

Heat Transfer Performance Improvement Technologies for Hot Gas Path Components in Gas Turbines

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Thesis submitted to the faculty of the
Virginia Polytechnic Institute and State University
in partial fulfillment of the requirements for the degree of

Master of Science
In
Mechanical Engineering

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April 29, 2016
Blacksburg, VA

Keywords: Computational Fluid Dynamics, Heat Transfer, Gas Turbines, Rib Turbulators,
Internal Cooling, Endwall, Film Cooling

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ABSTRACT

In the past few decades, the operating temperatures of gas turbine engines have increased significantly with a view towards increasing the overall thermal efficiency and specific power output. As a result of increased turbine inlet temperatures, the hot gas path components downstream of the combustor section are subjected to high heat loads. Though materials with improved temperature capabilities are used in the construction of the hot gas path components, in order to ensure safe and durable operation, the hot gas path components are additionally supplemented with thermal barrier coatings (TBCs) and sophisticated cooling techniques. The present study focusses on two aspects of gas turbine cooling, namely augmented internal cooling and external film cooling.

One of the commonly used methods for cooling the vanes involves passing coolant air bled from the compressor through serpentine passages inside the airfoils. The walls of the internal cooling passages are usually roughened with turbulence promoters like ribs to enhance heat transfer. Though the ribs help in augmenting the heat transfer, they have an associated pressure penalty as well. Therefore, it is important to study the thermal-hydraulic performance of ribbed internal cooling passages. The first section of the thesis deals with the numerical investigation of flow and heat transfer characteristics in a ribbed two-pass channel. Four different rib shapes- 45° angled, V-shaped, W-shaped and M-shaped, were studied. This study further aims at exploring the performance of different rib-shapes at a large rib pitch-to-height ratio ($p/e=16$) which has potential applications in land-based gas turbines operating at high Reynolds numbers. Detailed flow and heat transfer analysis have been presented to illustrate how the innate flow physics associated with the bend region and the different rib shapes contribute to heat transfer enhancement in the two-pass channel. The bend-induced secondary flows were observed to significantly affect the flow and heat transfer distribution in the 2nd pass. The thermal-hydraulic performance of V-shaped and 45° angled ribs were better than W-shaped and M-shaped ribs.

The second section of the study deals with the analysis of film cooling performance of different hole configurations on the endwall upstream of a first stage nozzle guide vane. The flow along the endwall of the airfoils is highly complex, dominated by 3-dimensional secondary flows. The presence of complex secondary flows makes the cooling of the airfoil endwalls challenging. These secondary flows strongly influence endwall film cooling and the associated heat transfer. In this study, three different cooling configurations- slot, cylindrical holes and tripod holes were studied. Steady-state experiments were conducted in a low speed, linear cascade wind tunnel. The adiabatic film cooling effectiveness on the endwall was computed based on the spatially resolved temperature data obtained from the infrared camera. The effect of mass flow ratio on the film cooling performance of the different configurations was also explored. For all the configurations, the coolant jets were unable to overcome the strong secondary flows inside the passage at low mass flow ratios. However, the coolant jets were observed to provide much better film coverage at higher mass flow ratios. In case of cylindrical ejection, the effectiveness values were observed to be very low which could be because of jet lift-off. The effectiveness of tripod ejection was comparable to slot ejection at mass flow ratios between 0.5-1.5, while at higher mass flow ratios, slot ejection was observed to outperform tripod ejection.

GENERAL ABSTRACT

Natural gas based power generation contributes to approximately one-fifth of the global electricity production. Gas turbine engines produce power by burning fuel in a combustion chamber and using the high temperature, high pressure combustion gases to drive the power turbine. With the rising power demand, there is a growing need to augment the cycle efficiency and specific power output of gas turbine engines. Higher turbine inlet temperatures are desirable from the standpoint of cycle efficiency and power output, but they in turn impose high thermal loads on the hot gas path components. These hot gas path components, commonly made of nickel-based superalloys are further supplemented with insulating thermal barrier coatings (TBCs) and sophisticated cooling arrangements in order to ensure durability and safe operation. Two commonly used techniques which are employed for cooling the hot gas path components are internal convective cooling and film cooling. In internal convective cooling, the coolant air bled from the compressor is passed through serpentine passages inside the vanes. On the other hand, film cooling involves passing cold air from the compressor through cooling holes on the surface to form a thin insulating film which protects the surface from the hot mainstream gases. This study focusses on the aforementioned two commonly used techniques for cooling the hot gas path components. The first part of this work deals with the investigation of heat transfer augmentation techniques for the internal cooling channels while the second part focusses on the analysis of the performance of different film cooling configurations on the turbine endwall. The results from this study could provide valuable insights for turbine designers to improve the techniques used for cooling the hot gas path components.

ACKNOWLEDGEMENTS

I would like to thank Dr. Srinath Ekkad for giving me the opportunity to work with him and supporting me in all my endeavours. His valuable guidance and expert advice have contributed immensely to my holistic growth. He has been and will continue to be a great source of inspiration to me. I would also like to thank Dr. Scott Huxtable for his continued support and guidance. I would like to extend my thanks to Dr. Rui Qiao for serving on my committee.

I would like to express my gratitude to Dr. Jaideep Pandit, David Gomez and Sridharan Ramesh for sharing their expertise and valuable insights on subject matters. Special thanks to Prashant Singh for all the thoughtful discussions, support and encouragement. I would also like to acknowledge the other members of the Heat, Energy and Fluids Transport (HEFT) Lab, past and present, Kris, Kartikeya, Samruddhi, Prethive, Sandeep, Siddhartha and Shubham. I have truly cherished the two years that I have spent in the HEFT lab. Thanks for all the fun times and wonderful memories.

I have always been fortunate to have been surrounded by wonderful friends. Adithya, Ajay, Anand, Ashwin, Aravind, Balaji, Kabir, Karthik B, Karthik R, Kaushik, Lakshmanan, Narayanan, Rahul and Shankar- thanks for all the support and encouragement.

All this would not have been possible without the unconditional love, support and encouragement of my family. I would like to express my heartfelt gratitude to my mom, dad, brother Vinu and sister-in-law Varuna.

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CHAPTER 1

NUMERICAL INVESTIGATION OF TURBULENT FLOW AND HEAT TRANSFER IN A TWO-PASS SQUARE CHANNEL WITH DIFFERENT RIB CONFIGURATIONS

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Submitted to the International Journal of Thermal Sciences

ABSTRACT

In this study, the heat transfer and friction characteristics of four different rib geometries- 45° angled, V-shaped, W-shaped and M-shaped ribs in a two-pass stationary channel have been numerically investigated. The aspect ratio (Height to Width) of the cooling channel was 1:1 (square). The rib pitch-to-rib height ratio (p/e) and the rib-height-to-channel hydraulic diameter ratio (e/D_h) were 16 and 0.125 respectively. The Reynolds number was varied from 19500 to 69000. For the computations, the Reynolds averaged Navier–Stokes (RANS) equations were solved with the commercial software ANSYS Fluent using the realizable version of $k-\epsilon$ (RKE) model. The heat transfer results were benchmarked with experiments on a test rig with similar geometries and flow conditions. Detailed analysis of the flow characteristics in the two-pass channel was carried out so as to understand the interaction of the rib-induced secondary flows and the bend-induced secondary flows and their contribution to heat transfer enhancement. The heat transfer enhancement provided by V-shaped ribs was 7% higher than 45° ribs, 28% higher than W-shaped ribs and 35% higher than M-shaped ribs. However, the pressure penalty for V-shaped ribs was 19 % higher than 45° ribs, 24% higher than W-shaped ribs and 28% higher than M-shaped ribs. On comparing the overall thermal hydraulic performance, V-shaped and 45° ribs were observed to perform significantly better than W-shaped and M-shaped ribs.

NOMENCLATURE

D_h	Hydraulic diameter [m]
e	Rib height [m]
f	Friction factor
f_0	Friction factor for fully developed tube flow

f_s	Friction factor for smooth channel
h	Heat transfer coefficient [$\text{W/m}^2\text{K}$]
k	Thermal conductivity [W/mK]
L	Channel Length [m]
Nu	Nusselt number
Nu_0	Nusselt number from Dittus-Boelter equation
Nu_s	Nusselt number for smooth channel
P	Rib pitch [m]
p	Pressure [Pa]
p^*	Normalized pressure $(p-p_{in}) / (0.5\rho u_{in}^2)$
Pr	Prandtl number
q''	Heat flux [W/m^2]
Re	Reynolds number $(u_{in}D_h/\nu)$
s	Distance in streamwise direction along the two-pass channel [m]
T	Temperature [K]
T^*	Normalized temperature (T/T_{in})
T_{bulk}	Bulk fluid temperature [K]
T_{in}	Fluid temperature at inlet [K]
T_w	Wall temperature [K]
TKE	Turbulent kinetic energy [m^2/s^2]
TKE^*	Normalized turbulent kinetic energy (TKE/u_{in}^2)
u	Total fluid velocity [m/s]
u_m	Mean fluid velocity in the channel [m/s]
u^*	Normalized velocity (u/u_{in})
ρ	Density (kg/m^3)
ν	Kinematic viscosity (m^2/s)
η	Thermal hydraulic performance factor

INTRODUCTION

Modern gas turbines are operated at elevated turbine inlet temperatures, as high as 1600°C , which exceed the melting temperature of the super-alloy metals used in the construction of the hot

gas path components. Under such extreme conditions, the vanes and blades are subjected to high thermal stresses. It is therefore imperative to provide adequate cooling for the hot gas path components so as to prolong their life and ensure safe and reliable operation. A widely used method for cooling the blades involves passing coolant air bled from the compressor through serpentine internal passages inside the airfoils. The internal cooling passages consists of a series of straight ducts connected by 180° bends forming a multi-pass channel. The aspect-ratio and shape of the cooling channels depend on the aerodynamic profile and the portion of the vane where they are located. So as to enhance the heat transfer between the coolant and the hot walls, roughness elements or turbulence promoters (ribs) are installed on the walls of the serpentine passages. The ribs breakdown the laminar sub-layer and cause re-development of the boundary layer which helps in enhancing the heat transfer. When the ribs disturb the incoming boundary layer, they create local wall turbulence which facilitates the heat dissipation from the near-wall fluid to the mainstream owing to turbulent mixing. Although the complex flow field generated by the ribs provides significant heat transfer enhancement, the presence of these roughness elements results in pressure penalty. In the past, numerous experimental and numerical studies have been carried out to assess the thermal-hydraulic performance of the roughening features.

Han et al. [1] presented a comprehensive overview about internal cooling with various roughness elements like ribs, pin fins etc. Taslim et al. [2] experimentally investigated the heat transfer and friction characteristics of 90° , 45° , V-shaped and discrete angled ribs. They observed that 45° and V-shaped ribs provide higher heat transfer enhancement than discrete angled and 90° ribs. Ekkad and Han [3] used transient liquid crystal technique to obtain detailed Nusselt number distribution for a two-pass square channel with different rib turbulators. Lee et al. [4] experimentally investigated the heat transfer distribution in a high aspect ratio rotating ribbed channel with V-shaped and angled ribs. V-shaped ribs were observed to provide a higher heat transfer enhancement than angled ribs for both the stationary and rotating cases. Mochizuki et al. [5] studied the heat transfer and friction characteristics in a two-pass channel with different rib configurations. They concluded that the orientation of the ribs before and after the bend results in significant differences in the pressure drop and thermal performance of the two-pass channel due to interactions between the rib-induced and bend-induced secondary flows. Han et al. [6] carried out heat transfer and pressure measurements in square channels with parallel, crossed and V-shaped ribs. They reported that V-shaped ribs provide the best performance amongst the

configurations studied. Kumar and Amano [7] compared the performance of different arrangements of 60°-V and 60°-V broken ribs in a two-pass channel. Lamont et al. [8] compared the heat transfer characteristics of 90°, W-shaped and M-shaped ribs in a developing channel flow at a Reynolds number of 12000 using transient liquid crystal technique. Maurer et al. [9] investigated the heat transfer and pressure drop characteristics of V and W-shaped ribs for different rib pitch-to-rib height ratios. They observed that the thermal hydraulic performance of W-shaped ribs was better than V-shaped ribs. Wright et al. [10] investigated the thermal-hydraulic performance of different continuous and discrete ribs in a high aspect-ratio rotating channel with Reynolds number ranging from 10000 to 40000. They observed that discrete V-shaped and W-shaped ribs performed better than the other configurations studied.

Several numerical studies have been conducted to study the flow and heat transfer characteristics in two-pass ribbed cooling channels. Lin et al. [11] performed computations to study the flow and heat transfer in a smooth and ribbed U-shaped duct under rotating and non-rotating conditions using the $k-\omega$ SST turbulence model. They observed the interaction of rib induced secondary flows with the Dean-type secondary flows induced by the bend. Shih et al. [12] numerically investigated the three-dimensional flow and heat transfer in a ribbed U-shaped duct under rotating and non-rotating conditions using the $k-\omega$ SST model. In the absence of rotation, the heat transfer in the bend was observed to be dominated by the bend induced pressure gradients at low Reynolds number and by the rib-induced secondary flows at high Reynolds number. Al-Qahtani et al. [13] numerically predicted the flow and heat transfer for a two-pass rectangular channel with 45° rib turbulators using a multi-block RANS method under stationary and rotating conditions. Su et al. [14] performed computations to study the three-dimensional flow and heat transfer in a rectangular channel with 45° V-shaped ribs. Jia et al. [15] numerically studied the flow and heat transfer in a straight square duct with V-shaped ribs pointing upstream and downstream using the low-Reynolds number $k-\epsilon$ turbulence model. Bonhoff et al. [16] numerically studied the flow characteristics in a stationary square ribbed coolant channel with various turbulence models and validated it against PIV measurements. They observed that the results obtained with RSM model were most consistent with the experimental results amongst the models studied. Shevchuk et al. [17] investigated the heat transfer distribution in a two-pass varying aspect ratio channel with different divider wall-to-tip wall distances at a Reynolds number of 100,000. They reported good agreement between their computations and PIV measurements thereby

illustrating that RANS models are capable of predicting the flow and heat transfer in a two-pass channel reasonably. Similar studies were carried out by Siddique et al. [18] and Pape et al. [19] who investigated the effect of divider-to-tip wall distance on the heat transfer enhancement and pressure drop in a two-pass channel using realizable $k-\epsilon$ model with enhanced wall treatment.

The objective of the present study is to assess the thermal-hydraulic performance of four different rib configurations- 45° angled, V-shaped, W-shaped, and M-shaped using numerical simulations. Although V-shaped and 45° ribs have been well documented in the literature, studies pertaining to flow and heat transfer characteristics of W-shaped and M-shaped ribs are sparse. To the authors' knowledge, detailed flow and heat transfer analysis of the investigated rib configurations in a two-pass channel are limited. Furthermore, while most of the previous studies have focused on rib pitch-to-height ratios lesser than or equal to 10, this study is directed at exploring the performance of the different rib shapes at a large rib pitch-to-height ratio ($p/e = 16$). This could be helpful in designing two-pass ribbed channels for high Reynolds number flows (land based gas turbines) in which case the pressure penalty at low rib pitch-to-height ratios could be significant. Also, this study examines the performance of the different rib configurations in a developing channel flow which is common in internal cooling channels.

COMPUTATIONAL METHODOLOGY

In this section, an overview of the computational methodology including details about the computational domain, grid generation, solver, turbulence model and boundary conditions has been presented.

Computational Domain and Grid Generation

Figure 1 shows the computational domain which was used for the numerical simulations. The names of the channel walls and the planes used for visualizing the flow field in the subsequent discussions have been indicated in the figure. The two-pass channel had a uniform aspect ratio of 1:1. The pitch-to-rib height ratio (p/e) and the rib-height-to-channel hydraulic diameter ratio (e/D_h) were 16 and 0.125 respectively. The total channel length (L/D_h) was 23.5. There were 5 ribs each in the first and second pass with a 90° rib in the bend region. The first rib was located at a distance of $2.25 D_h$ downstream of the inlet which implies that the flow is still developing when it reaches the first rib. Note that only the bottom wall of the channel was roughened with ribs. The various

rib shapes investigated in this study have been shown in Fig. 2. The angle between the individual legs of the M-shaped, W-shaped and V-shaped ribs was 90° . The numbers marked on the M-rib configuration indicate the zones which were used in calculating the regionally averaged parameters.

