height or lowering the rib pitch will simply increase friction, lowering thermohydraulic performance. Furthermore, the efficacy of impingement cooling and film cooling will deteriorate, reducing the turbine blade's effective service life. Considering the above factors, this work has been performed with an aim to achieve high thermohydraulic performance inside a ribbed passage ($AR=1:5$), which is geometrically similar to an actual cooling passage located closer to the turbine blade leading edge. To accomplish this, discrete upstream pointing W and V-ribs were chosen, and essential geometric factors such as pitch, height, and angle-of-attack were tuned to get the optimal outcomes. To the best of our knowledge, ribs with the geometry noted in table 1 have been evaluated for the first time in a ribbed cooling passage with an aspect ratio of 1:5, and the results for the Re range 30000-75000 are promising. 3.2 and 1.7 are the highest and lowest thermohydraulic performances, respectively.

Table.1 Details of rib geometry

Type of ribs	Rib Height (e)	Pitch – to- height ratio (P/e)	Height –to- hydraulic diameter ratio (e/D _h)	Angle of attack (α)	Recommended range in literature		
					P/e	e/D _h	α
Discrete ‘V’	3	12	0.06	45°	5-15	0.05-0.1	30° - 60°
Discrete ‘W’	3	12	0.06	30°			

II.EXPERIMENTAL FACILITY

The system designed for carrying out experiments is illustrated in figure 1. Centrifugal type blower is used to feed air to the system. A bleed valve and control valve are provided to control air flow. A calibrated venturimeter with U-tube water manometer is fitted in the mainline to determine the rate of flow. Downstream of venturimeter is connected to test section. Calibrated thermocouples are mounted at the inlet and outlet ends of test section to record coolant air temperature. Test section is divided into three sections namely developing, ribbed, and exit sections.



- | | | |
|----------------------|-----------------------|---------------------------------------|
| 1. Blower | 6. Developing section | 11. Laptop |
| 2. Bypass Ball valve | 7. Ribbed section | 12. Differential Pressure Transmitter |
| 3. Inlet ball valve | 8. Exit section | 13. Thermal Infrared camera |
| 4. Venturimeter | 9. DC Source | 14. Inlet Thermocouples (K-Type) |
| 5. Manometer | 10. Data logger | 15. Exit Thermocouples (K-Type) |

Fig.1 Schematic of experimental test facility

Test facility is fabricated using 12 mm Perspex sheet, which possesses good strength, low thermal conductivity and ability to withstand temperatures up to 120°C. Cross section (W x H) of test section is 30 mm x 150 mm and the length is 1500 mm. For development of flow, 600 mm long developing section is provided. Ribbed section comprising of two smooth and two ribbed walls is placed at the downstream end of developing section, which is 600 mm long. This section is provided with taps to facilitate pressure drop measurements with the help of pressure transmitter. Thereafter, 300 mm long exit section is provided to ensure that the coolant flow parameters remain unaffected view abrupt expansion of coolant air to atmosphere. The ribbed wall is prepared by using very low thickness stainless steel sheet (0.2 mm, Grade 304). Copper block (25 mm × 28 mm × 3 mm) with brass stud is fixed to the opposite ends of ribbed wall. Ribbed walls are then connected to 1200 W (150 A, 8 V) Direct Current rectifier, which is used as the source of power. Due to very low thickness of ribbed wall (0.2 mm), negligible temperature difference was observed

between inner and outer surfaces, therefore lateral heat conduction between the surfaces is not considered in this work. The effective area of ribbed wall is 400 mm x 30 mm. Electrical resistance of each wall is less than 0.05 Ω. Figure 2 depicts the 3D model of the test section and ribs. Ribs are made from 3 mm balsa wood. Balsa wood is light, possesses high strength and used extensively in aero modeling and rib cooling studies [21]. Discrete V- ribs have two rib arms and discrete W-ribs have four rib arms made into two V-shapes. Arms of both ribs are square in cross section (3 mm x 3 mm). Two arms of discrete V-ribs are positioned with a gap of 3 mm along centerline. Similarly, the two V-shapes of the discrete W-ribs are positioned with a gap of 3 mm along the centerline. Silicone sealant has been utilized to fix the ribs to the wall surface.

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The two ribbed walls are then placed over the perspex sheet and held in position by fasteners. Additionally, heat resistant silicone tapes are also used to prevent air leaks. Exposed surfaces are coated with matt finish black paint to obtain

high emissivity. A thermal imaging camera has been used in the present work to measure the surface temperatures of the ribbed walls.

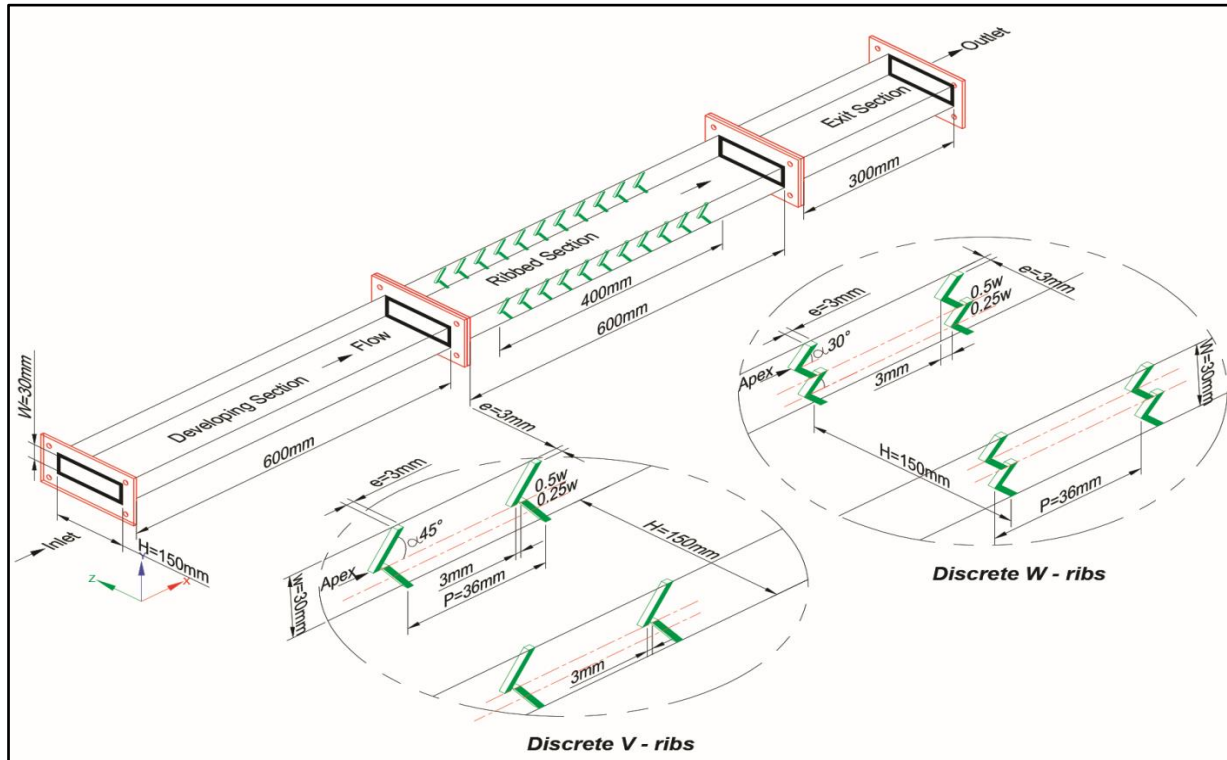


Fig.2 Test section model

III.METHODOLOGY

To produce the differential pressure head matching to Re under investigation, the coolant flow rate is changed. After the flow inside the test section has stabilised, the power source is turned on, and critical parameters such as ribbed wall temperatures, pressure drop across the ribbed section, inlet/exit bulk air temperatures, voltage drop across the copper blocks, and coolant velocity are recorded after 30 minutes. The infrared camera is then used to record ribbed wall images, which are then analysed to establish local wall temperatures. After determining wall temperatures, thermal hydraulic performances are estimated by using following equations

Reynolds number is given by Eq. (1)

$$Re = \frac{\rho v D_h}{\mu} \quad (1)$$

Convective and Radiation losses (Q_{loss}) are deducted from the heat input ($Q_{input} = VI$) for determining net heat input (Q_{net}) to the ribbed walls. Maximum heat loss of 10% is observed at Reynolds number 30000.

Inlet/exit air temperatures measured by thermocouples are verified using energy balance Eq. (2)

$$Q_{net} = \dot{m} c_p (T_{bin} - T_{bout}) \quad (2)$$

Coolant air temperature at a particular location is obtained by interpolating the inlet and exit air temperatures. Air temperature at a particular cross section of the ribbed wall is assumed to be constant. After determining the local wall and coolant air temperatures, local heat transfer coefficients (h) are obtained by using Eq. (3). Subsequently, area averaged Nusselt numbers (Nu) for the smooth and ribbed walls are obtained by using Eq. (4) and Eq. (5) respectively.

$$h = \frac{Q_{net}}{A(T_{wall} - T_{bulk})} \quad (3)$$

$$Nu_s = \frac{\bar{h}_s D_h}{K} \quad (4)$$

$$Nu = \frac{\bar{h} D_h}{K} \quad (5)$$

Nusselt numbers calculated are then normalized with Dittus-Boelter correlation represented by Eq. (6)

$$Nu_o = 0.023 Re^{0.8} Pr^{0.4} \quad (6)$$

Friction factor for the smooth and ribbed walls are determined from Eq. (7) and Eq. (8)

$$f_s = \frac{\Delta P_s D_h}{2 \rho L v^2} \quad (7)$$

$$f = \frac{\Delta P D_h}{2 \rho L v^2} \quad (8)$$

Friction factors are then normalized with Blasius equation represented by Eq. (9)

$$f_o = 0.079 Re^{-1/4} \quad (9)$$

The Thermohydraulic performances at constant pumping power are then determined by using Eq. (10) and Eq. (11)

$$\eta_s = \frac{(Nu/Nu_s)}{(f/f_s)^{1/3}} \quad (10)$$

$$\eta_o = \frac{(Nu/Nu_o)}{(f/f_o)^{1/3}} \quad (11)$$

Uncertainty Estimation. The uncertainty in the Nusselt number and friction factor recorded in this study is primarily due to measurement errors in coolant flow rate and ribbed wall temperatures. The percentage error in measuring coolant flow rate by venturimeter and wall temperatures by thermal camera was estimated using a hot wire anemometer and a calibrated thermocouple (K-type). The pressure difference across the manometer attached to the venturimeter determines the coolant flow rate. The velocity of coolant over the manometer that corresponds to a static differential head is compared to the velocity measured by the anemometer. The error observed ranged from 0.63 m/s to 1.2 m/s over the range of Re tested. A calibrated

thermocouple is fixed to a stainless steel metal sheet (0.2 mm, Grade 304) covered with matt finish black paint used to paint the ribbed walls to estimate the error in wall temperature measurement. The stainless steel sheet is then gradually electrified in steps up to 100° C using the power source. The camera and thermocouple are used to measure steady state temperatures. At 100°C, there is a maximum error of 3°C between camera and thermocouple data. A thermo water bath is used to calibrate the thermocouples used for measurement. The errors are found to be within ±0.05°C. Table 2 highlights the precision and variety of measuring tools used in this study. The highest uncertainty in estimating the Nusselt number is 9.5 percent and friction is 7% for Re 30000 using the methods described by Kline and McClintock [26].