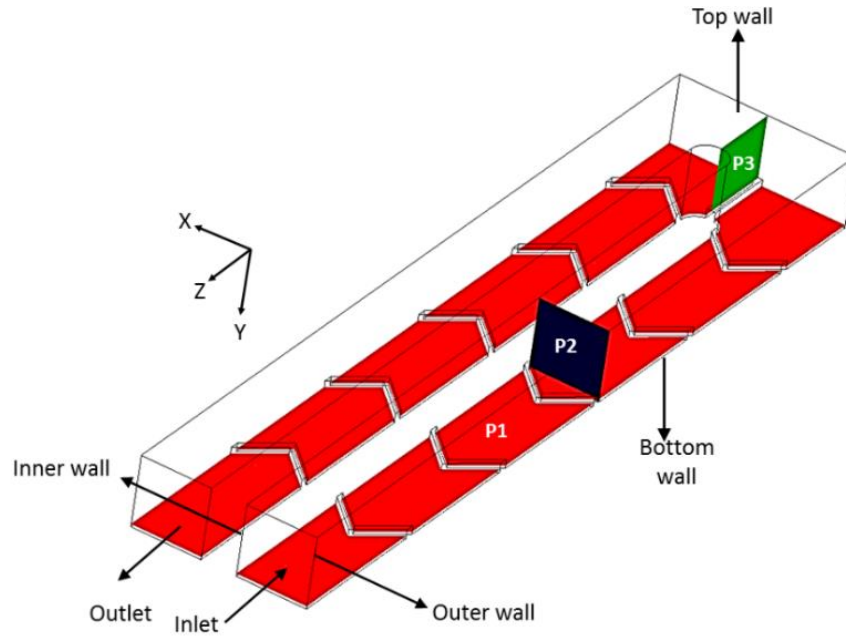


Figure 1: Computational domain

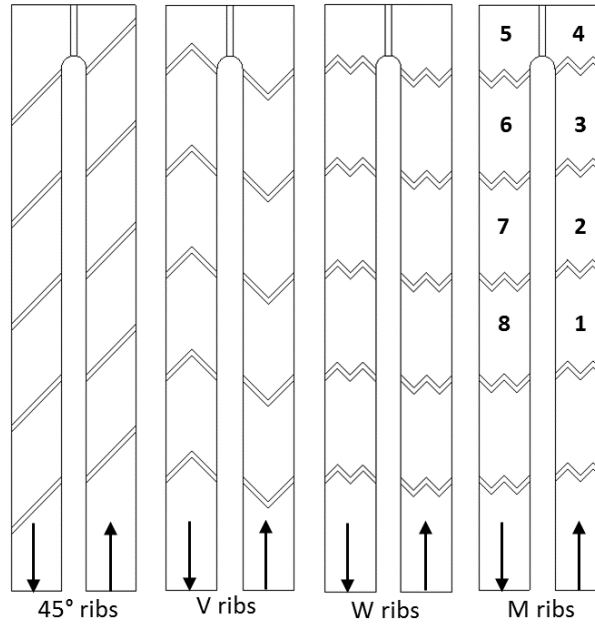


Figure 2: Rib configurations

A hybrid unstructured mesh consisting of tetrahedrons, pyramids and prism elements was generated using ANSYS CFX meshing software. Figure 3 shows the computational grid which was used for the study. Using the sizing functions, a high mesh density was maintained in the bend and regions close to the bottom wall and ribs. Similar sizing functions were used for the different configurations as to ensure that the grid generated was uniform for all the cases. The dimensionless wall distance y^+ was maintained in the range 1-3 which is a requirement for the near wall treatment used in this study. The number of prism layers was around 12 with a growth rate of 1.1.

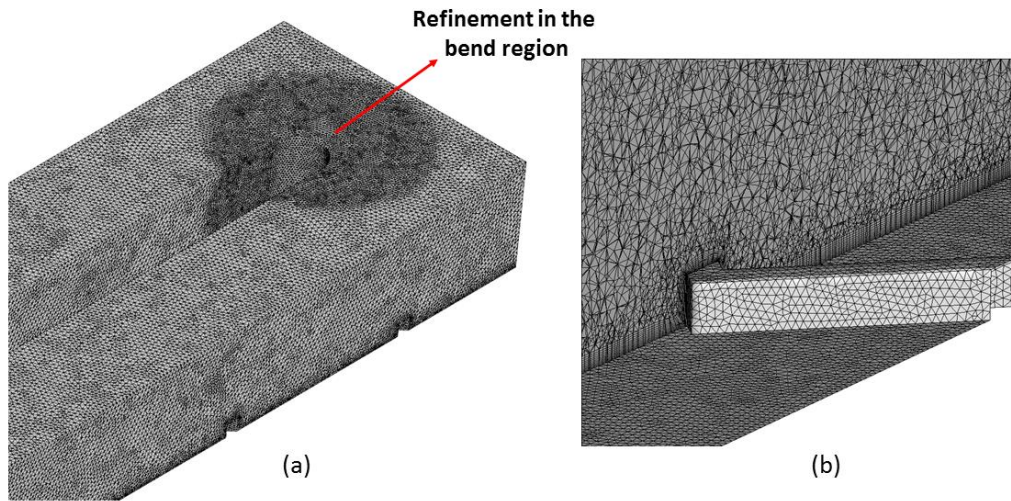


Figure 3: (a) Numerical grid (b) Cut-section view of the mesh

Solver and Turbulence Model

For the numerical simulations, three-dimensional CFD analysis was carried out using the commercial solver ANSYS *Fluent* 14.5 which is a finite volume based solver. Steady state Reynolds Averaged Navier-Stokes (RANS) equations were solved using second order discretization schemes. The working fluid was assumed to be incompressible and hence the pressure-based solver was used. SIMPLE algorithm was used for pressure-velocity coupling. Past studies have shown that RANS models are capable of predicting the flow and heat transfer in a two-pass channel quite accurately. The realizable k - ϵ model has been found to perform well with regard to predicting separated flow behind ribs and capturing the flow physics in the bend region

which is critical for this study. Moreover, several studies have reported that realizable k- ϵ model is capable of accurately predicting the pressure loss and heat transfer in a two-pass channel [17,18,20-22]. Therefore, the turbulent flow field in this study was modeled using realizable version of k- ϵ model with enhanced wall treatment.

Grid Independence Study and Convergence Criteria

In order to ensure that the solution obtained was independent of the mesh size, a grid independence study was carried out for one rib configuration (45°) at the highest Reynolds number. Three different grid systems consisting of 5 million, 7.5 million and 11.5 million elements were considered. For grid independence study, the overall normalized Nusselt number (Nu/Nu_0) was compared and the results are shown in Fig. 4. Between the finest and coarsest mesh, a difference of 2.3% was observed in the monitored parameter. So as to maintain a balance between computational economy and accuracy, a grid system with 7.5 million elements was chosen for all the computations. Convergence was deemed to have been reached if the residuals were smaller than 10^{-5} for the continuity, momentum, turbulent transport equations and 10^{-8} for the energy equation. In addition, the average temperature of the bottom wall was monitored to ensure that a steady, converged solution was obtained.

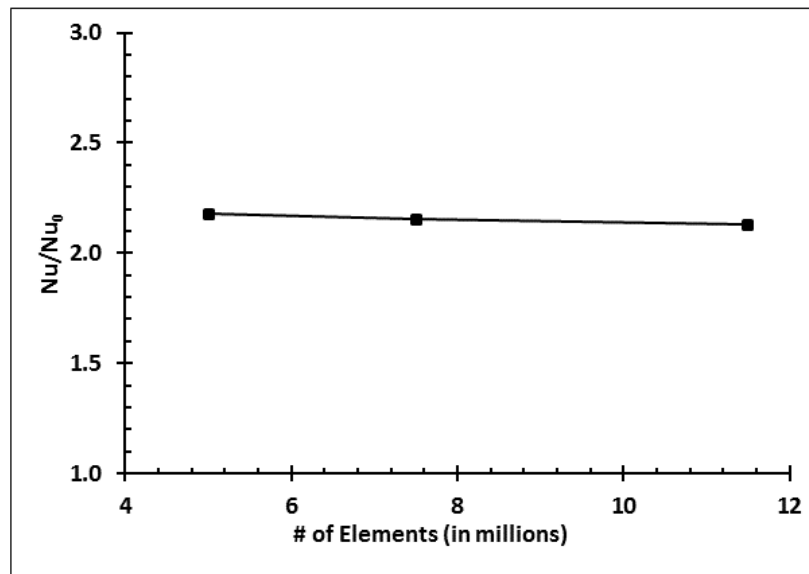


Figure 4: Overall average Nusselt number enhancement for different grid systems

Boundary Conditions

The working fluid (air) was assumed to be incompressible with constant fluid properties. The flow was assumed to be steady, three-dimensional and turbulent. Uniform velocity and temperature was specified as the inlet boundary conditions. The velocity at the inlet was calculated based on the Reynolds number referred to the channel hydraulic diameter and entrance flow conditions. A turbulence intensity of 5% and a length scale equal to 10% of the channel hydraulic diameter was specified at the inlet. Pressure boundary condition was imposed at the outlet, set to zero gauge. For all the cases, a constant heat flux of 5000 W/m² was imposed on the bottom wall and ribs. Note that the Nusselt number enhancement due to the ribs was calculated only on the bottom wall. No-slip, adiabatic boundary condition was imposed on the other walls in the domain.

DATA REDUCTION

The local heat transfer coefficient on the ribbed wall was calculated from the wall heat flux, surface temperature and the local bulk fluid temperature as

$$h = \frac{q''}{(T_w - T_{bulk})} \quad (1)$$

In order to determine the local bulk fluid temperature, the mass averaged fluid temperature was calculated at a number of cross-sections in the streamwise direction along the length of the channel. A linear fit was then used to compute the local bulk fluid temperature.

The Nusselt number was calculated as

$$Nu = \frac{hD_h}{k} \quad (2)$$

The calculated Nusselt number was normalized by the Nusselt number for fully developed turbulent flow in a smooth circular tube obtained from the Dittus-Boelter correlation (Nu_0).

$$Nu_0 = 0.023Re^{0.8}Pr^{0.4} \quad (3)$$

The Fanning friction factor was computed using Eq. (4) and normalized by the friction factor for fully developed turbulent flow in a smooth tube obtained from Blasius equation (f_0).

$$f = \frac{\Delta P D_h}{2\rho u_{in}^2 L} \quad (4)$$

$$f_0 = 0.079 Re^{-0.25} \quad (5)$$

The thermal hydraulic performance factor of the ribbed channel was computed with respect to both standard correlations (η_0) and smooth channel results (η_s) using Eq. (6) and Eq. (7) respectively.

$$\eta_0 = \frac{\frac{Nu}{Nu_0}}{\left(\frac{f}{f_0}\right)^{1/3}} \quad (6)$$

$$\eta_s = \frac{\frac{Nu}{Nu_s}}{\left(\frac{f}{f_s}\right)^{1/3}} \quad (7)$$

RESULTS AND DISCUSSION

In this section, detailed discussions about the flow field and heat transfer characteristics of smooth and ribbed two-pass channels have been presented. For validation, the flow and heat transfer data from the simulations have been compared against experiments performed in the lab on similar geometries on the same test rig as used by Lamont et al. [8]. The location of the planes (P1, P2 and P3) chosen for visualizing the flow field have been indicated in Fig. 1. Plane P1 was parallel to the flow direction and located at a distance of 4% of hydraulic diameter above the bottom wall. Plane P2 was perpendicular to the flow direction and located immediately downstream of the 3rd rib in the 1st pass. Plane P3 was located in the middle of the bend perpendicular to the flow direction. In this study, all the results have been normalized with both standard correlations and smooth channel results so as to isolate the individual contribution of the bend and the ribs towards heat transfer enhancement and pressure drop.

Validation with Experiment

For the validation of numerical results, the static pressure and Nusselt number distribution in the two-pass channel were compared with the preliminary experimental results. Figure 5 shows the comparison of normalized pressure distribution in the two-pass channel for the different rib configurations at a Reynolds number of 35500. It can be seen that there is a good overall agreement in trends between the numerical and experimental results. In the 1st pass, where the flow physics is not very complicated, it can be observed that there is a very good agreement between the CFD and test data for all the cases. Even though CFD captures the trend in the bend region and the 2nd pass, it under-predicts the pressure. This could be because of the inability of the turbulence model to accurately predict the complex flow physics in the bend region. Downstream of the bend, there is a reasonable agreement between the predicted results and the test data.

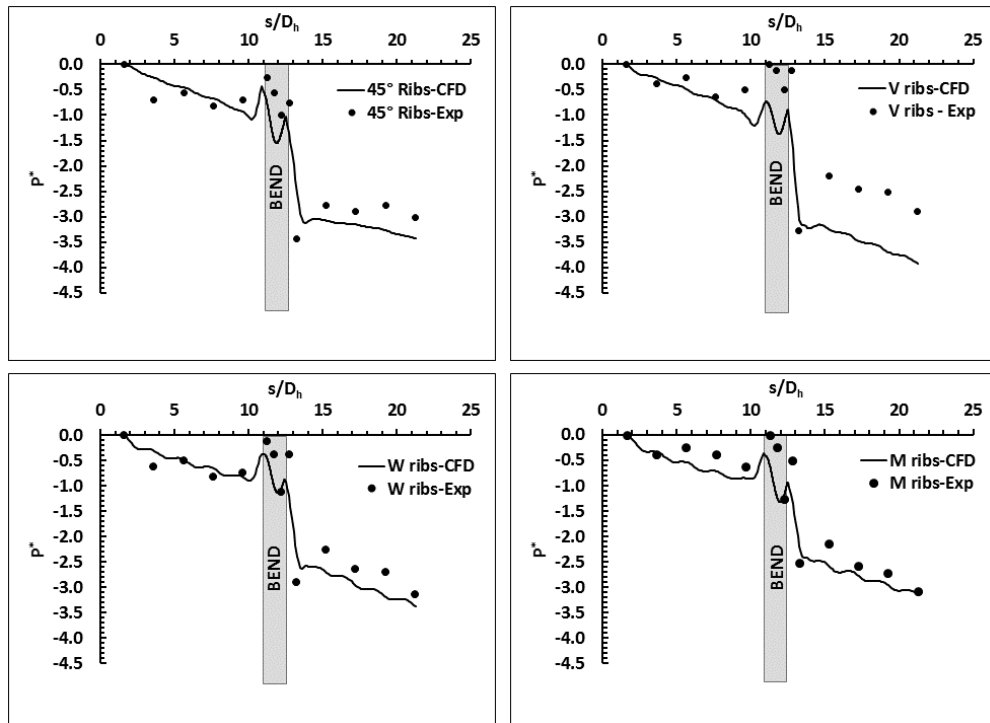


Figure 5: Validation of flow in two-pass channel

Comparison of the normalized Nusselt number contours (Nu/Nu_0) for 45° ribs at a Reynolds number of 35500 has been shown in Fig. 6. The qualitative features in the heat transfer distribution have been reproduced well by the simulations. There is a good match in the heat transfer

distribution pattern in the inter-rib region between simulations and experiments. The numerical simulations' prediction in the 2nd half of the bend and the regions further downstream are good.

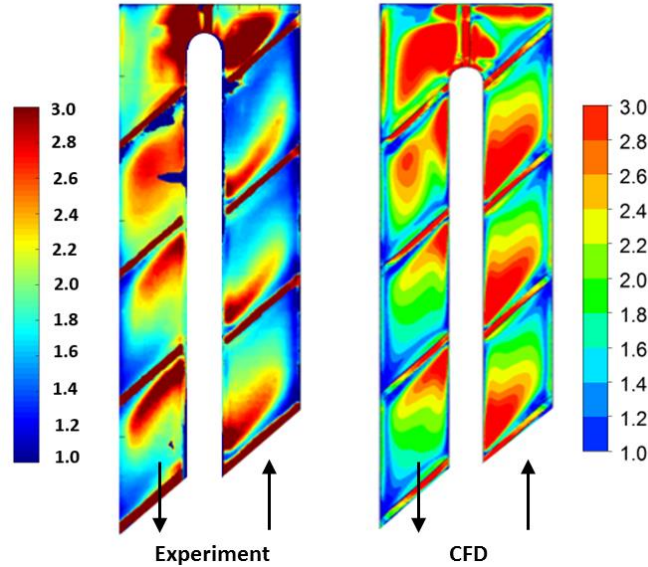


Figure 6: Comparison of Nusselt number distribution between experiment and CFD

The comparison of the distribution of Nusselt number normalized with smooth channel results (Nu/Nu_s) has been shown in Fig. 7. Looking at the Nu/Nu_s distribution, we observe that the simulations agree qualitatively with the experiments. An average difference of 22% was observed between the simulation and the experiments. It must be noted that, unlike the numerical simulations, the velocity profile at the inlet was not uniform in case of the experiments. As a result of the experimental inlet conditions (90° bend upstream of the inlet), the velocity profile of the flow entering the test section was highly turbulent and skewed, resulting in higher baseline heat transfer in the 1st pass. This resulted in lower enhancement because of the ribs in the first pass in case of experiments compared to CFD. Past studies [23-25] have reported that the simple eddy viscosity models are incapable of accurately predicting the complex flow structure and turbulence anisotropy in the bend and regions downstream of the bend which in turn results in under prediction of the Nusselt number enhancement. The overall agreement between experiments and simulations was considered to be acceptable and hence it was decided to proceed with the chosen turbulence model.

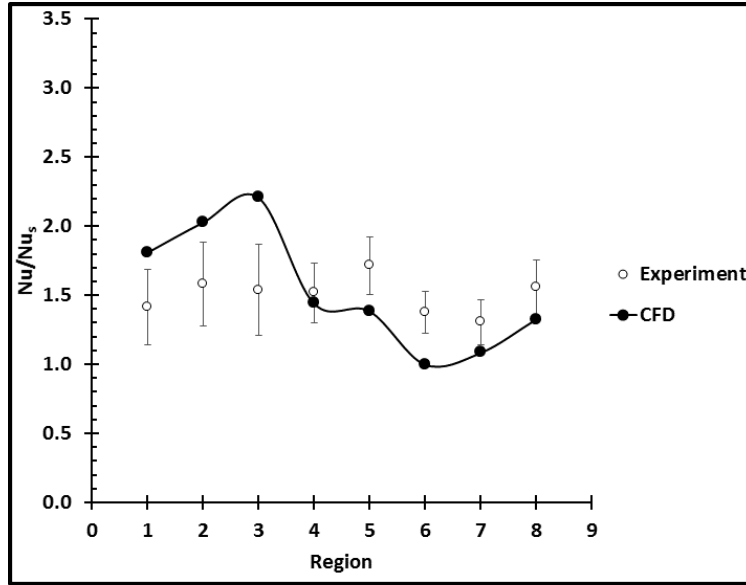


Figure 7: Comparison of regionally averaged Nusselt number

Smooth and Ribbed Two-pass Channel: Flow Characteristics

In order to study the flow and heat transfer characteristics of ribbed two-pass channels, it is important to understand the innate behavior of the rib induced secondary flows for the different rib shapes. Figure 8 shows the secondary flows induced by different rib shapes and the flow structure in the inter-rib region close to the bottom wall. The stream traces shown on the bottom of the figure have been projected on the P1 plane while the ones on top have been projected on the P2 plane.

In case of the 45° ribs, a single secondary flow vortex is produced. The sense of rotation of the secondary flow vortex depends on the orientation of the ribs with respect to the bulk flow. Also, two symmetric vortices would be generated if the ribs were present on the top wall as well. The secondary flow transports the cooler fluid from the core towards the inner wall which then travels along the ribbed wall towards the outer wall. Examining the streamtraces close to the ribbed wall, it can be seen that the coolant flows parallel to the rib from the inner wall to the outer wall. As a result, there are steep temperature gradients along the secondary flow direction owing to the development of thermal boundary layer. The secondary flow in case of V-shaped ribs can be better understood by considering the V-shaped ribs to be made up of two 45° ribs joined at the center of the channel. So, V-shaped ribs produce a counter rotating vortex pair. Streamtraces close to the bottom wall show that following the reattachment downstream of the ribs, the flow splits into two

streams in the form of a “V”, moving from the center of the passage towards the side walls. Since the secondary flows are symmetric about the center of the channel, V-shaped ribs produce a uniform heat transfer enhancement (discussed later) compared to 45° ribs.

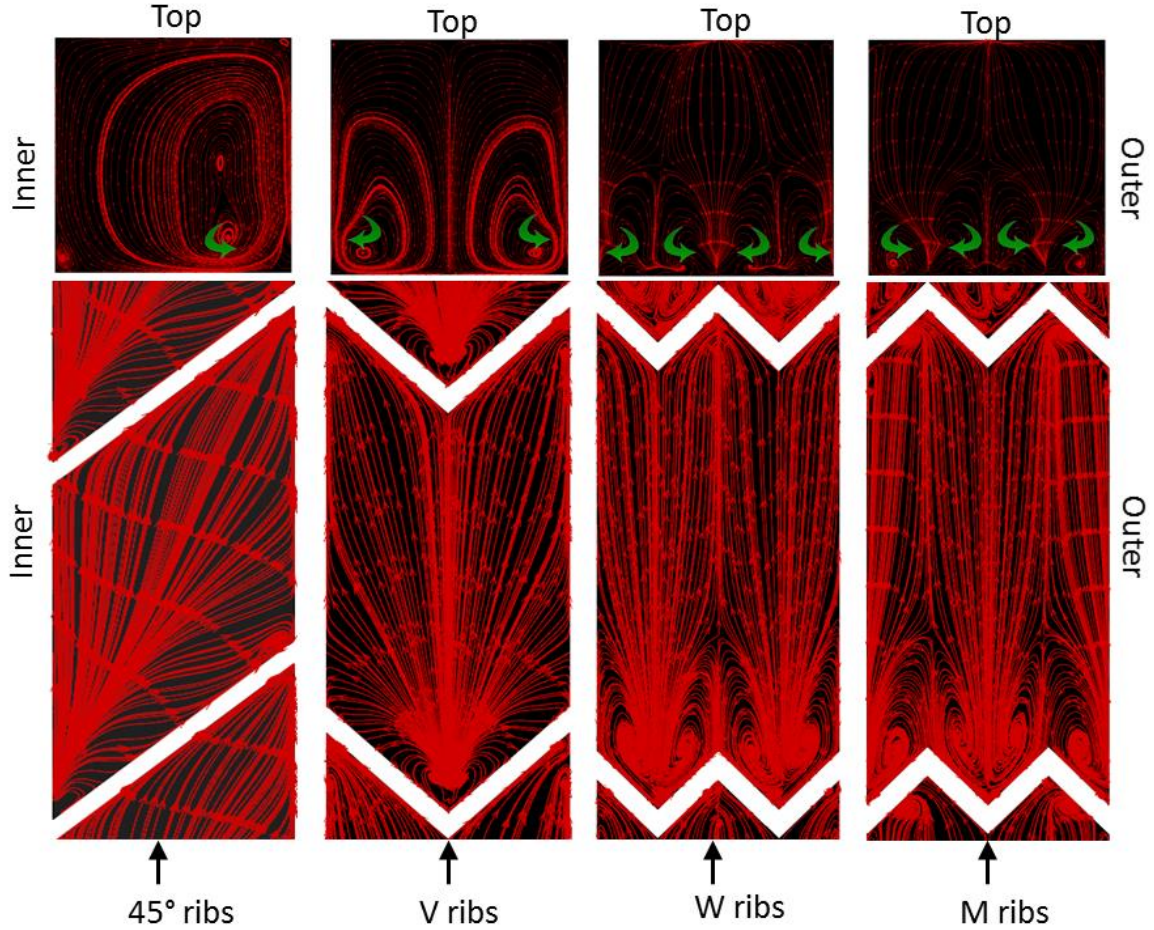


Figure 8: Rib-induced secondary flow field

W-shaped ribs can be visualized as two V-shaped ribs joined at the center of the channel and hence produce two counter rotating vortex pairs. Similarly M-shaped ribs can be visualized to be made up of two inverted V-shaped ribs connected at the center with the apex facing downstream. Recirculation can be seen downstream of the W-shaped ribs and the M-shaped ribs which correspond to regions of low heat transfer. The recirculating flows caused by the vortex pair downstream of the ribs tend to bring down the pressure penalty as well as the heat transfer enhancement [26]. Furthermore, it can be noted that the size of the secondary vortex is maximum for 45° ribs followed by V-shaped ribs. This secondary vortex is significantly smaller for W-

shaped and M-shaped ribs. The larger the size of the secondary vortex, the greater the amount of turbulent mixing and associated heat transfer.

Previous studies have shown that the bend region significantly contributes to the overall heat transfer enhancement in a two-pass channel [27]. It is therefore important to study how the presence of ribs influences the flow field in the bend region. Figure 9 shows the evolution of the flow field in the two-pass channel. Stream traces have been projected on slice planes (normal to the direction of bulk flow) at different streamwise locations along the two-pass channel. An enlarged view of the secondary flow generated due to the interaction between the bend and the ribs (on plane P3) has been shown in Fig. 10.

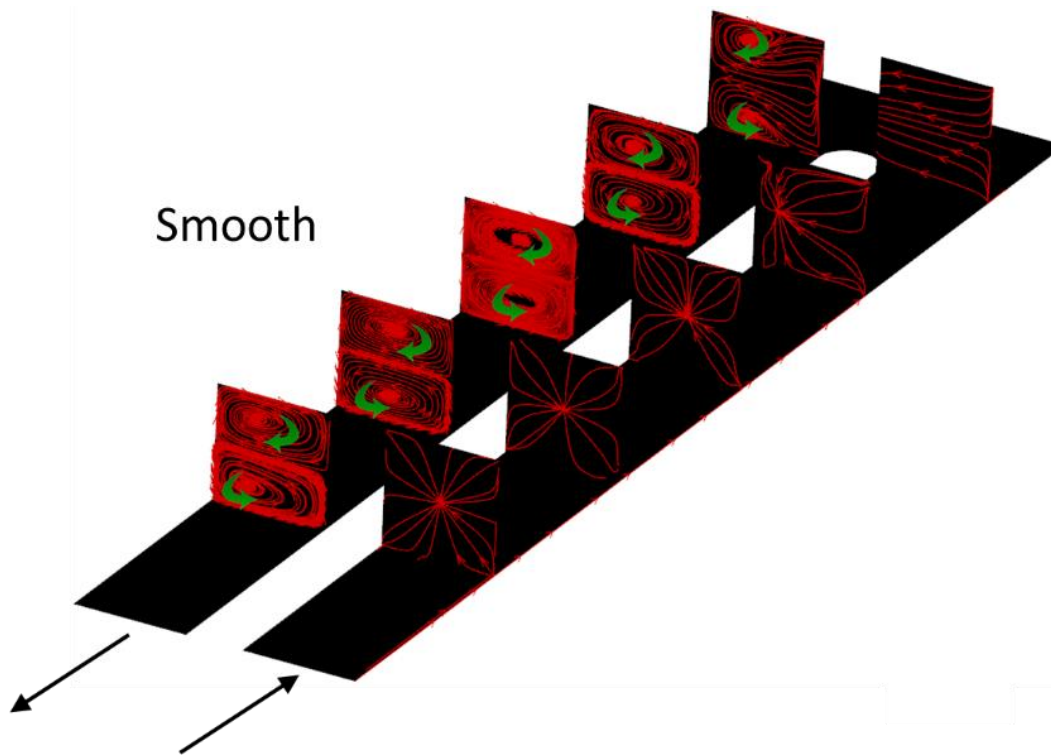


Figure 9: Evolution of flow field in smooth and ribbed two-pass channels

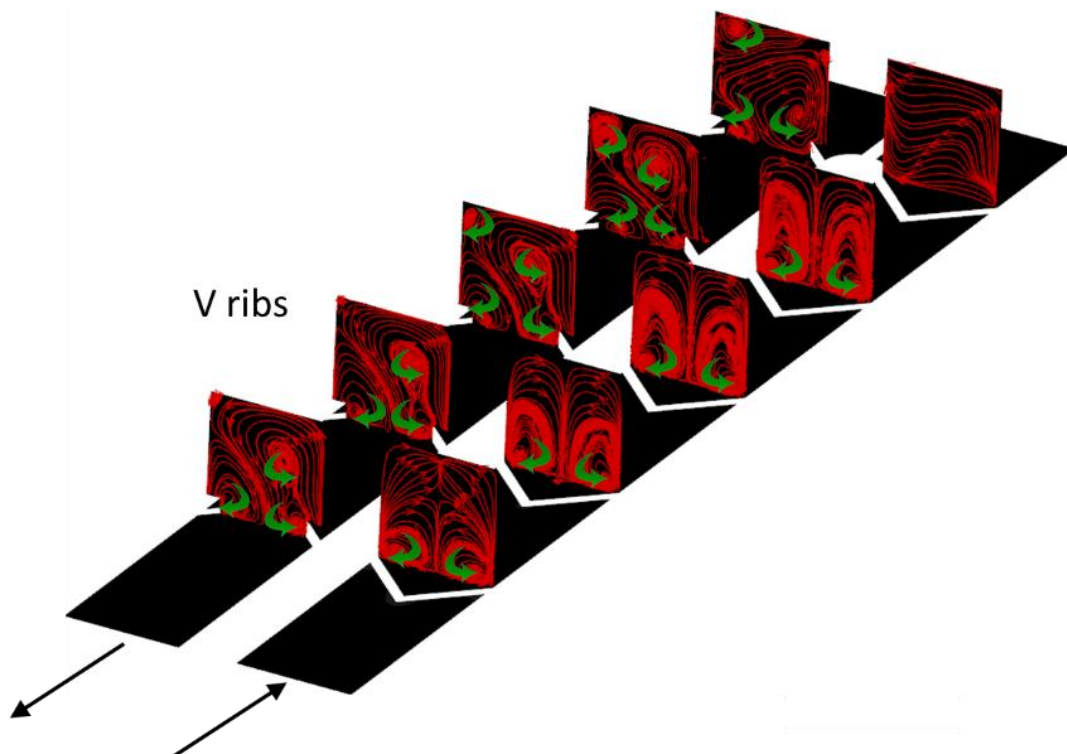
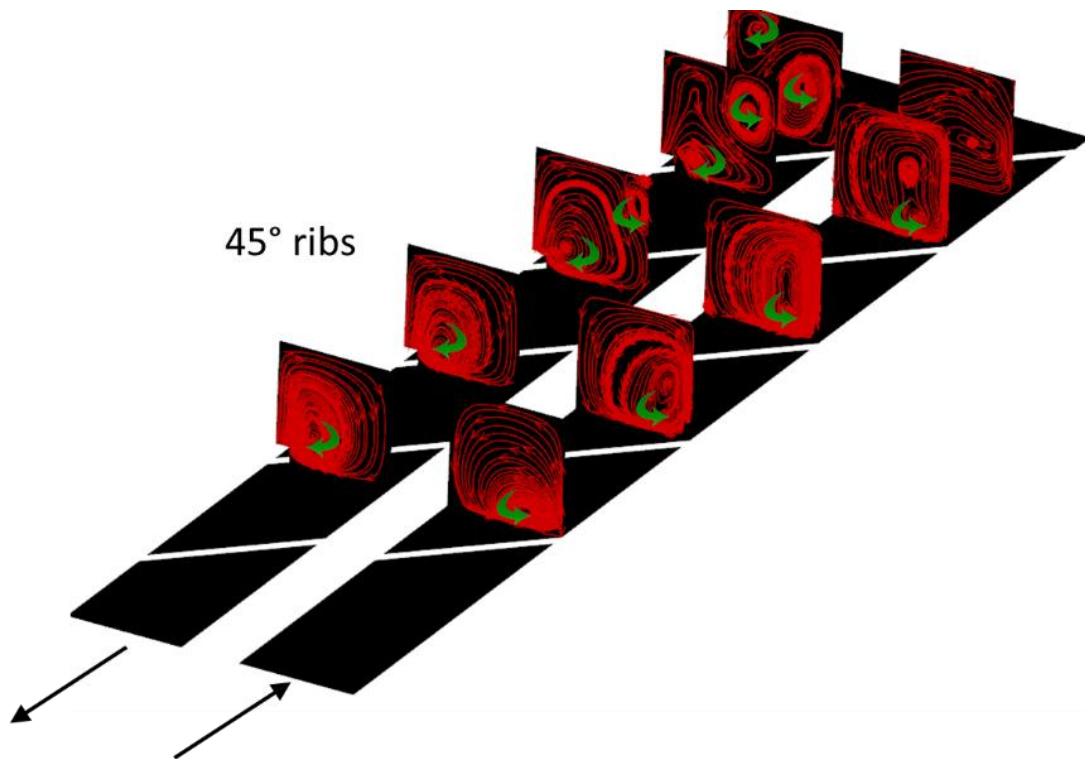


Figure 9: (Continued)

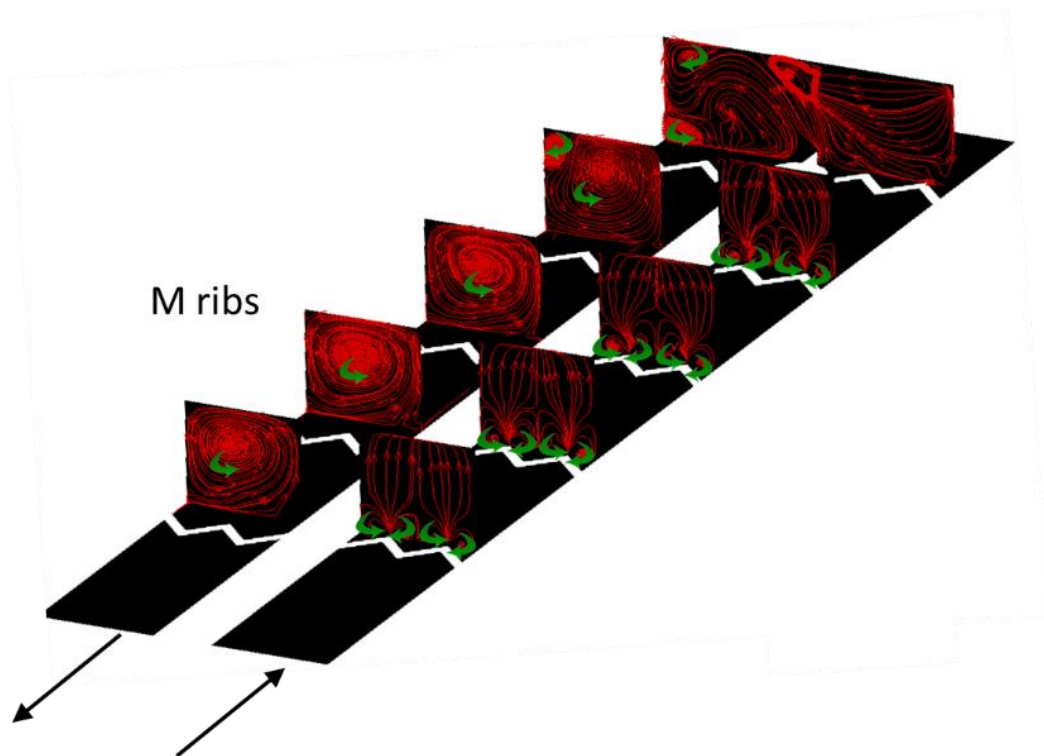
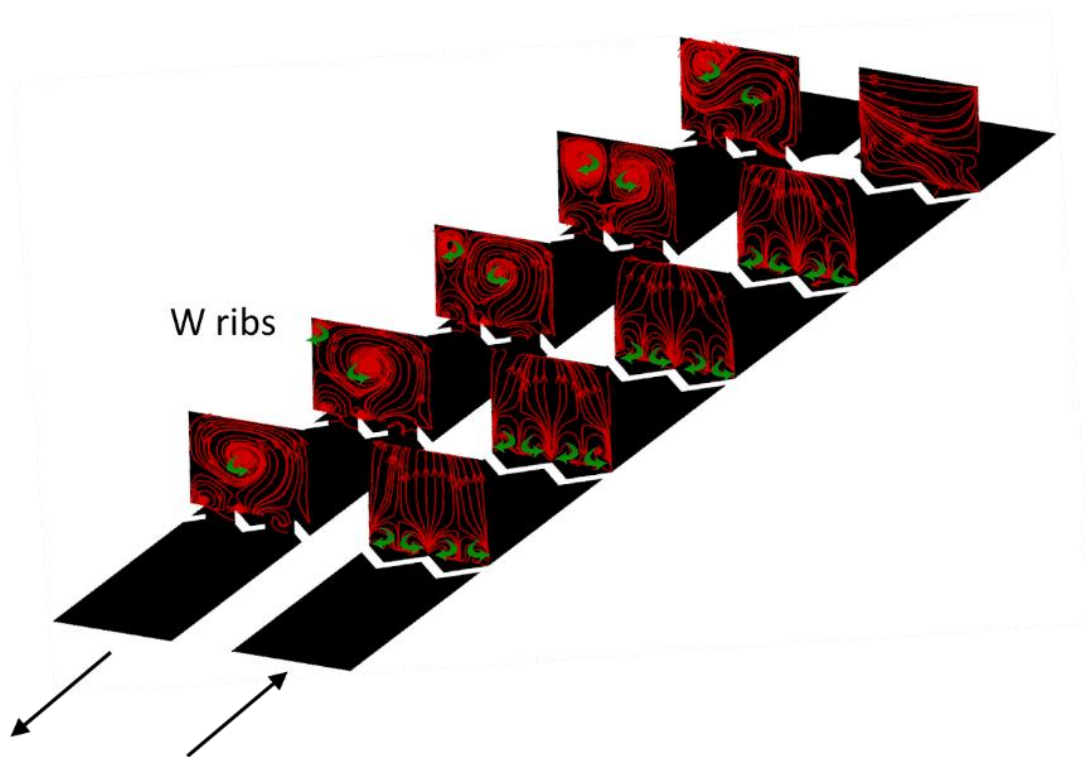