Table.2 Details of Measuring Instruments

Instruments	Type	Accuracy	Range
Voltage Multimeter	M 3900, M/s Mastech	± 0.5%	200 mV to 1000 V
Pressure Inducer	CP 100, M/s KIMO	± 1.5%	0 to 100 Pa
Thermal Image Camera	Ti 9, M/s Fluke	± 5°C	-20°C to 250°C
Anemometer	Testo 425	± 0.03 m/s	0 to 20 m/s
Thermocouple	K-Type	± 0.25%	0 to 600°C

IV.RESULTS AND DISCUSSION

Analysis of Local heat transfer coefficient (*h*). To enhance the life and performance of a gas turbine, uniform and high heat transfer coefficients are desirable inside the cooling passages. However, achieving 100% uniformity may not be possible due to the differences in pattern and strength of secondary flows generated by the ribs. Due to flow separation at the rib, heat transmission in the direction of the stream is mostly driven by the secondary flow. Heat transfer along the rib arms is controlled by the counter rotating vortices created by coolant flow along the arms. In general, characteristics like rib pitch, profile, angle-of-attack, and rib height determine the intensity of secondary flows. Heat transfer for isolated V and W-ribs is presented in this section for the Re range 30000 - 75000. Figures 3-6 show the variations in rib configurations for span lengths of 0.25 w and 0.5 w.

Discrete V-ribs

For the experimented range of Re, secondary flow is found to be dominant along the span 0.5 w (centreline) and therefore the heat transfer coefficients for span length 0.5 w are greater than 0.25 w. The magnitude of heat transfer coefficients between successive ribs varied along the stream and the variations for the experimented range of Re are comparable with the uncertainty estimated in measurement of Nusselt number. The phenomenon of heat transfer variation in the direction of stream is because of the growth of boundary layer which reduces the capacity of coolant to extract heat. Between the span lengths, the variation along 0.5 w is found to be less than 0.25 w due to sudden increase in velocity at the 3 mm gap between the rib arms. The acceleration at the gap is considered beneficial as it results to more uniform heat transfer than that observed for continuous V-ribs. The influence of acceleration on heat

transfer is noticed even along 0.25 w span length and the same is seen to be improving with Re. In addition, as the flow progressed, improvement in secondary flow is observed for all the Re which is speculated to be because of the increment in coolant momentum induced by the gap. In general, for the experimented range of Re, heat transfer in the side wall regions is observed to be lowest. With increase in Re, heat transfer in the side wall regions improved. Averaged variations between highest and lowest heat transfer coefficients has been determined for the span lengths 0.25 w and 0.5 w. At Re 30000, the variations are found to be 11% and 9.5% along 0.25 w and 0.5 w respectively. For the Re 45000 and 60000, variations along 0.25 w are seen to be 10% and 8.5%. Along 0.5 w, the variations are 7.5% and 9% respectively. At Re 75000, the variation is 9.5% along 0.25 w and 8.5% along 0.5 w. To appreciate the effects of ribs better, the wall temperature distribution for the experimented Re range is depicted in figure 7. For all the cases, the wall temperatures along the centreline and nearby regions are found to be less than other regions due to the collective effect of stronger secondary flow produced by the apex and the acceleration induced by the gap. Comparison of thermal plots reveals that, wall temperatures in the regions adjacent to the side walls are highest for Re 30000 due to weak secondary flow. Further, a reduction in difference between the maximum and minimum wall temperatures as a result of improvement in strength of secondary flow with Re is evident from the thermal contours.

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In addition, as the flow progressed, an increase in strength of secondary flow due to the increment in momentum can be well appreciated from the thermal contours corresponding to Re 30000, 45000 and 60000. The increment in strength along the flow is also observed at Re 75000 however, it is relatively lesser which is speculated to be as a result of reduction in coolant flow in the direction of stream due to flow diversion towards side walls.

Discrete W-ribs

For the experimented Re range, heat transfer in the span direction is uniform due to the effect of superior secondary flow produced by the two apexes of discrete W-ribs. Akin to discrete V-ribs, intensity of heat transfer varied in stream direction for both the span lengths and the variation along 0.5 w is found to be marginally lesser than 0.25 w due to influence of discreteness. Average variation along the span lengths has been evaluated and it is seen that the variation is 16% for 0.25 w and 15% for 0.5 w at Re 30000. For the Re 45000 and 60000, variations for span length 0.25 w are 21% and 25% respectively. For the span length 0.5 w, variations are 18% and 19% respectively. At Re 75000, the variations for 0.25 w and 0.5 w span lengths are 19% and 17% respectively. From the above analysis, we can conclude that

the presence of gap helps in achieving more uniform heat transfer which in turn will enhance the service life of the blade. Besides, we can also conclude that, variations along the span are likely to reduce further beyond Re 75000 due to the increase in coolant diversion owing to reduction in boundary layer thickness. As a result of diversion, coolant concentration along 0.5 w and side wall regions increases suddenly which in turn leads to an improvement in uniformity. Wall temperature distribution is depicted in figure 8. In general, for the experimented Re range, difference in wall temperatures along the span is found to be negligible indicative of uniform heat transfer. Comparison of thermal plots reveals that, average wall temperature is highest for Re 30000 and hence the heat transfer is lowest. In addition, the reduction in average wall temperature for an increase in Re can be appreciated from the thermal contours. The influence of 3 mm gap between the rib arms on heat transfer is present however, it is less prominent in comparison to discrete V-ribs. For Re 75000, patches of very low wall temperatures are seen in the regions adjacent to 0.5 w, which is not seen for other Re. This is speculated to be as a result of sudden increase in coolant momentum along the centreline due to the influx of diverted coolant.

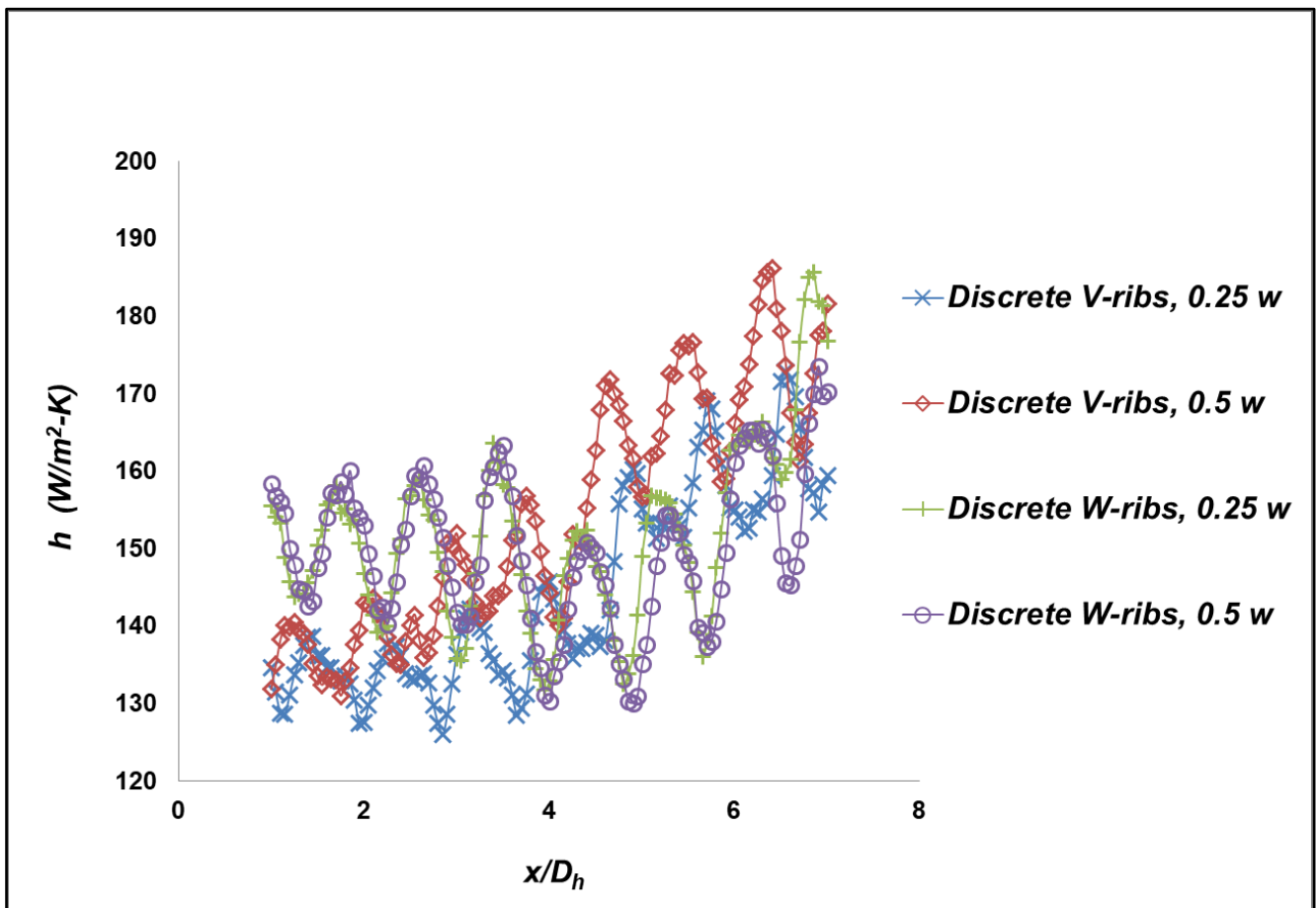


Fig.3 Heat transfer coefficient (h) along the stream for Re 30000

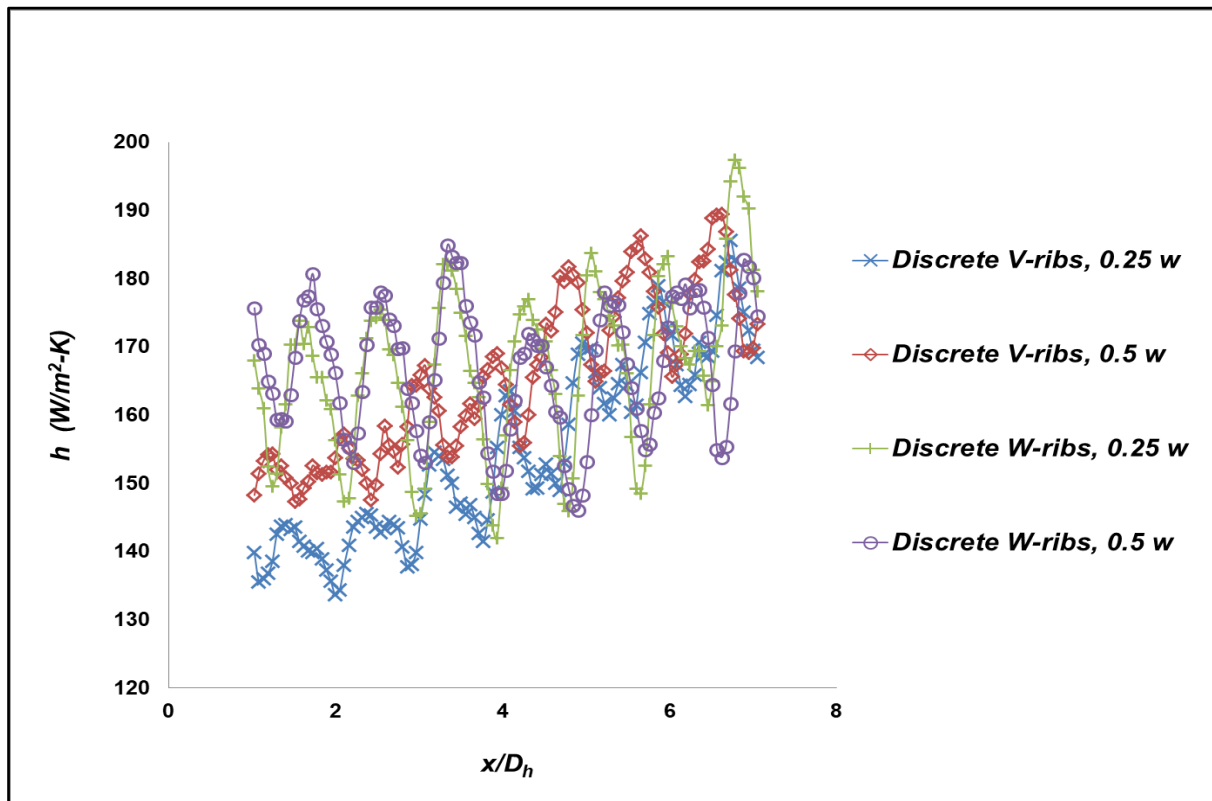


Fig.4 Heat transfer coefficient (h) along the stream for Re 45000

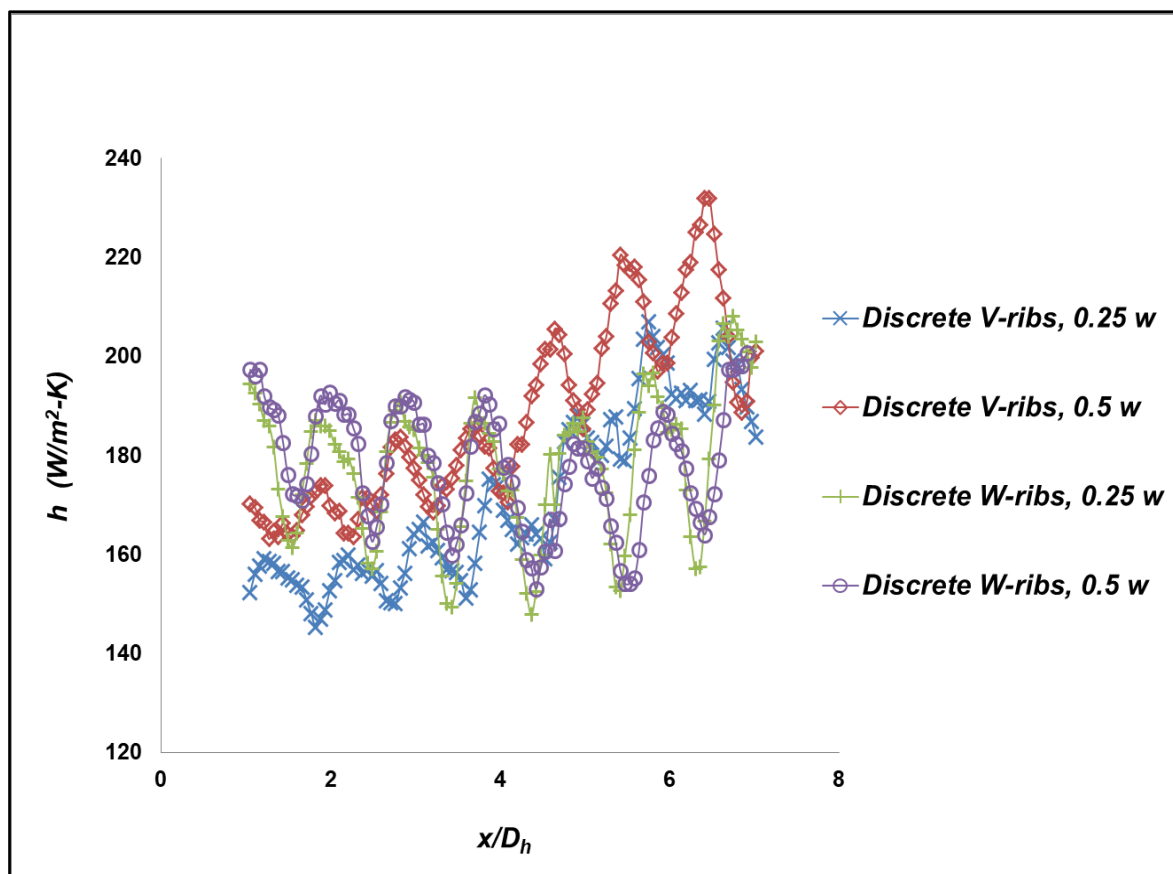


Fig.5 Heat transfer coefficient (h) along the stream for Re 60000

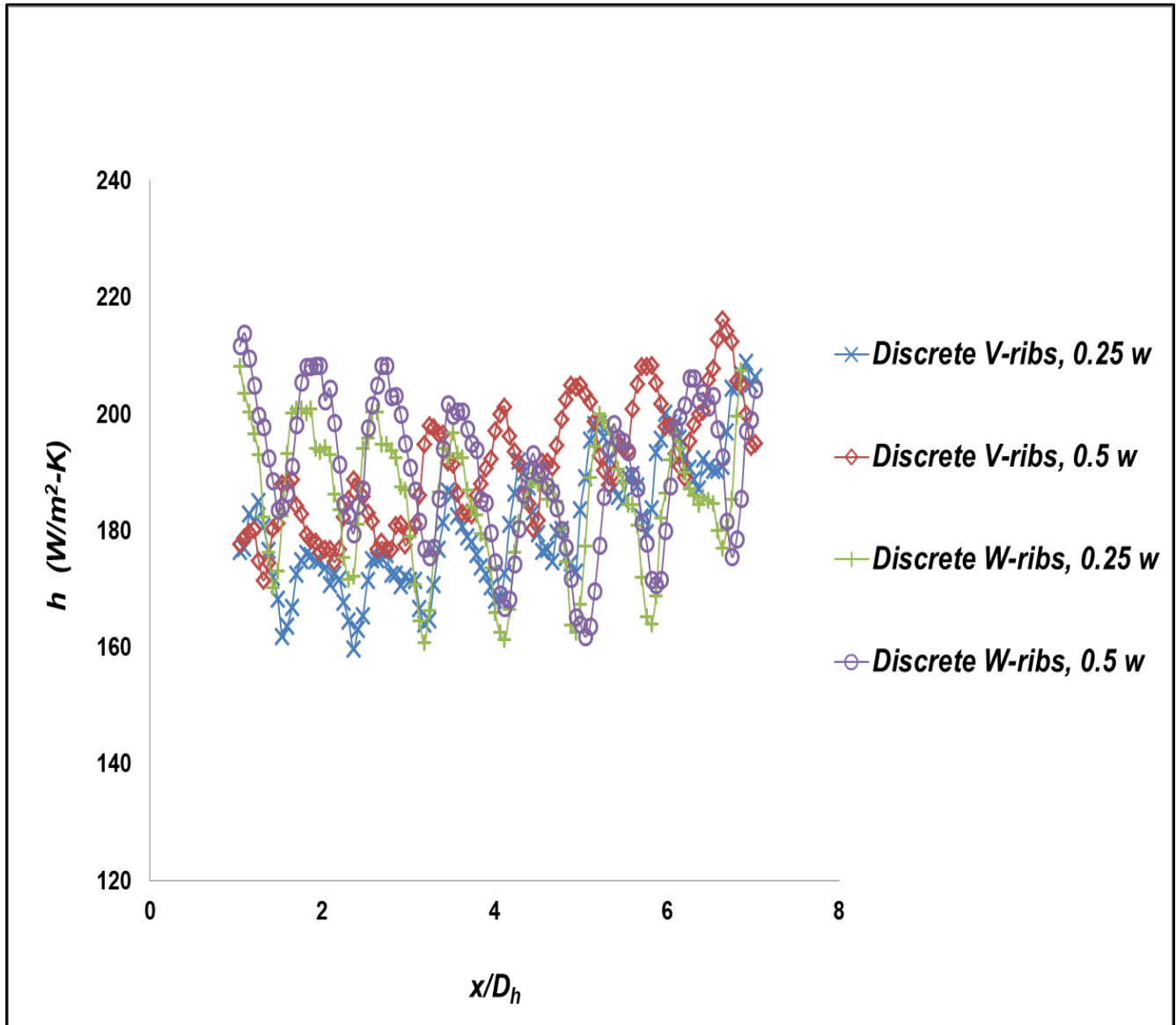
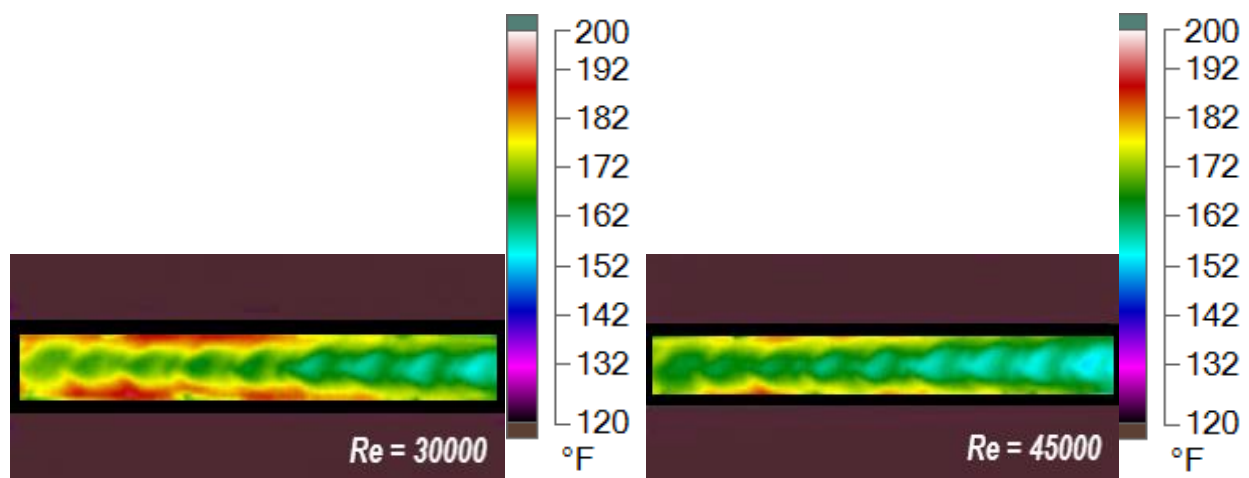