Figure 9: (Continued)

The bend affects the flow distribution in the 1st pass couple of hydraulic diameters upstream of the turn for the smooth channel (Fig. 9). Near the entrance to the bend, there is a strong fluid motion from the outer to the inner wall. Dean-type secondary vortices are formed in the bend region due to a combination of centrifugal force and curvature induced pressure gradient. They transport the fluid from the center of the channel to the outer wall along the mid-plane as shown in Fig. 10. The Dean-type vortices can even be seen significantly downstream of the bend in the 2nd pass.

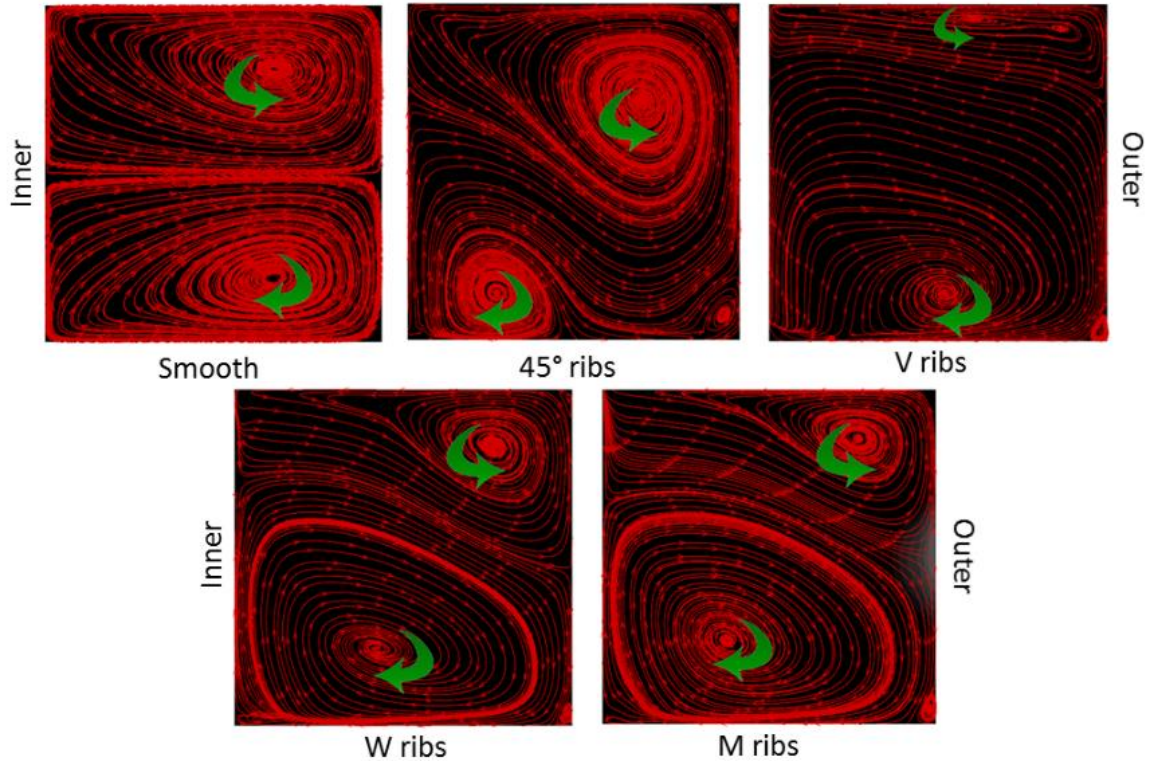


Figure 10: Secondary flow in the bend region

All the ribbed two-pass channels in this study feature a 90° rib in the middle of the bend. The flow-field in the bend region is influenced by the 90° rib, curvature of the bend and the shape of the ribs upstream and downstream of the bend. In case of the 45° ribs, the strength of the rib-induced vortices increases along the 1st pass. Downstream of the last rib, there is a strong fluid motion from the outer wall to the inner wall because of the turn which overpowers the rib-induced secondary flow. The Dean-type vortex near the bottom wall becomes smaller, with its core shifted close to the inner wall. This is because the flow field downstream of the last rib in the 1st pass

opposes the Dean-type vortex close to the bottom wall. However, the Dean-type vortex in the upper half of the channel is reinforced by the rib-induced secondary flow and is larger in size compared to the smooth channel case. Along the 2nd pass, the rib-induced secondary flows start to dominate and the Dean-type vortices are completely suppressed after the first couple of ribs.

Similar to 45° ribs, the strength of the secondary vortex generated by V-shaped ribs increases along the first pass. Downstream of the last rib, the bend induced fluid motion overpowers the rib induced secondary flows. The Dean-type vortex near the bottom wall is reinforced, with its core shifted close to the center. It can be noticed that the Dean-type vortex completely overpowers the secondary vortex generated by the 1st rib downstream of the bend close to the inner wall. The flow field downstream of the ribs in the 2nd pass is completely different compared to the 1st pass due to the interaction between the Dean-type vortices and rib induced secondary flows. The stronger Dean-type vortex becomes more pronounced in the 2nd pass due to the interaction with the rib-induced secondary flows. The weaker Dean-type vortex is suppressed towards the end of the 2nd pass.

In case of W-shaped ribs, the size of the two counter rotating vortex pairs (CRVPs) doesn't change significantly along the 1st pass. The flow near the bend entrance is similar to 45° and V-shaped ribs. The Dean-type vortex near the bottom wall is reinforced, with its strength shifted towards the channel center. Along the 2nd pass, the rib-induced counter rotating vortex pairs can be seen on the top of the ribs. Furthermore, it appears as if the stronger Dean-type vortex is further reinforced, with its center shifted away from the bottom wall. At the same time, the weaker vortex close to the outer wall is suppressed.

The flow field in the 1st pass for M-shaped ribs is similar to the W-shaped ribs. The Dean-type vortex near the bottom wall is reinforced near the bend. Along the 2nd pass, the Dean-type vortex near the bottom wall is further augmented whereas the vortex close to the top wall is completely suppressed. Furthermore, since the planes in the 2nd pass are located significantly downstream of the ribs, the weak rib induced secondary flows close to the bottom wall can be hardly seen. Thus, except for the 45° ribs, the Dean-type vortices significantly influence the flow and heat transfer characteristics in the 2nd pass for the other rib configurations.

Figures 11 and 12 show the temperature and velocity contours superimposed with velocity vectors in the bend region on the P1 plane. From the velocity contour for the smooth channel, it can be seen that the flow distribution in the first pass is affected by the bend couple of hydraulic

diameters upstream of the bend. The region of maximum velocity across the channel shifts towards the inner wall. Flow reversal can be seen towards the end of the first pass near the bend region. Along the bend, the flow decelerates close to the outer wall and accelerates close to the inner wall. A small separation region exists near the end of the bend in the 2nd pass. In the 2nd pass, the Dean-type secondary vortices transport the cooler fluid in the channel towards the outer wall as can be seen in Fig. 11. After the flow turns around the bend, it impinges on the outer wall and accelerates along it (Fig. 12). These observations are consistent with existing literature [11,12,20,28].

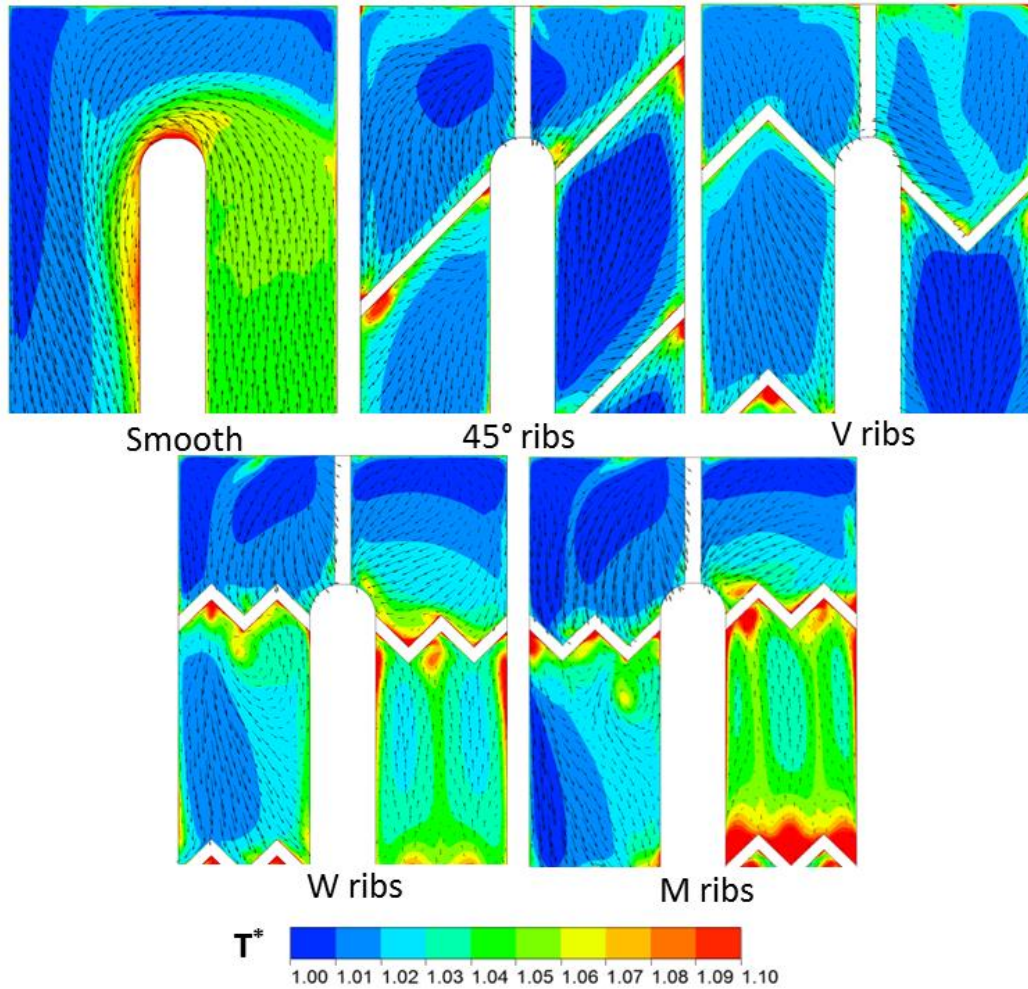


Figure 11: Temperature contours superimposed with velocity vectors close to the bottom wall

For the ribbed channels, we see that the flow near the bottom wall downstream of the last rib no longer follows the path shown in Fig. 8. This is because the bend overpowers the effect of the

ribs. The flow in the bend region is similar for M and W-shaped ribs. For the M, W and V-shaped ribs, the effect of the bend can be observed even downstream of the first rib in the 2nd pass since the flow near the bottom wall moves from the outer wall to the inner wall similar to the baseline case. However, in case of 45° ribs, the rib effect overpowers the curvature effect since the spiral-like streamwise flow pattern inherent to 45° ribs can be seen immediately downstream of the first rib in the 2nd pass. From the temperature contours, it can be seen that the fluid close to the outer wall in the 2nd pass is at a lower temperature for the M and W-shaped ribs which could be because of the dominant effect of the Dean-type vortices. In case of M and W-shaped ribs, recirculation can be seen immediately downstream of the 1st rib which corresponds to low heat transfer region.

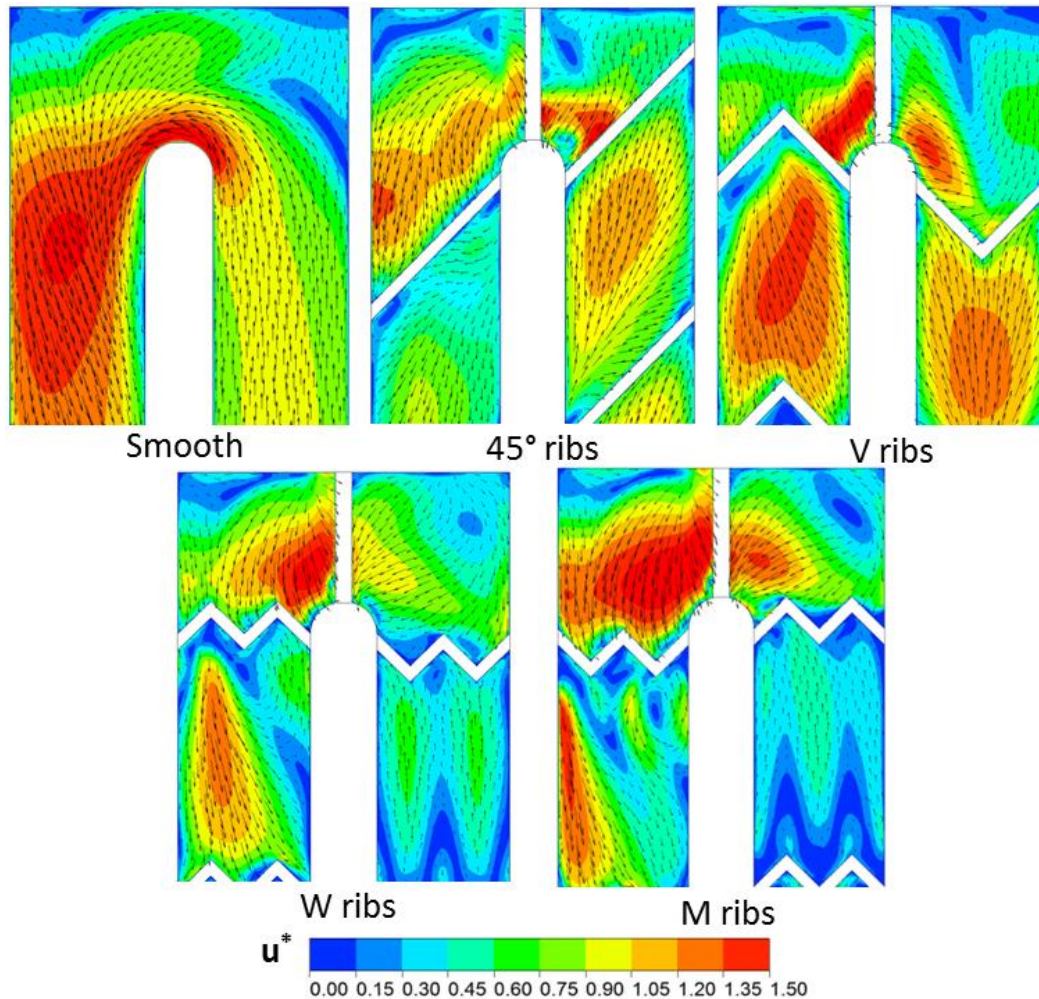


Figure 12: Velocity contours superimposed with velocity vectors close to the bottom wall

Figure 13 shows the normalized turbulent kinetic energy (TKE) contours on plane P1 for the different configurations at a Reynolds number of 35500. The TKE contours help in understanding the heat transfer characteristics of different rib configurations since the mixing caused by turbulence contributes significantly to the heat dissipation from the surface. In general, higher turbulent kinetic energy corresponds to higher heat transfer rate. Higher turbulent kinetic energy can be observed in the bend region for both the smooth and ribbed channels. For all the rib configurations, the TKE is higher in the reattachment region and lower in the recirculation region. The high TKE downstream of the 1st V-shaped rib in the 2nd pass could be due to the combined effect of rib-induced and bend-induced secondary flows. It can also be noted that the turbulent kinetic energy is high in the regions where the secondary flows impinge on the bottom wall and low in the regions where the secondary flow leaves the bottom wall [11].