Fig.6 Heat transfer coefficient (h) along the stream for Re 75000



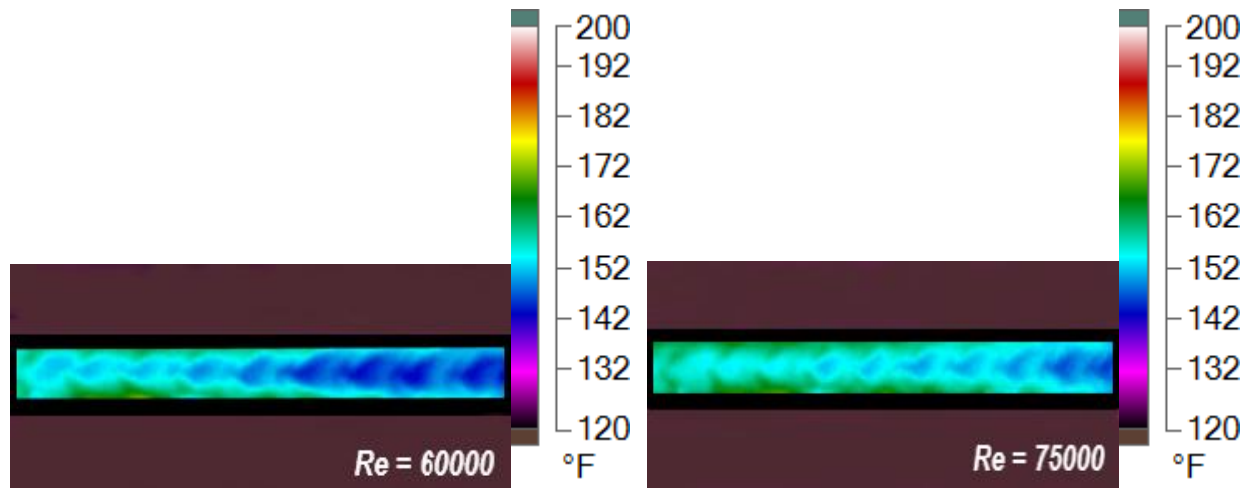


Fig.7 Thermal plots of discrete V-ribs for experimented Re range

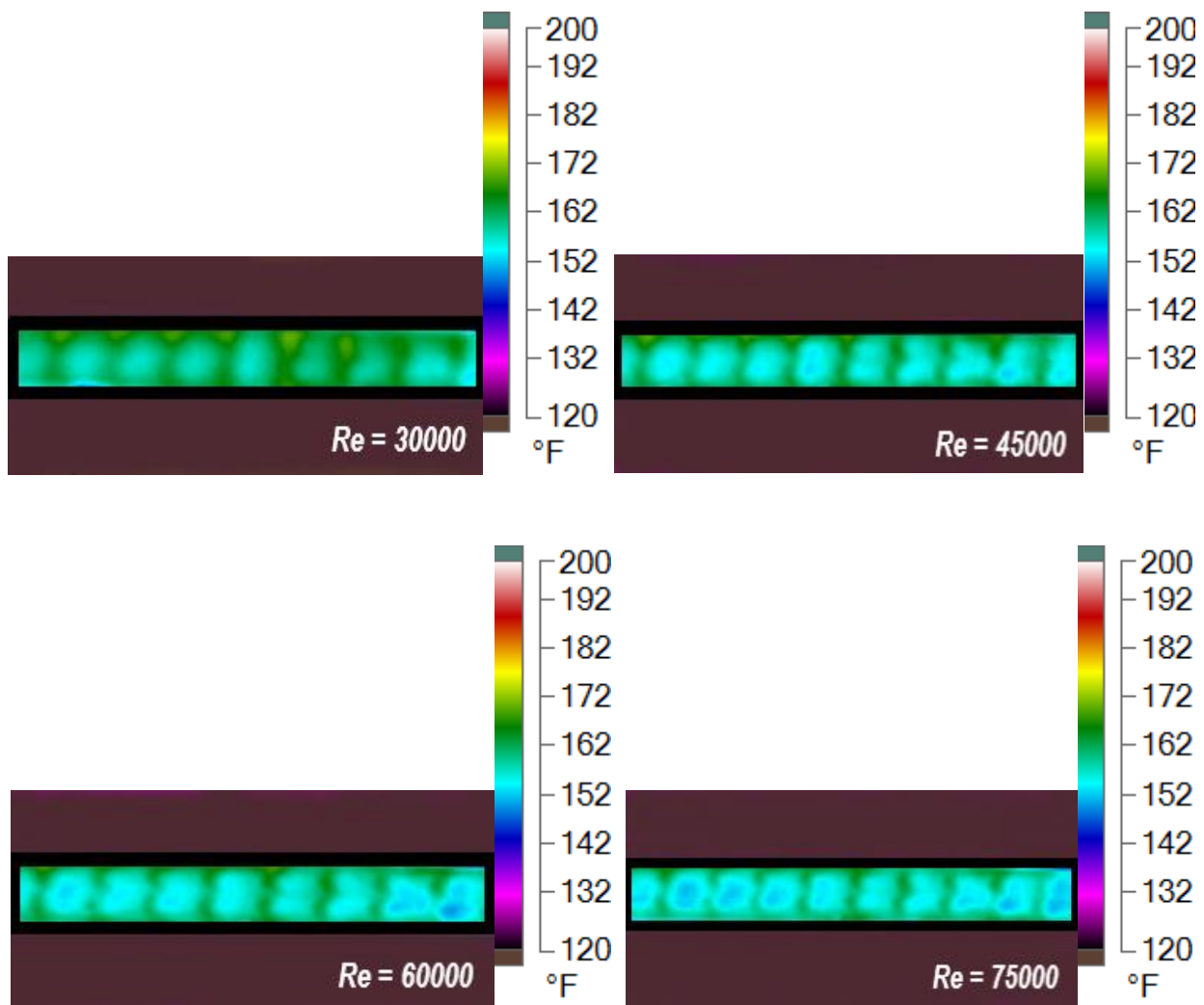


Fig.8 Thermal plots of discrete W-ribs for experimented Re range

Span averaged heat transfer coefficient (h_{avg}). Figure 9 shows h_{avg} corresponding to the investigated Reynolds numbers. For discrete V-ribs, averaged values estimated along 0.5 w are 6-8% higher than that estimated along 0.25 w. As the Re increased, h_{avg} also increased. Maximum increment of 11.5% is observed with increase in Re from 45000 to 60000. Between the Re 30000 and 45000, heat transfer increment is 7% and between the Re 60000 and 75000, the increment is 5%. Beyond Re 60000, the

enhancement is seen reducing in the direction of flow, because of coolant diversion owing to reduction in velocity boundary layer thickness.

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In general, it is noticed that, enhancement in heat transfer achieved between successive Re for discrete V-ribs are relatively lesser than the enhancements presented in literature for continuous V-ribs. From the results we can conclude that, discrete V-ribs in comparison to continuous V-ribs aid in achieving more uniform heat transfer for relatively lesser enhancements. Also, overall heat transfer augmentation in the direction of stream beyond Re 75000 is expected to reduce further because of coolant diversion. While the heat transfer along the stream may not be appreciable, the diversion of coolant will enhance the intensity of vortices along rib arms, which consecutively will augment the heat transfer in side wall and near-rib regions. In the case of discrete W-ribs, enhancements achieved for the span lengths 0.25 w and 0.5 w are comparable with a maximum difference of 2.5% at Re 75000. Akin to discrete V-ribs, for an increase in Re , h_{avg} increased and the increase between successive Re is

observed to be approximately 9.5% for Re up to 60000. Between the Re 60000 and 75000, increase in enhancement is lowest and is calculated to be 2% along 0.25 w and 6% along 0.5 w . The coolant diverted from the apex augmented the strength of secondary flow along span length 0.5 w and therefore the enhancement along 0.5 w is relatively higher than 0.25 w . Similar to V-ribs, coolant diversion along the rib arms due to low velocity boundary layer thickness is considered as the primary reason for reduction in overall enhancement at Re 75000. From the results we can conclude that, discrete W-ribs in comparison to continuous W-ribs aid in achieving more uniform heat transfer for relatively lesser enhancements. Besides, overall augmentation in heat transfer in the direction of stream after Re 75000 is likely to drop further as a result of diversion of coolant away from the stream.

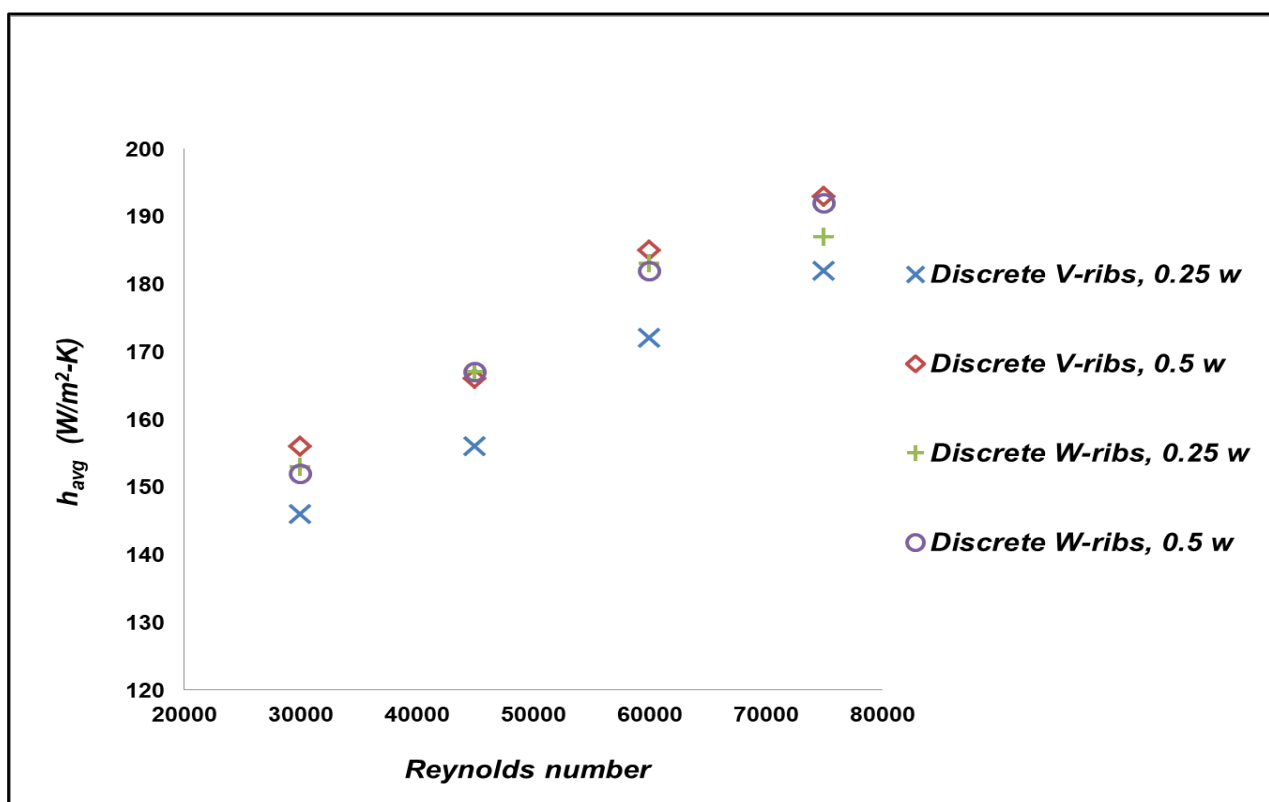


Fig.9 Span averaged heat transfer coefficient (h_{avg}) for investigated Re range

Area averaged Nusselt number ratio. The area averaged Nusselt number ratio (Nu/Nu_0) for the rib profiles is shown in Figure 10. As the Re grew, the Nu/Nu_0 for the rib profiles decreased. Values of 3.4 and 2 are recorded for discrete V-ribs, corresponding to Re 30000 and 75000, respectively. The Nu/Nu_0 is expected to be 2.7 at Re 45000 and 2.3 at Re 60000. Nu/Nu_0 values for discrete W-ribs are calculated to be 3.6 and 2.1 for Re 30000 and 75000, respectively. The values for the Re 45000 and 60000 are 2.8 and 2.4, respectively. Nu/Nu_0 is discovered to be high for both rib profiles, which is attributed to the larger strength of secondary flows created by the ribs inside the narrow

cooling passage. High and uniform heat transfer augmentations are desired to improve an engine's power and performance, and the discrete V and W-ribs assessed in this study are recommended for deployment in places subjected to harsh temperatures. The Nusselt number ratios (Nu/Nu_0) for the rib profiles are shown in Figure 11. Nu/Nu_0 values are fewer than Nu/Nu_0 values for the investigated range of Re . The Nu/Nu_0 values estimated for discrete V-ribs and discrete W-ribs at Re 30000 are 2.7 and 2.9 respectively. For Re 75000, Nu/Nu_0 for discrete V-ribs and discrete W-ribs are 1.8 and 1.9 respectively.

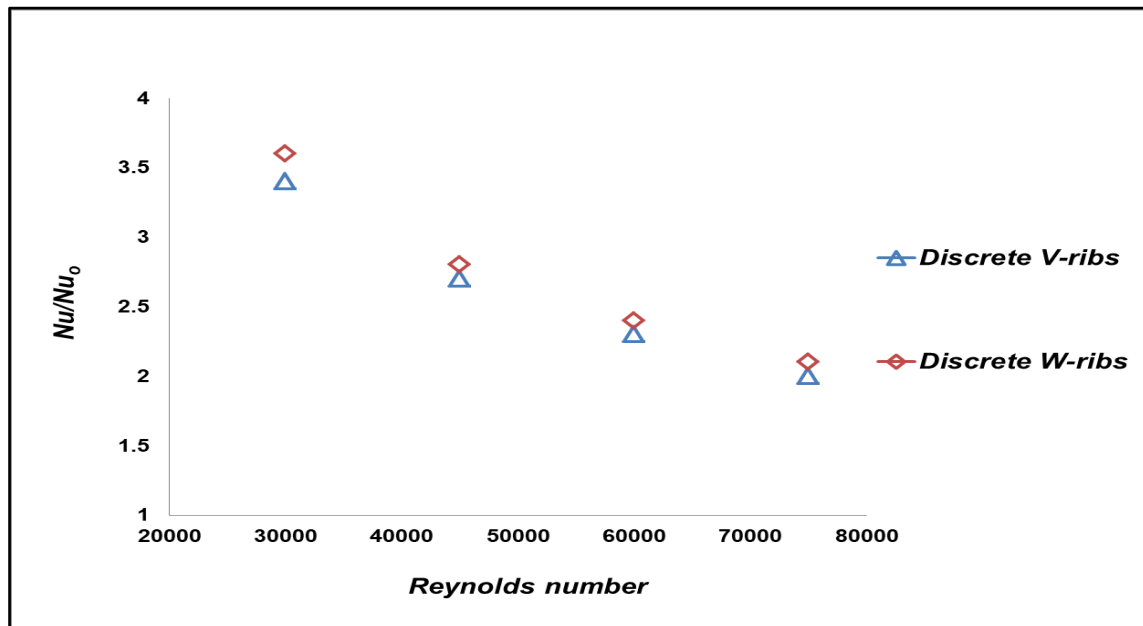


Fig.10 Area averaged Nusselt number ratio (Nu/Nu_0)

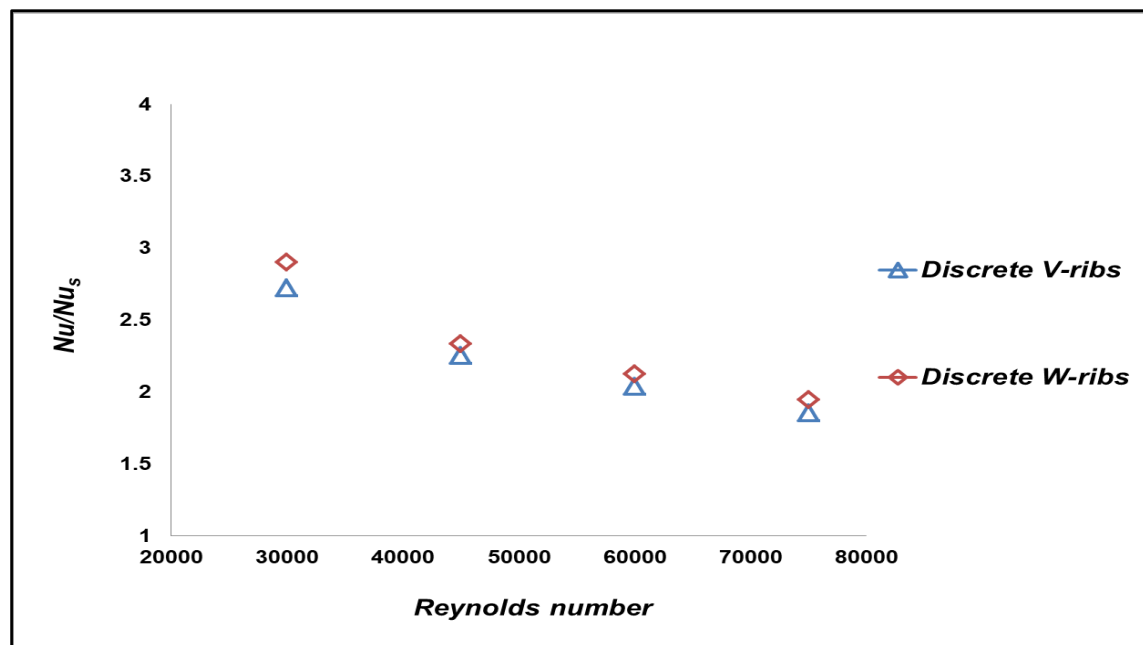


Fig.11 Area averaged Nusselt number ratio (Nu/Nu_s)

Normalized Friction factor. The magnitude of friction loss inside a ribbed cooling tunnel is mostly determined by the pressure drop caused by the ribs. The friction factor ratio (f/f_0) for the rib profiles is shown in Figure 12. Friction factor ratios for discrete V-ribs range from 1.2 to 1.7 for the Re range studied. Between consecutive Re, maximum pressure drop increment of 17% is observed between the Re 30,000 and 45,000. Subsequently, increment is seen to be 14% between the Re 45,000 and 60,000 and 6% between the Re 60,000 and 75,000. Appreciable reduction in f/f_0 increment between the Re 60,000 and 75,000 is because of the occurrence of coolant diversion towards side walls, which eventually led to a reduction in overall pressure drop. For discrete W-ribs, values corresponding to the experimented Re are observed to be ranging from 1.6-2. For Re up to 60,000, pressure drop increment between successive Re is noticed to be the same

at 6%. Between Re 60,000 and 75,000, one would anticipate a drop in increment due to the diversion of coolant. However, in the contrary, increase in pressure drop is seen to be highest at Re 75,000, which resulted to a difference of 11% between the friction factor ratios. The increase is attributed to the sudden rise in coolant concentration along span length 0.5 w due to diversion, which eventually resulted to an appreciable increase in pressure drop. Despite having the same blockage ratio and pitch, it can be seen that, pressure drop generated by the discrete W-ribs are higher than discrete V-ribs due to the combined effect of low angle-of-attack and large contact area.