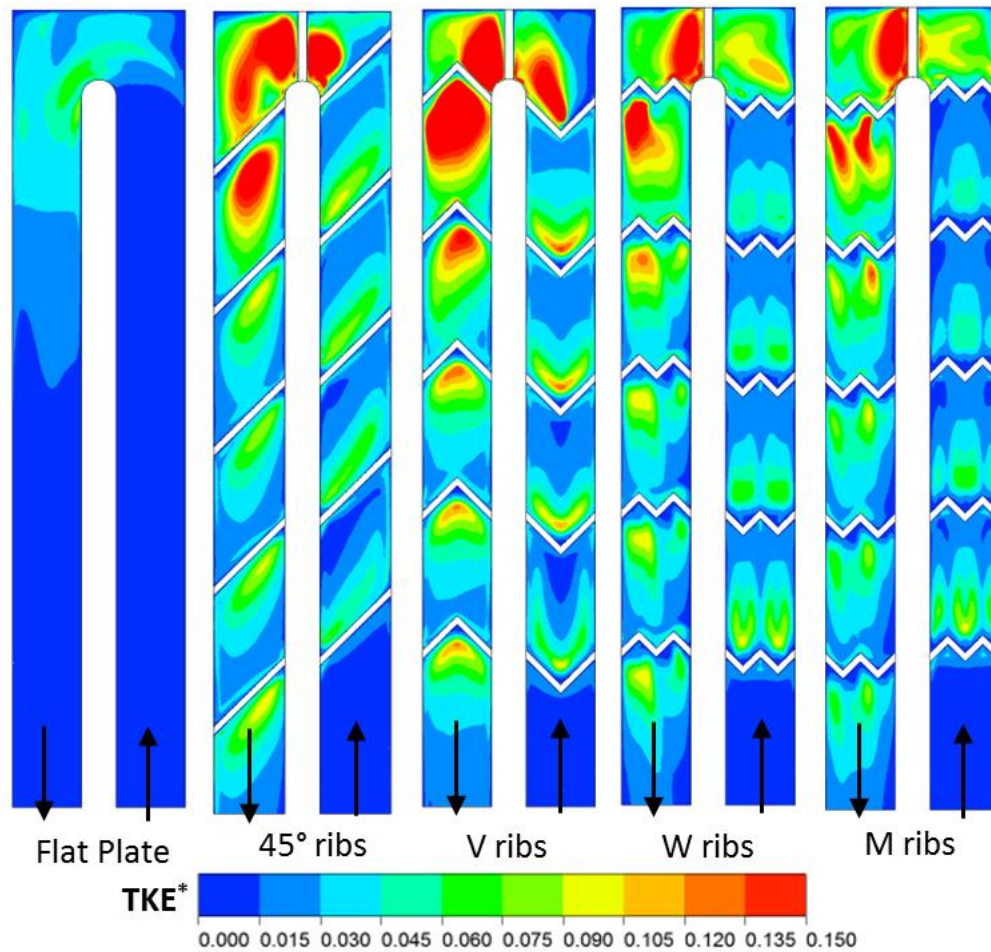


Figure 13: Distribution of turbulent kinetic energy close to the bottom wall

Smooth and Ribbed Two-pass Channel: Heat Transfer Characteristics

Figure 14 shows the normalized Nusselt number contours for the smooth and ribbed two-pass channels at a Reynolds number of 35500. The observed Nusselt number distribution can be correlated with the flow characteristics discussed in the previous section. For the smooth channel, the Nusselt number decreases along the 1st pass as the flow gets developed and approximately equals the fully developed flow Nusselt number obtained from the Dittus-Boelter correlation ($Nu/Nu_0 \sim 1$) close to the entrance of the bend. Near the bend, there is a significant enhancement because of the curvature induced secondary flows and impingement on the outer wall. The Nusselt number increases by a factor of two in the bend region. The Nusselt number increases by a factor of two in the bend region. Due to a combination of higher velocity and lower fluid temperature, the heat transfer is high close to the outer wall in the 2nd pass. Downstream of the bend, the enhancement in the 2nd pass gradually drops.

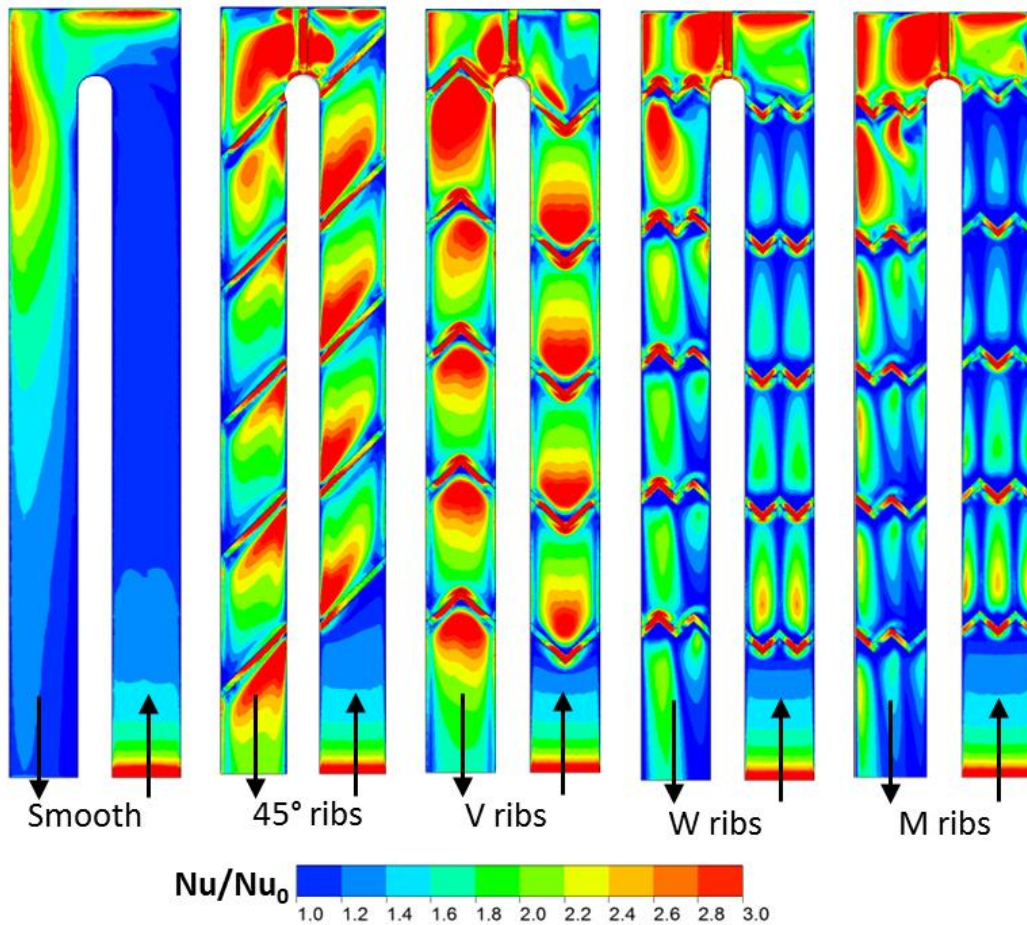


Figure 14: Normalized Nusselt number distribution on the bottom wall

For the ribbed channels, the flow is not fully developed when it reaches the first rib which is common in case of turbine internal cooling channels. In case of 45° ribs, the maximum enhancement on the rib occurs close to the inner wall since the flow reaches there first. Along the rib, the enhancement decays on account of boundary layer development. Downstream of the ribs, the maximum enhancement occurs close to the inner wall at the reattachment point. The flow near the ribbed surface moves from the inner to the outer wall parallel to the ribs. As a result, the enhancement in the inter-rib region gradually decays in the spanwise direction from the inner to the outer wall. Along the 1st pass, the enhancement increases because the rib induced secondary vortices grow in strength. High heat transfer region can be observed in the bend region close to the outer wall because of flow impingement. In the previous section, it was observed that the rib induced vortices suppress the Dean-type vortices in the 2nd pass. Therefore, the heat transfer distribution in the 2nd pass is very similar to the 1st pass since the rib-induced vortices dominate in both passes.

From the Nusselt number contour for V-shaped ribs, it can be observed that the enhancement in the spanwise direction is much more uniform than 45° ribs. This is because of the symmetric secondary vortices induced by the V-shaped ribs. The enhancement in the inter-rib region decreases in the streamwise direction because of boundary layer development. Along the ribs, the enhancement decreases from the center towards the inner/outer walls. High enhancement region is observed close to the apex of the V-shaped ribs because of the turbulence caused by flow impingement. Downstream of the apex, there is low heat transfer enhancement because of flow separation. As already discussed, the strength of secondary vortex pair induced by V-shaped ribs increases along the 1st pass which results in an increase in heat transfer enhancement [4]. Downstream of the last rib in the first pass, the Nusselt number contour is skewed towards the bend. It should be noted that the high heat transfer region due to impingement is absent. The enhancement in the 2nd half of the bend is less compared to 45° ribs. A high enhancement region is observed downstream of the 1st rib in the 2nd pass possibly due to the combined effect of bend induced and rib induced secondary flows.

In case of W-shaped ribs, a large separation region exists downstream of each rib. Within each rib pitch, two streaks of high enhancement regions can be observed downstream of the ribs. It can be seen that reattachment occurs significantly downstream of the rib. Furthermore, the enhancement in the streamwise direction decreases in both the 1st and the 2nd pass contrary to 45°

and V-shaped ribs. A high heat transfer region can be observed downstream of the first rib in the 2nd pass close to the outer wall owing to a combination high velocity and low near-wall fluid temperature. Throughout the 2nd pass, the enhancement close to the outer wall is higher than the enhancement close to the inner wall similar to the baseline case. The Nusselt number distribution for M-shaped ribs is very similar to W-shaped ribs. Here again, a large separation region can be observed downstream of the ribs. Similar to W-shaped ribs, the Nusselt number enhancement decays in the streamwise direction for both the passes and a high heat transfer region can be observed close to the outer wall in the 2nd pass. Low enhancement region is observed upstream of the ribs since the vortices from the previous rib die down before reaching the next rib. For both the M and W-shaped ribs, high enhancement can be seen in the second half of the bend region possibly because of the dominant effect of Dean-type vortices.

Regionally Averaged Nusselt Number Distribution

Figures 15 and 16 show the regionally averaged Nusselt number distribution normalized with Dittus-Boelter correlation for fully developed turbulent flow in circular duct (Nu/Nu_0) and smooth channel results (Nu/Nu_s).

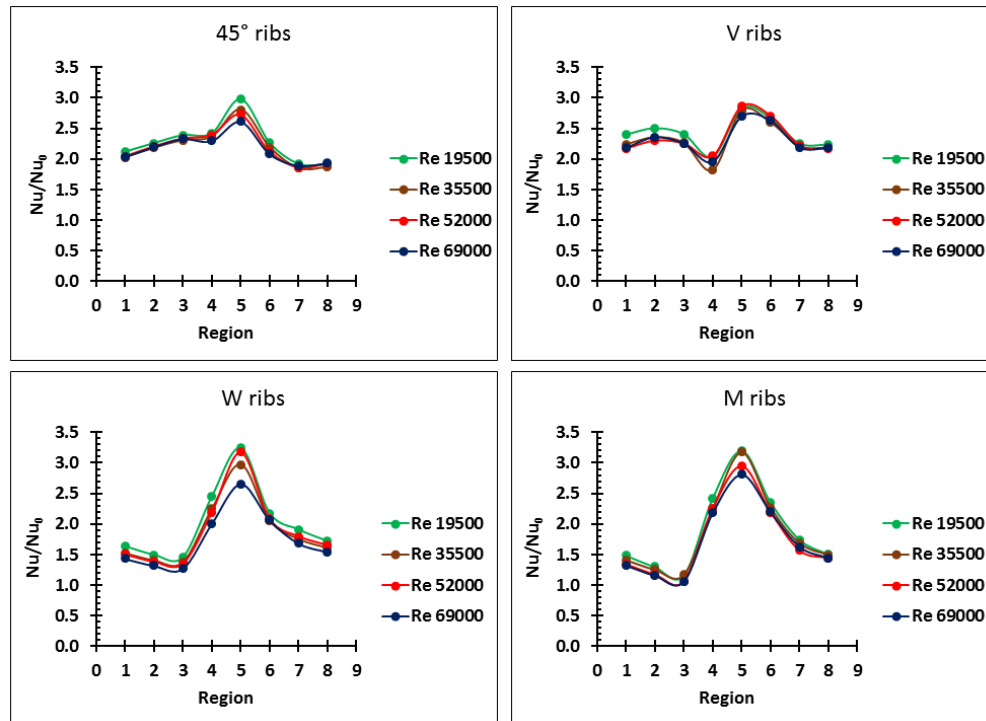


Figure 15: Regionally averaged Nusselt number normalized with correlations

With respect to the Nusselt number obtained from Dittus-Boelter correlation, the maximum heat transfer enhancement is observed in the bend region (region 5) for all the configurations. The enhancement in the bend region is higher for the W and M-shaped ribs when compared to V-shaped and 45° ribs. Looking at the Nusselt number normalized with smooth channel results (Nu/Nu_s), it can be seen that the enhancement in the bend region is significantly lower compared to the previous case for all the configurations. This illustrates that the major contribution to the enhancement in the bend region comes from the bend itself and not from the ribs.

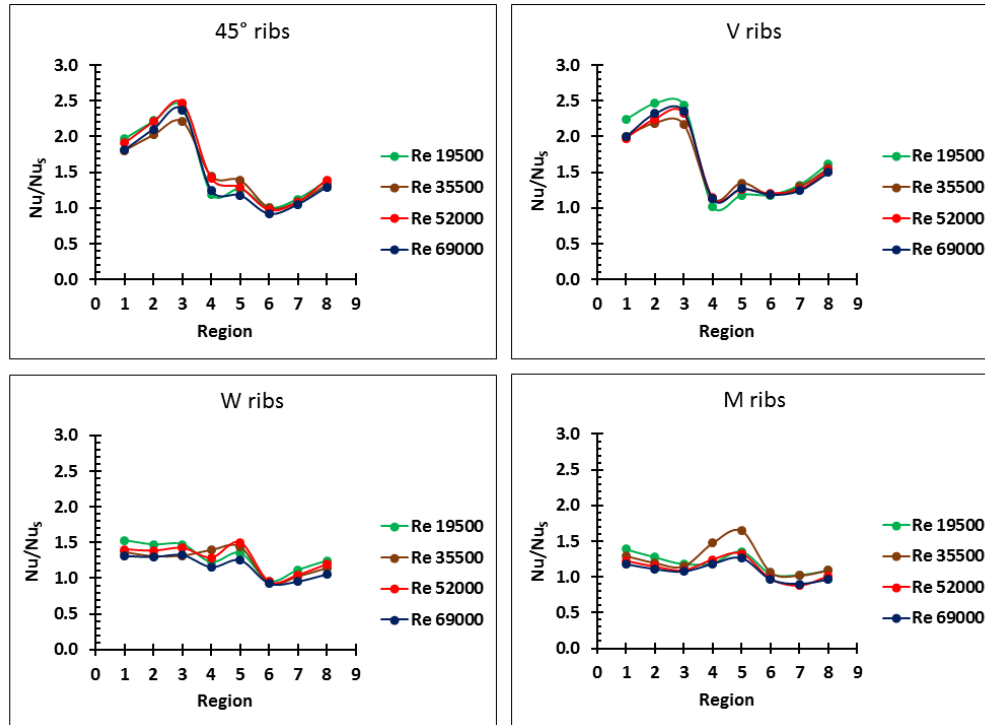


Figure 16: Regionally averaged Nusselt number normalized with smooth channel results

The average Nusselt number enhancement in the 1st pass, bend region and the 2nd pass has been shown in Fig. 17 for the different configurations at a Reynolds number of 35500. This gives an overall idea about how the ribs contribute the enhancement in different sections of the channel. Here again we see that the trends in the two plots (Nu/Nu_0 and Nu/Nu_s) are completely different. Furthermore, looking at the Nu/Nu_s plot, it can be observed that the enhancement in the 1st pass is higher than the 2nd pass, especially for the V-shaped and 45° ribs. In the 2nd pass, the bend induced turbulence contributes significantly to the heat transfer enhancement in the regions immediately

downstream of the bend. So the effectiveness of ribs is higher in the 1st pass where they entirely contribute to the heat transfer enhancement. For a ribbed two-pass channel, it is therefore important to normalize with smooth channel results rather than correlations so as to isolate the individual contributions of bend and ribs. With respect to the smooth channel, V-shaped ribs provide the maximum enhancement in the 1st and 2nd pass while M-shaped ribs provide the minimum enhancement. The enhancement provided by V-shaped ribs is 75% higher in the 1st pass and 27% higher in the 2nd pass compared to M-shaped ribs. However, in the bend region maximum and minimum enhancements were observed for the M-shaped and V-shaped ribs respectively (20% difference).