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Figure 13 presents the friction factor ratio (f/f_s) for discrete V and W-ribs. As expected, for the range of Re studied, f/f_s of discrete W-ribs are noticed to be higher than that of discrete V-ribs. Comparison of smooth wall friction (f_s) with predicted friction (f_o) reveals that the smooth wall friction is lesser than predicted friction. This may be because the flow is streamlined inside the test section with

AR =1:5 due to the large gap between the walls. For the Re range 30000 and 75000, differences noticed between friction factors f_s and f_o are nearly 5% and 21%. As the friction losses are observed to be low inside the aspect ratio 1:5 channel, higher blockage ratio ribs can be deployed to achieve higher enhancements with no major effect on friction loss.

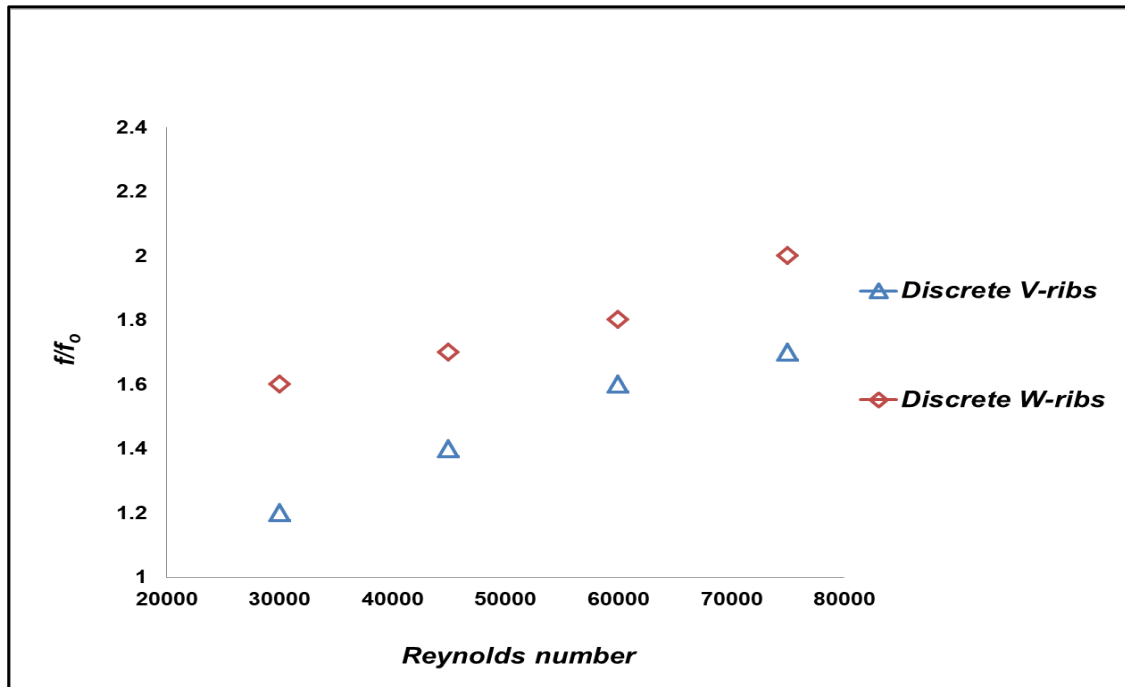


Fig.12 Normalized friction factor (f/f_o) for investigated range of Re

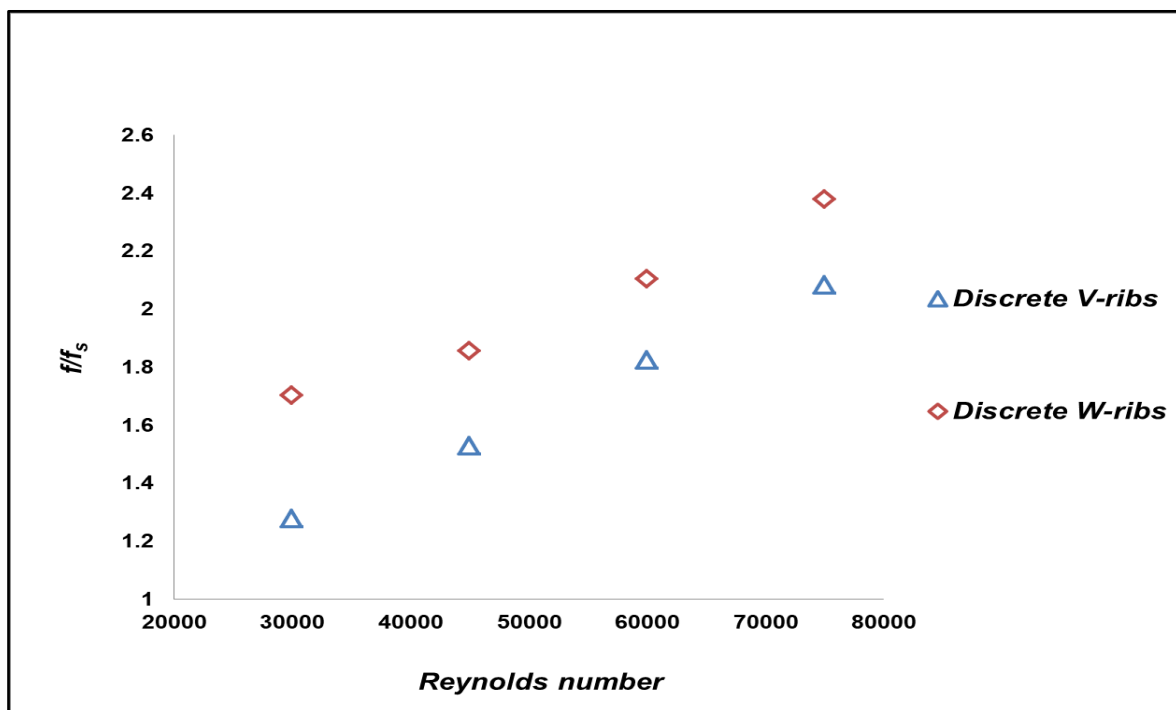


Fig.13 Normalized friction factor (f/f_s) for investigated range of Re

Thermohydraulic Performance. Figure 14 shows the Thermohydraulic Performance (THPo) values for the Re range investigated. For discrete V-ribs, THPo values corresponding to the maximum and minimum Re experimented are 3.2 and 1.7. For Re 45000 and 60000, the values are 2.4 and 2. For Re 45000 and 60000, the values are 2.4 and 2. Analysis reveals that, reduction is highest at 33% between Re 30000 and 45000. This is because, reduction in Nu/Nu_0 is relatively higher than the reduction noticed in f/f_0 between Re 30000 and 45000. For discrete W-ribs, maximum and minimum thermohydraulic performances for Re 30000 and 75000 are 3 and 1.65. The

THPo values estimated for Re 45000 and 60000 are 2.3 and 1.95. Similar to discrete V-ribs, between Re 30000 and 45000, reduction in Nu/Nu_0 is comparatively greater than the reduction noticed in f/f_0 and therefore the reduction in THPo is highest at 30% in this range of Re. Figure 15 depicts the Thermohydraulic Performance (THPs) values for the experimented Re. The THPs of discrete V and W-ribs are estimated to be lesser than THPo. Corresponding to Re 30000 and 75000, THPs values for discrete V-ribs are estimated as 2.6 and 1.5 respectively. For discrete W-ribs, the estimated values are 2.4 and 1.5.

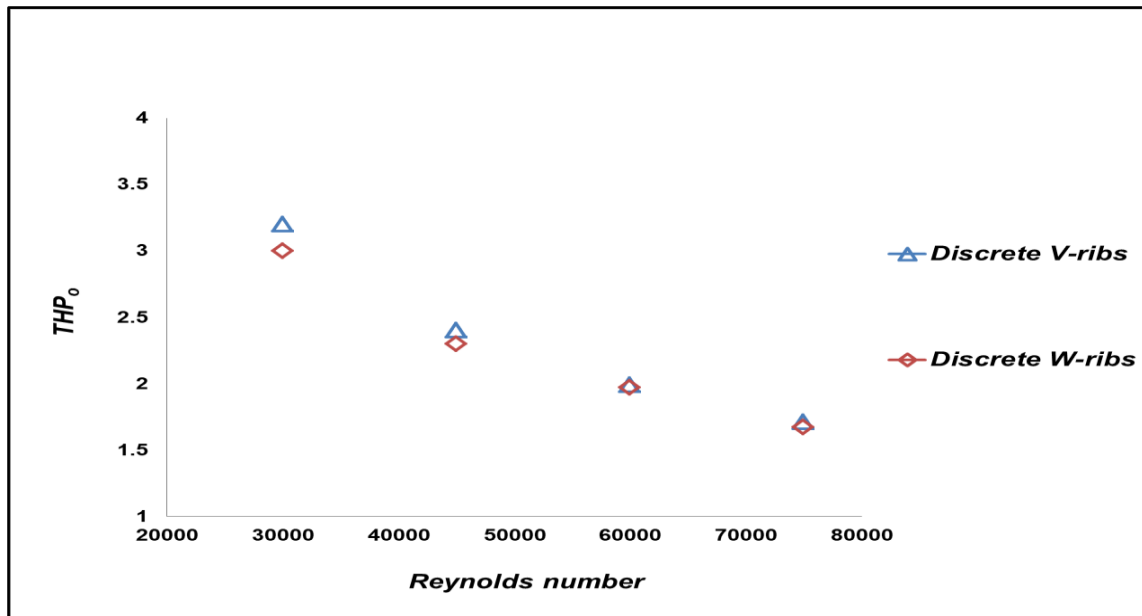


Fig.14 Thermohydraulic Performance (THP_o) of both ribs for investigated range of Re

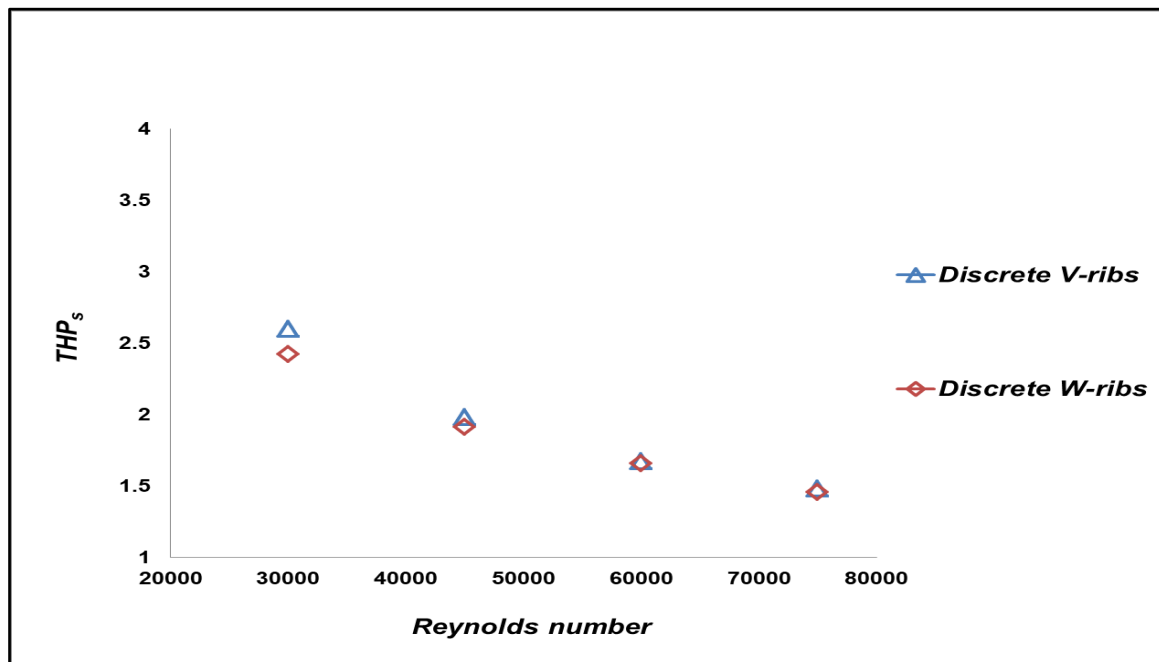


Fig.15 Thermohydraulic Performance (THP_s) of both ribs for investigated range of Re

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Comparative Study.

Table 3 and figure 16 shows the Nu/Nu_o values of the present and published studies. The Nu/Nu_o value for discrete V-ribs ($e/D_h=0.06$) corresponding to Re 30000 is 10% higher than broken V-ribs ($e/D_h=0.0625$) inside an aspect ratio 1:1 channel studied by Han and Zhang [3]. However, at Re 45000, value is found to be equal and for Re 60000 and 75000, Nu/Nu_o values reported by Han and Zhang [3] are higher than the present findings. Nu/Nu_o ratios for discrete V-ribs with blockage ratio 0.06 are 15-17 percent higher for Re 30000 and 60000 when compared to Ekkad and Han's results for broken V-ribs ($e/D_h=0.125$) inside an aspect ratio 1:1 channel [4]. When compared to Wright et al. [7]'s Nu/Nu_o value for discrete V-ribs ($e/D_h=0.078$) in a channel with an aspect ratio of 4:1, discrete V-ribs employed in this study generated an 8 percent higher enhancement at Re 30000. Fu et al. [9] discovered that the enhancement recorded for 45° ribs ($e/D_h=0.078$) in an aspect ratio 1:4 channel is 26% less than the enhancement calculated at Re 30000 for discrete V-ribs. Furthermore, the Nu/Nu_o value for discrete V-ribs at Re 30000 is 13% higher than that predicted by Prashant et al. [22] in an aspect ratio 1:1 channel for continuous V-ribs ($e/D_h=0.125$). At Re 60000, the Nu/Nu_o value estimated by Prashant et al. [22] is calculated to be 13% higher than the enhancement achieved for discrete V-ribs. In comparison to results presented by Krishnaswamy and Sivan [25] for continuous V-ribs ($e/D_h=0.0729$), the enhancements generated by discrete V-ribs with 21.5% less blockage ratio are seen to be lesser by 17-26%. The Nu/Nu_o values of discrete W-ribs, like those of discrete V-ribs, are found to be greater than those reported in the literature. When compared to the enhancement anticipated by Wright et al. [7] for discrete W-ribs ($e/D_h=0.078$) inside an aspect ratio 4:1 channel at Re 30000, the current work finds that

enhancement owing to discrete W-ribs ($e/D_h = 0.06$) is 6 percent higher. Furthermore, Nu/Nu_o values estimated for discrete W-ribs in our work are 44 percent and 14 percent greater than those predicted by Prashant et al. [22] for continuous W-ribs for the Re 30000 and 60000. Furthermore, discrete W-rib improvements are shown to be 17-24 percent lower than those reported by Krishnaswamy and Sivan [25] for continuous W-ribs ($e/D_h=0.0833$). The enhancement owing to discrete W-ribs at Re 30000 is noticed to be 33 percent larger than the enhancement predicted by Fu et al. [9] for 45° ribs ($e/D_h=0.078$) in an aspect ratio 1:4 channel. THPo values of discrete V-ribs and discrete W-ribs have been evaluated with different rib designs, as shown in table 4 and figure 17. The THPo values of discrete V and W-ribs inside cooling channels with aspect ratios of 4:1, 1:6, and 1:1 are 55 percent to 120 percent higher than those provided for other rib designs such as discrete V-ribs, discrete W-ribs, angled and continuous ribs. Furthermore, the THPo values of discrete V and W-ribs observed at Re 30000 are greater than those anticipated by DENG Honghu et al. [24] for 90° ribs inside a square channel with a 40% higher blockage ratio at Re 20000. Also, the THPo of discrete V and W-ribs are comparable with that of continuous V and W-ribs with higher blockage ratios studied by Krishnaswamy and Sivan [25]. The above findings reiterate the fact that, discrete V and W-ribs can produce superior thermohydraulic performances inside cooling passages with low aspect ratios and hence they can be deployed in regions exposed to extreme temperatures to deliver high thermal performance. Besides, the discrete ribs aid in improving the power-to-weight ratio as the ribs can produce high thermohydraulic performances for low blockage ratios.

Table.3 Nu/Nu_o comparison

	Type of Ribs	AR	e/D_h	P/e	α	Re	Nu/Nu_o
Present Study	Discrete V- ribs	1:5	0.06	12	45°	30000/ 45000/ 60000/ 75000	3.4/2.7/2.3/2
	Discrete W- ribs	1:5	0.06	12	30°		3.6/2.8/2.4/2.1
Han and Zhang [3]	V-shaped broken ribs	1:1	0.0625	10	45°	30000/ 45000/ 60000/ 75000	3.1/2.7/2.6/2.5
Ekkad and Han [4]	V-shaped broken ribs	1:1	0.125	10	60°	30000/ 60000	2.9/ 2
Wright et al. [7]	Discrete V- ribs	4:1	0.078	10	45°	30000	3.15
	Discrete W- ribs						3.4
Fu et al. [9]	Angled ribs	1:4	0.078	10	45°	30000	2.7
Prashant et al. [22]	V-ribs	1:1	0.125	16	45°	30000/ 60000	3.0/2.6
	W-ribs						2.5/2.1
Krishnaswamy and Sivan [25]	V-ribs	1:4	0.0729	10	45°	40000/ 60000/ 80000	3.4/2.7/2.4
	W-ribs		0.0833	8.75			3.3/2.8/2.6



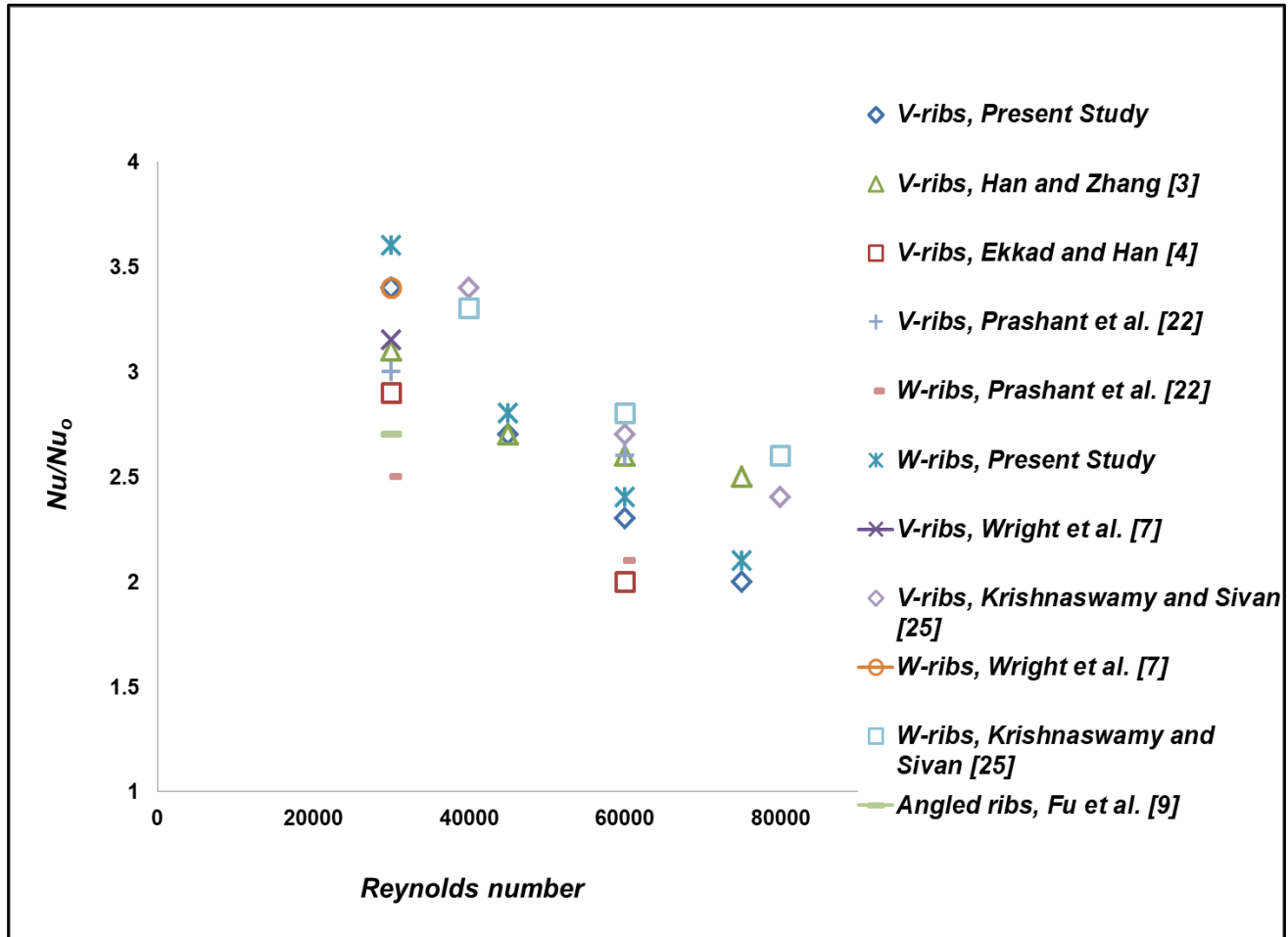


Fig. 16 Comparison of Nu/Nu_0 with published results

Table.4 THP_0 comparison

	Type of Ribs	AR	e/D_h	P/e	α	Re	THP_0
Present Study	Discrete V-ribs	1:5	0.06	9	30°	30000/ 45000/ 60000/ 75000	3.2/2.4/2/1.7
	Discrete W-ribs						3/2.3/1.95/1.65
Wright et al. [7]	Discrete V-ribs	4:1	0.078	10	45°	30000	1.55
	Discrete W-ribs						1.55
Smith et al. [19]	Angled ribs	1:6	0.058	10	45°	25000/ 50000/ 75000	1.5/1.2/1
Prashant et al. [22]	V-ribs	1:1	0.125	16	45°	30000/ 60000	1.55/1.3
	W-ribs						1.35/1.1
DENG Honghu et al. [24]	Orthogonal ribs	1:1	0.1	-	90°	20000	3
Krishnaswamy an Sivan [25]	V-ribs	1:4	0.0729	10	45°	40000/6000 0/80000	2.4/1.8/1.6
	W-ribs		0.0833	8.75			2.3/1.7/1.5