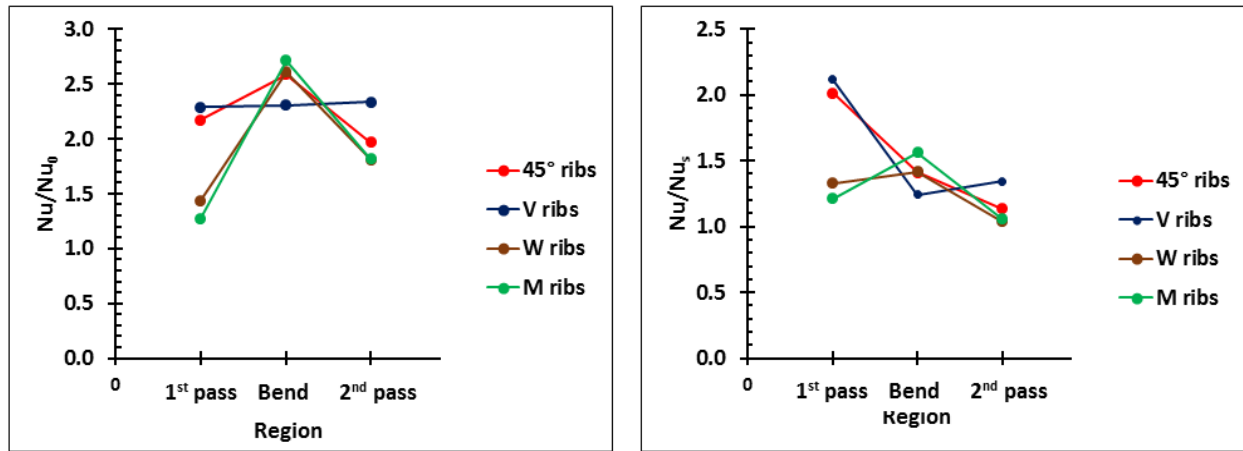


Figure 17: Nusselt number enhancement in different zones of the two-pass channel

Comparison of Overall Thermal-Hydraulic Performance

In this section, the overall thermal-hydraulic performance of the different rib configurations in the regions 1-8 have been compared. The globally averaged Nusselt number enhancement normalized with correlations and smooth channel results is shown in Fig. 18. The Nusselt number ratios were not observed to change significantly in the range of Reynolds number studied. V-shaped ribs provide the maximum heat transfer enhancement followed by 45° ribs. The enhancement in case of V-shaped ribs was observed to be 7% higher than 45° ribs, 28% higher than W-shaped ribs and 35% higher than M-shaped ribs. The enhancement in case of M and W-shaped ribs were observed to be significantly lower contrary to previous studies [9,10]. This could be because of the large rib spacing considered in this study.

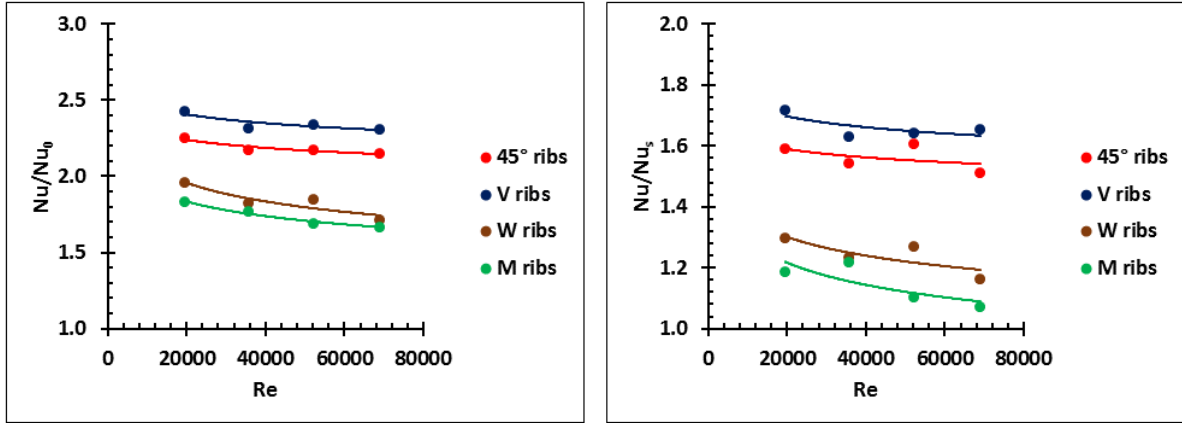


Figure 18: Comparison of overall Nusselt number enhancement

Figure 19 shows the comparison of the overall friction factor normalized with correlation and smooth channel results for the various configurations. Friction factor was calculated based on the pressure drop between the beginning of the 1st region and the end of the 8th region. The f/f_0 ratio was observed to increase with Reynolds number. For a two-pass channel, the bend contributes significantly to the overall pressure drop which is reflected in the f/f_0 and f/f_s plots. This again illustrates the significance of normalizing the results for a ribbed two-pass channel with smooth channel results. The pressure penalty for V-shaped ribs was 19 % higher than 45° ribs, 24% higher than W-shaped ribs and 28% higher than M-shaped ribs. The low pressure penalty for W-shaped and M-shaped ribs could be because of the recirculating flows caused by the vortex pair downstream of each rib as discussed before.

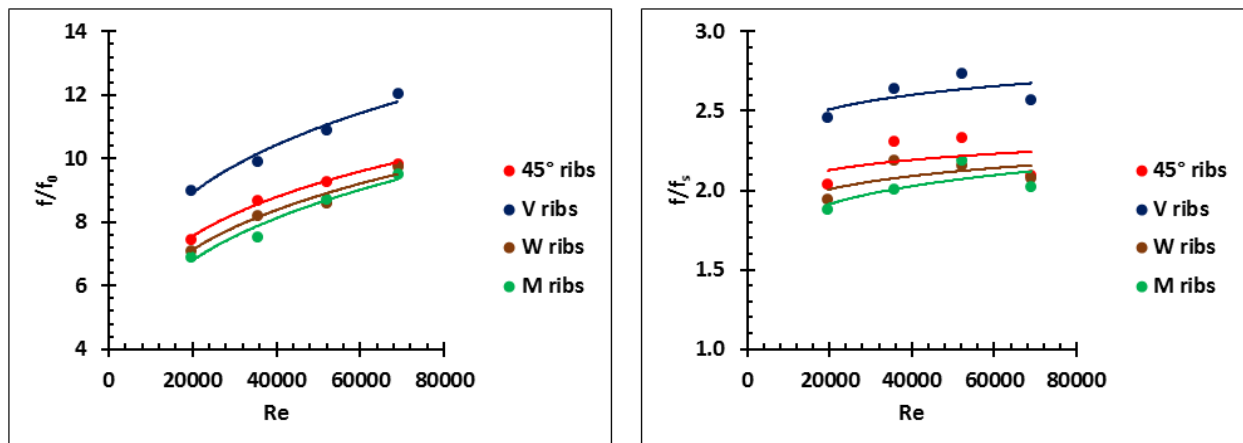


Figure 19: Comparison of normalized friction factor

The overall thermal hydraulic performance factors for the various configurations has been compared in Fig. 20. It can be seen that V-shaped and 45° ribs offer the best thermal-hydraulic performance amongst the configurations studied. However it must be noted that compared to 45° ribs, V-shaped ribs produce a much more uniform heat transfer distribution. Non-uniform heat transfer distribution will result in uneven temperature distribution which will in turn increase the thermal stresses in the material. Between V-shaped and M-shaped ribs a difference of 34% was observed in the thermal hydraulic performance factors. The overall performance of W-shaped and M-shaped ribs was comparable, with a difference of less than 10%.

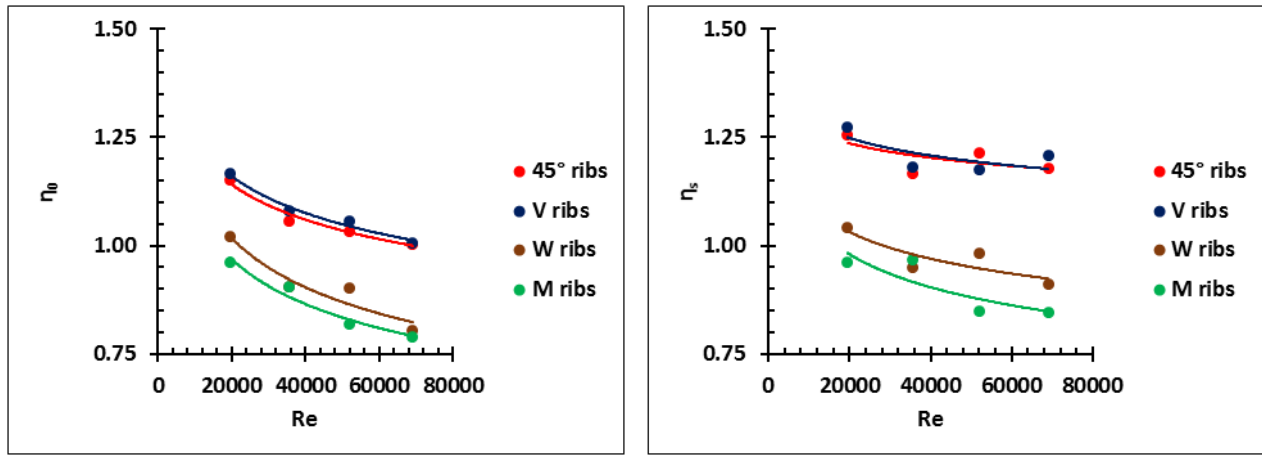


Figure 20: Comparison of thermal hydraulic performance

CONCLUSIONS

In this study, the flow field and heat transfer characteristics of a two-pass channel with different rib configurations was numerically investigated for a range of Reynolds numbers based on channel hydraulic number from 20000 to 70000. Four rib shapes - 45° angled, V-shaped, W-shaped and M-shaped were considered in this study. The performance of the ribs in a developing channel flow with high rib pitch-to-rib height ratio (p/e) was assessed. A reasonable agreement was observed between the results obtained from the simulations and experiments. However, the over predicted heat transfer enhancement in the first pass showed the significance of accurately modelling the entrance conditions which mimic the realistic setting in internal cooling channels. For the smooth channel, Dean-type vortices were found to dominate the flow and heat transfer characteristics in the bend and regions downstream of the bend. The interaction of the rib-induced secondary flows

and the curvature-induced secondary flows was studied. For the V-shaped, M-shaped and W-shaped ribs, the flow and heat transfer in the 2nd pass were found to be significantly influenced by the bend-induced vortices. The detailed investigation of flow and heat transfer characteristics presented in this study can be used by designers to optimize the shape, orientation and location of the rib turbulators in a two-pass cooling channel. V-shaped ribs provided the maximum enhancement in the 1st pass and 2nd pass while the M-shaped ribs provided the maximum enhancement in the bend region. V-shaped ribs were observed to provide the maximum overall heat transfer enhancement which was 7% higher than 45° ribs, 28% higher than W-shaped ribs and 35% higher than M-shaped ribs. However, the increase in heat transfer comes at the cost of increased pressure penalty. The friction factor for V-shaped ribs was 19 % higher than 45° ribs, 24% higher than W-shaped ribs and 28% higher than M-shaped ribs. Amongst the configurations studied, V-shaped and 45° ribs provide the best overall performance. The overall performance of W-shaped and M-shaped ribs were found to be lower than V-shaped and 45° ribs.

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CHAPTER 2

FILM COOLING PERFORMANCE OF TRIPOD HOLES ON THE ENDWALL UPSTREAM OF A FIRST STAGE NOZZLE GUIDE VANE

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ASME Paper No. IMECE2015-53133

ABSTRACT

In view of the growing energy demand, there is an increasing need to augment the thermal efficiency of gas turbine engines. The thermal efficiency and power output of gas turbine engines increase with increasing overall pressure ratio which in turn leads to an increase in turbine inlet temperature. The maximum permissible turbine inlet temperature is limited by the material strength of the components of the gas turbine engines. In this regard, it is important to ensure that the endwalls of the first stage nozzle guide vane, which is one of the critical regions, are adequately cooled. The cooling of the endwall is of particular interest because the leading edge region along the endwall of the stator vane experiences high heat transfer rates resulting from formation of horseshoe vortices. In this paper, the performance of upstream purge slot has been compared against discrete film cooling holes. Three different cooling configurations- slot, cylindrical holes and tripod holes have been investigated by comparing the adiabatic film cooling effectiveness. Furthermore, the effect of coolant to mainstream mass flow ratio on the effectiveness of the different cooling schemes has also been studied. The steady-state experiments were conducted in a low speed, linear cascade wind tunnel. Spatially resolved temperature data was captured using infrared thermography technique to compute adiabatic film cooling effectiveness. Amongst the configurations studied, slot ejection offered the best cooling performance at all mass flow ratios. The performance of tripod ejection was comparable to slot ejection at mass flow ratios between 0.5 and 1.5, with the difference in laterally averaged effectiveness being ~5%. However, at the highest mass flow ratio (MFR=2.5), the difference increased to ~20%. Low effectiveness values were observed downstream of cylindrical ejection which could be attributed to jet lift-off.

NOMENCLATURE

C_{ax}	vane axial chord length (m)
d	hole diameter (m)
l	hole length (m)
M	blowing ratio, $(\rho_c U_c) / (\rho_m U_m)$
MFR	mass flow ratio
IR	Infrared
p	hole pitch (m)
Re	Reynolds number, $\rho U C_{ax} / \mu$
T	temperature (K)
T_w	wall temperature (K)
T_c	coolant temperature (K)
T_m	mainstream temperature (K)
U	velocity (m/s)
ρ	density (kg/m^3)
η	adiabatic effectiveness
μ	dynamic viscosity (N-s/m^2)

SUBSCRIPTS

c	coolant
m	mainstream

INTRODUCTION

With increasing turbine inlet temperature and pressure, cooling of airfoil endwalls is of utmost importance. The presence of complex secondary flow makes endwall cooling complicated. Several methods have been employed to reduce the secondary flows like endwall profiling, upstream ejection, and endwall contouring. With upstream ejection, there is a possibility that the secondary flows might blow away the film jet thereby deterring proper film coverage. However, by optimizing the location, geometry and operating parameters, the coolant jets can be used to suppress the secondary flows. Thus, upstream ejection serves the dual purpose of suppressing the secondary flows and providing cooling benefit to the endwall surface.

In the past, several studies have been carried out to study passage secondary flows and their influence on endwall cooling. Goldstein et al. [1] used heat/ mass transfer analogy to confirm the existence of the various flow features associated with cascade endwalls like the corner vortex, passage vortex etc. Han et al. [2] presented a comprehensive review of commonly used platform cooling technologies. Blair [3] was the earliest to study the influence of secondary flow on endwall film cooling. He measured film cooling effectiveness and convective heat transfer coefficient distributions on the endwall of a large-scale turbine vane passage in the presence of an upstream cooling slot. He concluded that vane passage endwall heat transfer is strongly influenced by secondary flow vortex. Harasgama et al. [4] and Papa et al. [5] investigated the effect of blowing rate on endwall film cooling effectiveness. They observed that at low blowing rates the coolant was convected by the passage secondary flow resulting in ineffective cooling of the pressure side trailing edge region. They also pointed out that a much more uniform effectiveness distribution could be obtained by increasing the blowing rate.

Kost et al. [6,7] studied the interaction of the coolant and flow field and its influence on passage heat transfer for an endwall with upstream slot and film holes. They observed that the strength of endwall crossflow and passage vortex was significantly reduced by coolant ejection from both the slot and the holes. The horseshoe vortex was significantly strengthened by the coolant emerging from the slot. They attributed this to the slot being positioned in the region of the saddle point in front of the blade nose. Furthermore, they suggested that the intensification of the horseshoe vortex could be avoided by positioning the slot at a larger distance from the leading edge. Similar observations were reported by Thrift et al. [8], who studied the interaction of upstream slot leakage flow and formation of horseshoe vortex on an axisymmetric contoured endwall. Nicklas [9] carried out detailed temperature measurements to study the heat transfer and film cooling effectiveness in turbine endwalls and recommended improvements in design directed towards augmenting the effectiveness.

Oke et al. [10] performed film cooling experiments on the endwall of a first stage nozzle guide vane with two rows of upstream slots of varying area distribution. They found out that the secondary flow can be used to control the pitchwise coolant distribution. In addition, they also observed that the coolant was carried away by the secondary flow towards the suction side at low blowing rates, similar to Harasgama et al. Zhang et al. [11] measured surface film cooling effectiveness on a turbine vane endwall using pressure-sensitive paint (PSP) technique. A double

staggered row of holes and a single row of discrete slots were used to provide film cooling in front of the nozzle cascade leading edges. They also compared the performance of slot injection and hole injection and like previous studies observed coolant migration and decay at low blowing rates. They also pointed out that upstream slot injection results in a uniform effectiveness distribution at higher mass flow ratios whereas staggered hole injection provides higher effectiveness near the trailing edge region. The combined effect of back-facing step and jet velocity ratio was studied by Zhang et al. [12] for a double staggered row of holes in a warm cascade simulating realistic engine conditions. They observed that the back-facing step causes an unstable boundary layer which enhances the formation of the passage vortex and ultimately destroys the film coverage. It was demonstrated that by optimizing the jet velocity ratio the adverse effect of the back-facing step could be reduced.