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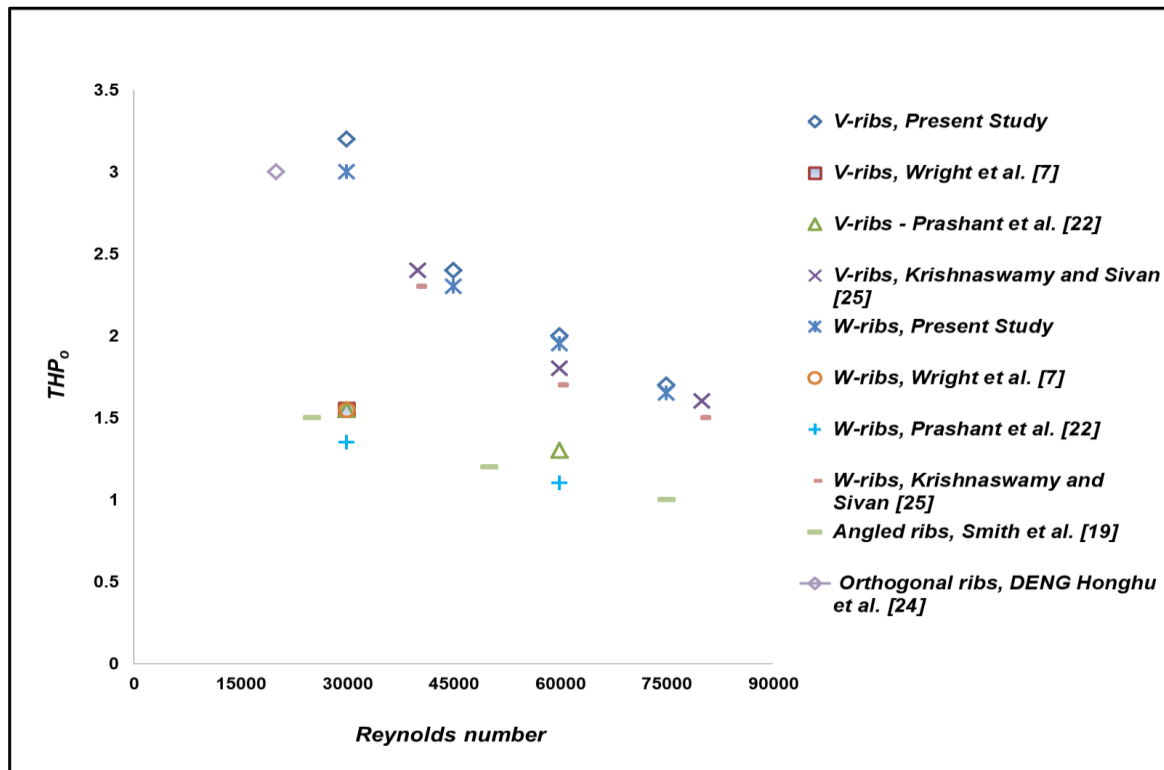


Fig.17 Comparison of THP_o with published results

V.CONCLUSIONS

Nusselt number ratio (Nu/Nu_o) and Thermohydraulic Performance (THP_o) values for the experimented Re range 30000-75000 are found to reasonably good for both the rib profiles. The inferences drawn from the results are as shown below:-

Discrete V-ribs

- For Re range investigated, wall temperatures along the span length 0.5 w (centreline) and nearby regions are found to be lesser than other regions due to the collective effect of stronger secondary flow induced by the apex and the acceleration induced by the gap.
- Improvement in secondary flow in the direction of stream is observed for all Re which is speculated to be because of the increment in coolant momentum induced by the discreteness. In general, for the experimented Re range, heat transfer in the side wall regions is observed to be lowest which subsequently improved with Re .
- The magnitude of heat transfer coefficients between successive ribs varied along the stream and the variations for the experimented range of Re are comparable with the uncertainty estimated in measurement of Nusselt number.
- Between the span lengths, variation along 0.5 w is found to be less than 0.25 w due to abrupt increase in velocity at the gap between the rib arms. The influence of acceleration on heat transfer is noticed even along 0.25 w span length and the same is found to be increasing with Re . The maximum average variation is 11% along the span length 0.25 w at Re 30000. As the Re increased, variation reduced.

- As the Re increased, heat transfer coefficients also increased. Maximum increase of 11.5% is observed between Re 45000 and 60000. At Re 75000, the overall increment reduced because of reduction in intensity of secondary flow along the stream as a result of coolant diversion. The Nu/Nu_o values are observed to be 3.4 and 2 corresponding to Re 30000 and 75000 respectively. Nu/Nu_s values estimated for Re 30000 and 75000 are 2.7 and 1.8 respectively.
- Friction factor ratio (f/f_o) increased as the Re increased. The values ranged from 1.2-1.7 for the Re range investigated. Between consecutive Re , maximum pressure drop increment of 17% is observed between Re 30000 and 45000. Between the Re 60000 and 75000, increment is found to be lowest at 6%. Appreciable reduction in f/f_o increment between Re 60000 and 75000 is because of the coolant diversion towards side walls which subsequently reduced the overall pressure drop. The f/f_s values are higher than f/f_o and they range from 1.3-2.1.
- The THP_o values corresponding to the maximum and minimum Re experimented are 3.2 and 1.7. For Re 45000 and 60000, the thermohydraulic performances estimated are 2.4 and 2. THP_s values are comparatively lesser than THP_o and are estimated to be 2.6 and 1.5 for Re 30000 and 75000.

- From the inferences drawn, discrete V-rib profile is recommended for deployment in aspect ratio 1:5 channel for Re ranging from 45000 to 75000.

Discrete W-ribs

- For the experimented Re range, heat transfer along the span is uniform due to the effect of superior secondary flow produced by the two apexes. The intensity of heat transfer varied in the direction of stream and the variation along 0.5 w span length is found to be marginally lesser than 0.25 w due to influence of discreteness.
- Unlike discrete V-ribs, variations in heat transfer in the direction of stream increased with Re and is found to be highest at Re 60000. Average variation for span lengths 0.25 w and 0.5 w are estimated to be 25% and 19%. At Re 75000, as a result of reduction in intensity of secondary flow due to coolant diversion, variation along 0.25 w and 0.5 w reduced and are estimated to be 19% and 17% respectively.
- The enhancements achieved along the span lengths 0.25 w and 0.5 w are comparable with a maximum difference of 2.5% at Re 75000. The heat transfer coefficient increased with Re and the increase between successive Re is observed to be approximately 9.5% for Re up to 60000. Between Re 60000 and 75000, increase in enhancement is lowest as a result of coolant diversion and is calculated to be 2% for the span length 0.25 w and 6% for the span length 0.5 w. The area averaged Nusselt number ratios (Nu/Nu_o) for Re 30000 and 75000 are 3.6 and 2.1 respectively. Nu/Nu_s for the same Re are 2.9 and 1.9.
- The f/f_o values corresponding to the experimented Re range from 1.6-2. For Re up to 60000, increment in f/f_o between consecutive Re is noticed to be the same at 6%. Between the Re 60000 and 75000, increment in f/f_o is observed to be highest at 11% due to sudden influx of coolant between the converging rib arms. In comparison to f/f_o values, friction factor ratio (f/f_s) values are higher and they range from 1.7-2.4.
- The maximum and minimum thermohydraulic performances for Re 30000 and 75000 are 3 and 1.65. THP_s values are comparatively lesser than THP_o and the values are 2.4 and 1.5 for Re 30000 and 75000 respectively.
- From the inferences drawn, discrete W-rib profile is recommended for deployment in aspect ratio 1:5 channel for Re ranging from 30000 to 75000.

Nomenclature

α	Angle of attack
P	Rib pitch, mm
e	Height of the rib, mm
D_h	Hydraulic diameter of test section, mm
W	Width of the test section, mm
H	Height of the test section, mm
v	Velocity of air, m/s
w	Width of the ribbed wall, mm
\dot{m}	mass flow rate, Kg/s
h	Local heat transfer coefficient, W/m ² -K
h_{avg}	Span averaged heat transfer coefficient, W/m ² -K

\bar{h}	Area averaged heat transfer coefficient for ribbed wall, W/m ² -K
\bar{h}_s	Area averaged heat transfer coefficient for smooth wall, W/m ² -K
Re	Reynolds number
Nu	Area averaged Nusselt number for ribbed wall
Nu_s	Area averaged Nusselt number for smooth wall
Nu_o	Nusselt number for fully developed turbulent flow in a smooth circular tube (Dittus- Boelter correlation)
Pr	Prandtl number
ρ	Density of air, Kg/m ³
k	Thermal Conductivity of air, W/m-K
Q_{input}	Heat input, W
Q_{loss}	Heat loss, W
Q_{net}	Net heat input, W
A	Effective ribbed surface area, m ²
L	Length between the pressure taps, m
T_{bulk}	Local bulk temperature of air, K
T_{bin}	Bulk air inlet temperature, K
T_{bout}	Bulk air outlet temperature, K
T_{wall}	Local wall temperature of air, K
c_p	Specific heat of air, J/Kg K
I	Current
V	Voltage
f	Friction factor for ribbed wall
f_s	Friction factor for smooth wall
f_o	Friction factor for fully developed turbulent flow in a smooth tube (Blasius equation)
η_o	Thermal Hydraulic Performance with respect to Dittus-Boelter correlation (THP_o)
η_s	Thermal Hydraulic Performance with respect to smooth wall (THP_s)
ΔP	Pressure drop across the ribbed section, N/m ²
ΔP_s	Pressure drop across the smooth section, N/m ²
A	Ampere
V	Volt
mV	Millivolt
W	Watt
Pa	Pascal

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