Ranson et al. [13] carried out computational and experimental studies to analyse the cooling benefit from leakage flows resulting from slots at three different locations- upstream, mid-passage and downstream. They observed that the leakage flow from the mid-passage slot provided very little cooling benefit whereas the leakage flow from the upstream slot provided significant cooling to the platform surface. Thole et al. [14] studied the combined effect of upstream slot and film cooling holes in the endwall of a turbine vane. They observed that the upstream conditions significantly influenced the passage secondary flows. They also pointed out that the upstream slot cooling has a positive effect on film cooling effectiveness in the passage since a large portion of the mid-passage region along the endwall was sufficiently cooled by the slot flow alone and did not require additional cooling holes. Colban et al. [15] compared the aerodynamic and heat transfer performance of cylindrical and fan-shaped holes on a turbine endwall. They observed that the fan-shaped holes performed much better than the cylindrical holes from both aerodynamic and thermal perspective. Similar studies were carried out by Blot et al. [16] and Roy et al. [17] to analyse the thermal and aerodynamic performance of upstream purge slot. They observed that the backward-facing step present with the upstream slot has a strong influence on the strength of the passage vortex and the associated endwall heat transfer.

Heidmann and Ekkad [18] proposed the concept of tripod holes which has two holes which bifurcate from the center hole. The side holes produce vortices which counteract and suppress the effect of the counter rotating vortex pair (CRVP) generated by the main hole. Following this, there has been a significant interest in tripod holes and its application in gas turbine cooling. Leblanc et

al. [19] reported superior performance of tripod holes compared to cylindrical holes over a wide range of blowing ratios. Leblanc et al. [20] also investigated the effect of breakout angle on the performance of tripod holes. They observed that a breakout angle of 15° gave the best results and also noted that increasing the breakout angle to 30° significantly reduces the effectiveness.

While most of the research on upstream coolant ejection in the past have focused on slot and cylindrical ejection, there has not been any study on upstream tripod ejection. Past studies have shown that tripod holes outperform cylindrical holes when it comes to film cooling performance on flat plates. The present study investigates the film cooling performance of tripod coolant ejection upstream of a first stage nozzle guide vane at different mass flow ratios in comparison to slot and cylindrical ejection.

EXPERIMENTAL SET-UP

Figure 21 shows the schematic of the experimental set-up consisting of a blower, annular sections similar to a gas turbine combustion chamber, a nozzle and a 5-vane linear cascade. The experimental set-up is similar to that described in Ramesh et al. [21]. The blower was driven by a 15 HP motor and the mainstream flow rate could be varied by using the V-TAC automation controller. The linear vane cascade consisted of five GE E³ vanes, built from low conductivity ABS resin by Fused Deposition Method (FDM). The details of the cascade are given in Table 1.

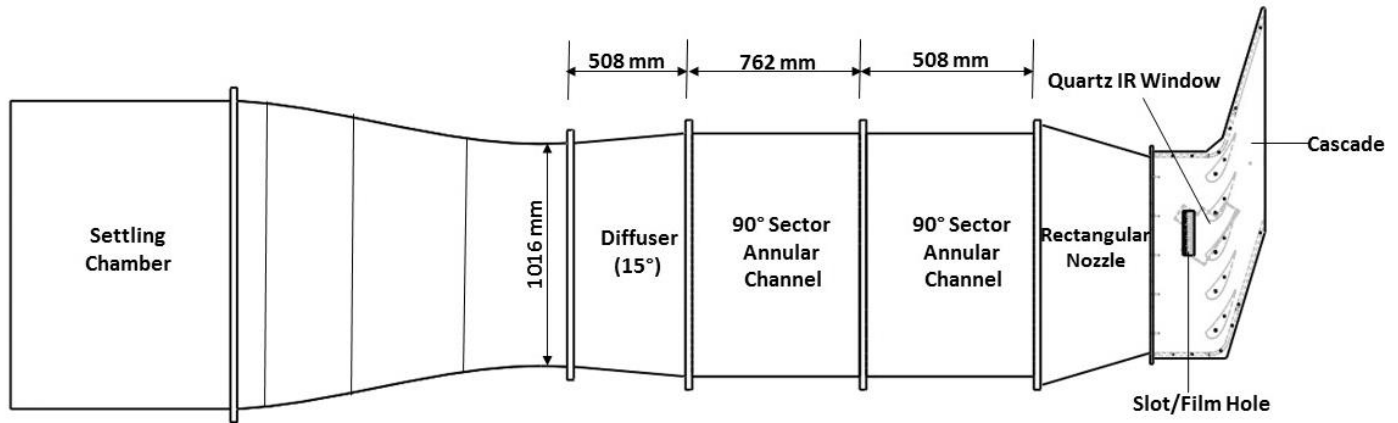


Figure 21: Low speed subsonic wind tunnel with a 5 vane linear cascade

The endwalls of the linear cascade were made of plastic acrylic glass which had a very low thermal conductivity of around $0.18 \text{ W m}^{-1} \text{ K}^{-1}$. The test surface was painted black to increase its emissivity. The IR window made of quartz was mounted on the slot cut on the opposite endwall to provide optical access for IR measurements. The infrared data was recorded using FLIR SC 6700 camera. The camera had a standard temperature measurement range of -20°C to $+350^{\circ}\text{C}$, a pixel resolution of 640 (H) x 512 (V), and an accuracy of $\pm 2^{\circ}\text{C}$ or $\pm 2\%$ of reading. Measurements were made on the portion of the endwall between the second and third vane as shown in Figure 22.

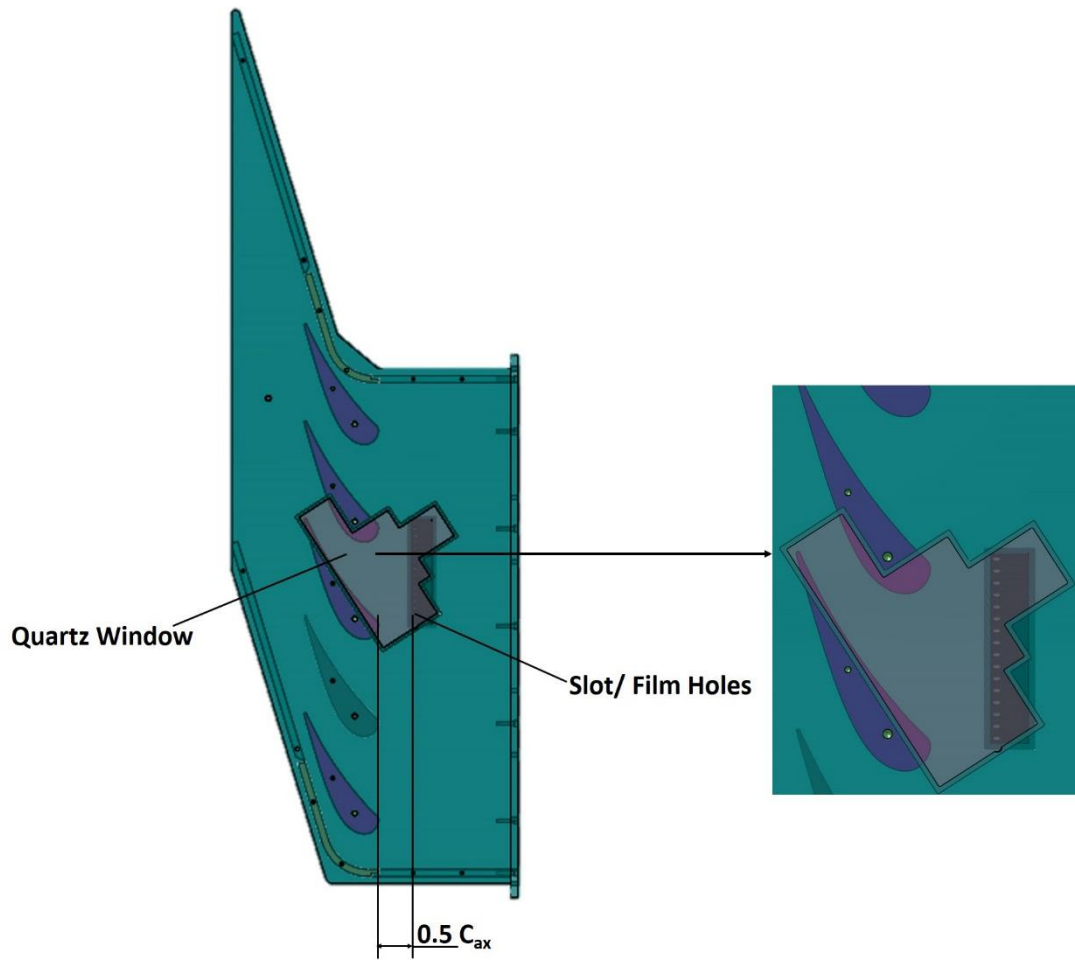


Figure 22: Schematic of linear cascade assembly along with the test surface seen through IR window

Figure 23 shows the secondary flow loop consisting of an air compressor, an inline heater and a plenum mounted on the backside of the endwall which supplies the coolant (or heated secondary

fluid). The temperature of the coolant was measured close to the hole exit using T-type thermocouple. An inline heater was used to heat the coolant to the desired temperature which could be adjusted using a variable auto transformer. The coolant mass flow rate was monitored using an orifice meter and controlled using butterfly valve. The mainstream temperature and velocity was measured 1-axial chord upstream of the vane leading edge. For all the cases, the mainstream velocity at the inlet was kept constant at 12 m/s. This corresponds to a Reynolds Number of 107000 based on the vane axial chord length.

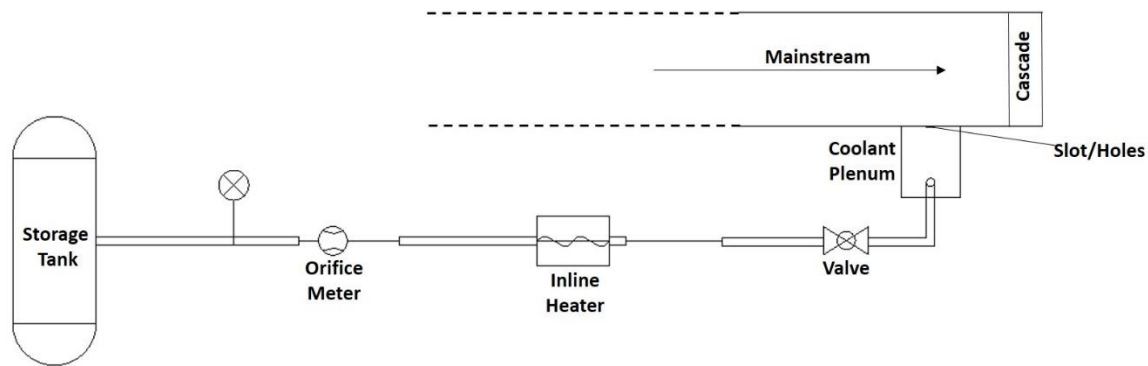


Figure 23: Schematic of the secondary flow loop of the test section

In this study, three different cooling configurations have been studied- slot, cylindrical holes and tripod holes. The geometric details of each of them have been elucidated in Figure 24. The exits of the slot and holes were located $0.5 C_{ax}$ upstream of the vane leading edge. The surface of the slot/ hole insert was sanded down so that it was flush with the endwall. The first configuration consisted of a slot, 190 mm long and designed in such a way that the trailing edge of the slot at the exit was inclined at an angle of 45° with respect to the streamwise direction. The second configuration consisted of an array of 8 tripod holes with a diameter of 4 mm. All the three holes of the tripod hole set were inclined 30° in the streamwise direction. The side holes had a branch out angle of 15° in the span-wise direction. The l/d ratio of the center and the side holes were 8 and 8.3 respectively. The p/d ratio between the main holes was $6d$, between the center and the side hole was $1.75d$ and between the side holes of the adjacent tripod hole unit was $2.5d$. The geometry of the tripod configuration is similar to that described in LeBlanc et al. [19]. The last configuration consisted of an array of 16 simple cylindrical holes of diameter 4 mm and inclined at 30° in the

streamwise direction. The hole length-to-diameter ratio (l/d) was 8 and the pitch-to-diameter ratio (p/d) was 3.

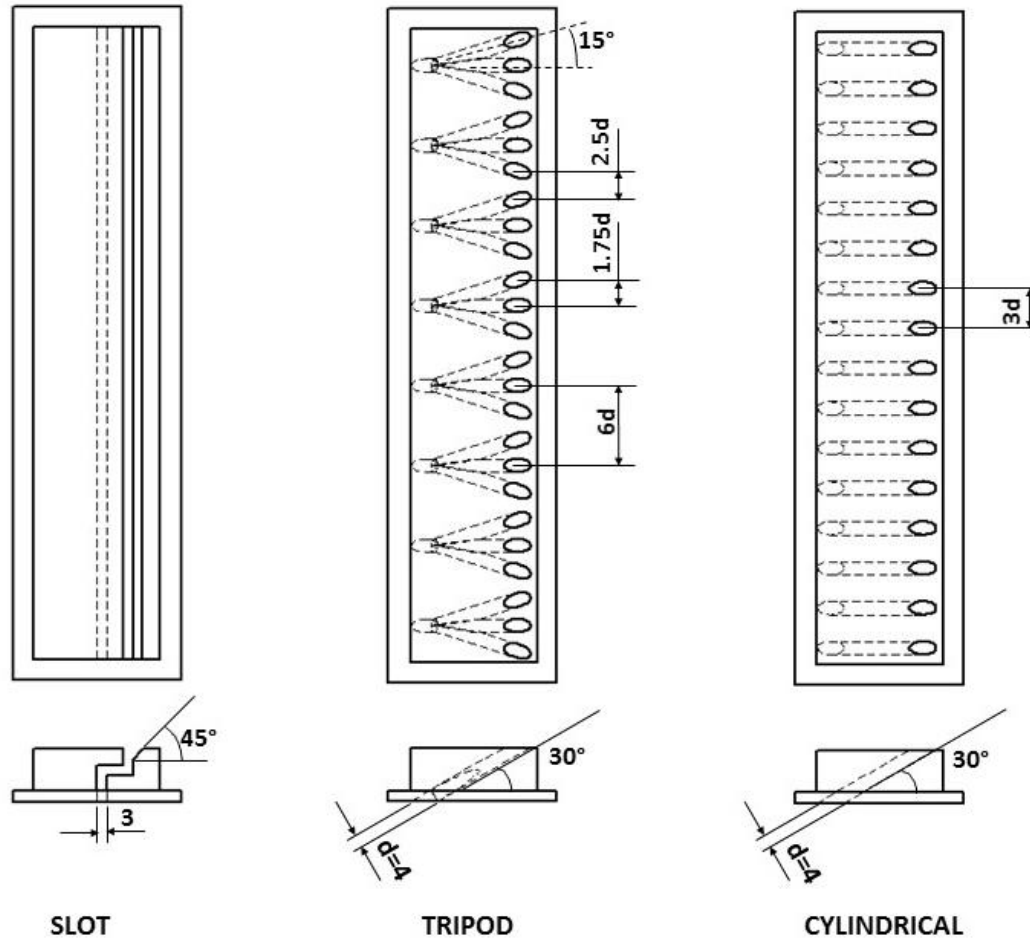


Figure 24: Film cooling configurations

MEASUREMENT METHODOLOGY

The endwall surface temperature distribution was obtained using infrared thermography technique. The infrared camera was in-situ calibrated using the T-type thermocouple mounted on the test surface. The test surface was heated by passing the hot air from the secondary flow loop. Upon reaching steady state, the thermocouple temperature and the temperature from the infrared camera were noted. This procedure was repeated at different surface temperatures and a linear fit was obtained between the thermocouple temperature and the temperature noted from the IR camera

(Figure 25). During the actual experiment, the corrected temperature from the IR camera was obtained by fitting the measured temperature in the curve.

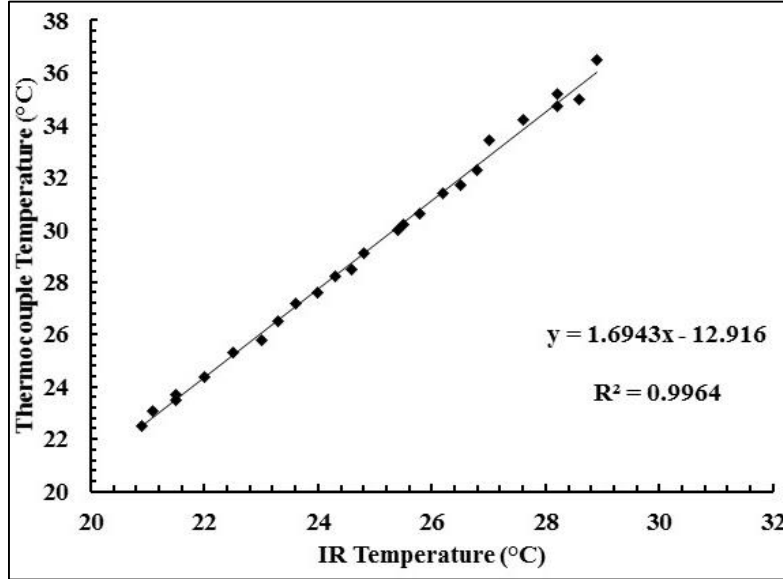


Figure 25: Calibration data for IR camera

During the experiment, the mainstream velocity was set at a constant value of 12 m/s while the coolant flow rate was adjusted to obtain the desired mass flow ratio. The variable auto transformer enabled setting of the inline heater to maintain an average temperature difference of 30 °C between the coolant and the mainstream for all the test cases. The mainstream, coolant and surface temperatures were continuously monitored during the experiment using a data acquisition system. When steady state conditions were reached, a series of IR images were taken. The endwall surface temperature distribution was obtained by averaging the data from the images. The adiabatic film cooling effectiveness was then calculated using the following equation:

$$\eta = \frac{T_m - T_w}{T_m - T_c} \quad (1)$$

Cross-conduction effects (from the plenum side) will be present only in the region immediately upstream and downstream of the hole/slot. The region of interest i.e. the portion of the endwall between the vanes is not likely to be affected by cross-conduction effects. Based on conjugate CFD simulations, ~2% error in effectiveness was observed close to the holes/slot due to conduction.

Since a comparison has been made between the different configurations and conduction error was expected to be the same for all the cases, no correction has been applied to the data.

Table 1: Geometric details of the cascade

Parameters	Values
Number of blades	5
Axial Chord	0.1349 m
Pitch	0.1778 m
Span	0.1524 m
Absolute flow angle	73.1°

EXPERIMENTAL UNCERTAINTY

The uncertainty in adiabatic film cooling effectiveness was estimated based on the method described by Kline and McClintock [22]. T-type thermocouples were used for measuring the mainstream and coolant temperature. The thermocouples were calibrated with water bath and thermometer to an accuracy of $\pm 0.3^\circ\text{C}$ and this was taken as the uncertainty associated with the coolant and the mainstream temperature measurement. As mentioned in the previous section, the infrared camera was calibrated using a T-type thermocouple and therefore, the uncertainty in surface temperature measurement was taken as $\pm 0.3^\circ\text{C}$. The uncertainty in adiabatic effectiveness was found to be $\pm 7.32\%$ at $\eta = 0.2$ and $\pm 30\%$ at $\eta = 0.05$.

RESULTS AND DISCUSSION

In this study, the performance of the different cooling configurations were studied at five different coolant to mainstream mass flow ratios- 0.5%, 1.0%, 1.5%, 2.0% and 2.5%. Mass flow ratio is defined as the ratio of the coolant mass flow rate to the per-passage mainstream mass flow rate. The mainstream mass flow rate was fixed and the coolant mass flow rate was adjusted so as to obtain the desired mass flow ratio. The adiabatic film cooling effectiveness was computed using the endwall surface temperature distribution obtained using the IR camera. Since the infrared window was composed of discrete quartz windows, temperature data could not be obtained at the

window junctions. The temperature data in these regions was obtained using a higher order interpolation.

Figures 26, 27 and 28 show the adiabatic film cooling effectiveness distribution for purge slot, tripod holes and cylindrical holes respectively. In general, it can be seen that the coolant is convected towards the suction surface by the secondary flow at all mass flow ratios and hardly any coolant reaches the pressure surface. This is because of the cross passage pressure gradient that exists between the suction and pressure sides of the vane. From Figure 26, it can also be seen that for all the mass flow ratios studied, the coolant exits uniformly through the slot. This could be because of the fact that the slot is situated considerably upstream of the vane leading edge ($0.5C_{ax}$), as a result of which the coolant ejection is not influenced by the conditions downstream. Close to the stagnation region, there exists a large uncooled region since the secondary flow at the vane leading edge sweeps away the coolant around the suction side. With increasing mass flow ratio, the coolant reaches farther downstream close to the suction surface. At higher mass flow ratios, traces of coolant can be seen close to the pressure surface and also around the leading edge. This could be because of the fact that at higher mass flow ratios, the coolant coming out of the slot has sufficient momentum to overcome the secondary flow. The effectiveness contours for slot ejection show good qualitative agreement with that of Thole et. al [14]. However, it is difficult to draw a quantitative comparison because of differences in slot location and flow parameters.

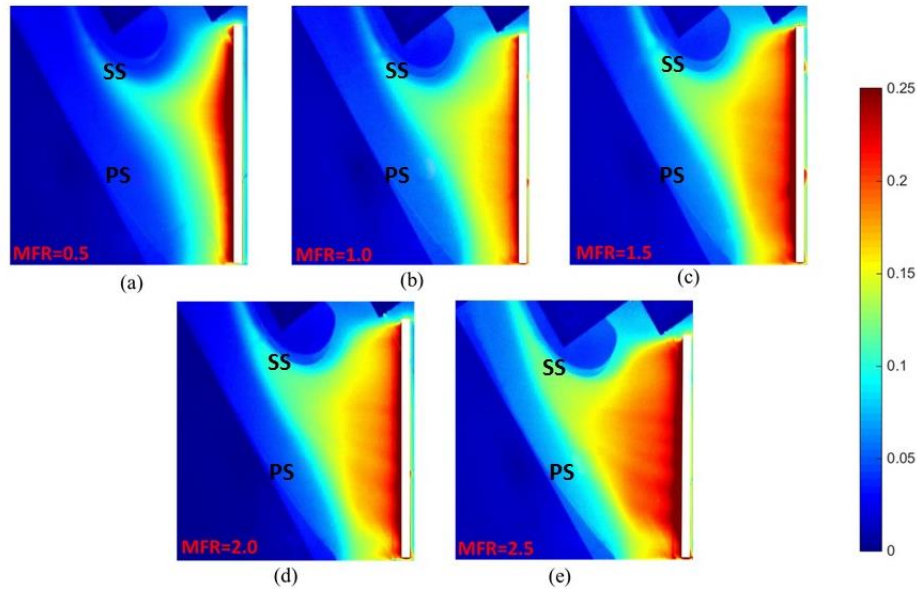


Figure 26: Adiabatic film cooling effectiveness contours for purge slot ejection

The effectiveness distribution for tripod ejection is shown in Figure 27. The high effectiveness region expands with increasing mass flow ratio. It can also be seen that the discrete coolant streaks start turning towards the suction surface upstream of the vane leading edge. As the mass flow ratio increases, the traces of coolant streaks can be seen farther downstream, sometimes even beyond the leading edge. The funneling effect resulting in the convection of the coolant towards the suction surface can be observed in the case of tripod ejection as well. At higher mass flow ratios, though the effectiveness values close to the pressure surface are lower ($\eta \approx 0.1$), we can see that more coolant reaches the pressure surface close to the leading edge in case of tripod ejection when compared to slot ejection.

Figure 28 shows the effectiveness distribution for cylindrical holes. The low effectiveness values across all mass flow ratios indicate sweeping away of the coolant by the mainstream. For cylindrical ejection, mass flow ratio of 1.0 corresponds to a blowing ratio of 1.4. It has been observed in past studies (Bogard et al. [23]) that separation effects start to dominate cylindrical ejection at blowing ratios greater than 1.0. At blowing ratios greater than 1.0, there is a tendency for the coolant to lift-off from the surface and diffuse into the mainstream. The lower effectiveness values for cylindrical ejection could be attributed to this. It can also be noted that, at high mass flow ratios, the tripod holes are more resistant to jet lift-off in comparison to cylindrical holes.

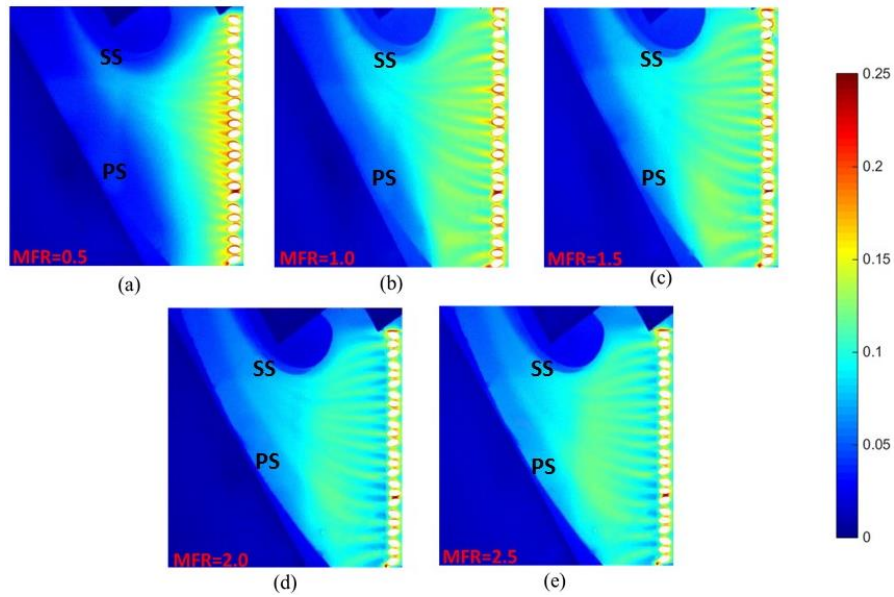


Figure 27: Adiabatic film cooling effectiveness contours for tripod ejection

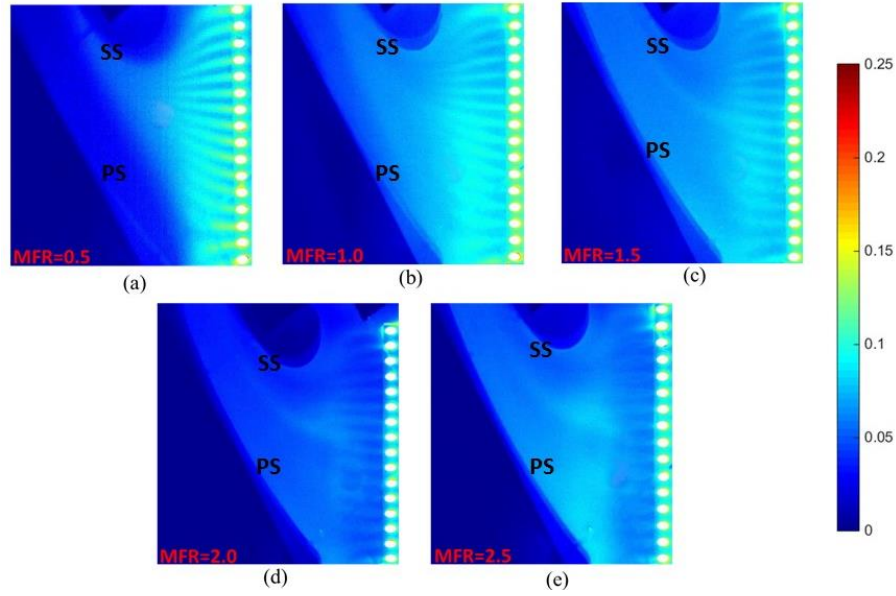


Figure 28: Adiabatic film cooling effectiveness contours for cylindrical ejection

Figures 29, 30 and 31 show the laterally averaged adiabatic effectiveness for slot, tripod and cylindrical ejection at different downstream locations starting from the vane leading edge ($x/C_{ax} = 0$). Similar trends are observed for slot and tripod ejection. In general, effectiveness is observed to increase with increasing mass flow ratio. As expected, the laterally averaged effectiveness decreases as we move downstream. Low laterally averaged effectiveness values for cylindrical ejection indicate that hardly any coolant reaches the vane passage.

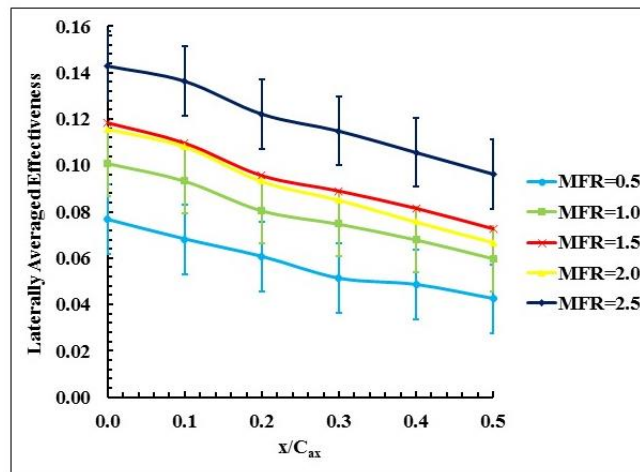


Figure 29: Laterally averaged effectiveness for purge slot ejection

At low mass flow ratios, the performance of tripod ejection is comparable to slot ejection. On comparing the laterally averaged effectiveness at different axial locations for slot and tripod ejection, an average difference of 5% was observed at mass flow ratios between 0.5 and 1.5. However, at the highest mass flow ratio, a difference of 20% was observed.

Error bars have been included in the laterally averaged effectiveness plots for MFR's- 0.5, 1.0 and 2.5. For cylindrical ejection, since all the curves are within the uncertainty limits, it is difficult to make any inference from the graph (Figure 31). Since the slot/ holes were located far upstream of the vane ($0.5 C_{ax}$), low effectiveness values were observed in the passage which resulted in a high % uncertainty in effectiveness.

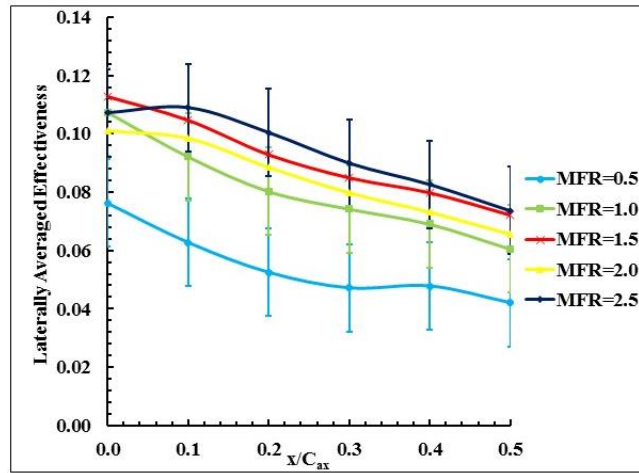


Figure 30: Laterally averaged effectiveness for tripod ejection

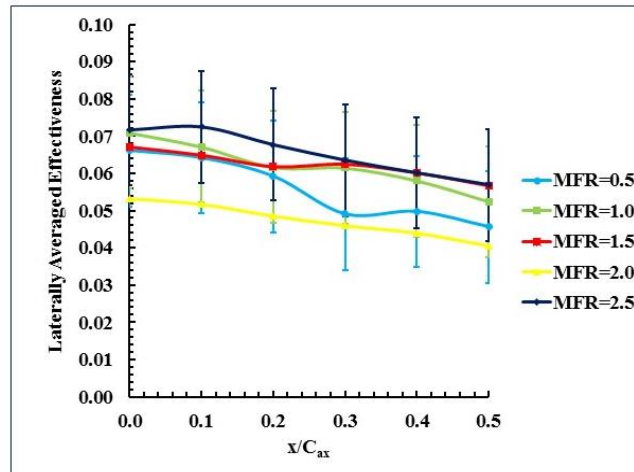


Figure 31: Laterally averaged effectiveness for cylindrical ejection

Figure 32 shows the area- averaged effectiveness for the various configurations. The area- averaged effectiveness values were computed over the endwall from $x/C_{ax} = 0$ to $x/C_{ax} = 0.5$ across one vane pitch. The area- averaged effectiveness plot gives an overall idea about the performance of the different configurations across all mass flow ratios. It reiterates the superior performance of slot configuration in comparison to the hole configuration. It can be seen that the performance of tripod ejection is comparable to slot ejection at mass flow ratios between 0.5 and 1.5. However, at higher mass flow ratios, slot ejection outperforms tripod ejection with differences as high as 15% in area- averaged effectiveness at the highest mass flow ratio.

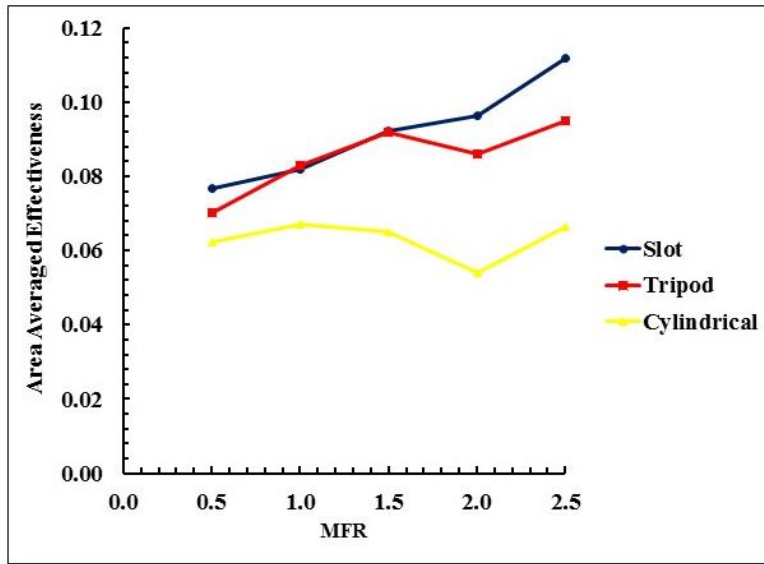


Figure 32: Comparison of area averaged effectiveness at different MFR's

It should be noted that the ejection angle is different for the slot (45°) and hole (30°) configurations. The optimum angle for slot ejection has been observed to be 30-40° by Jia et al. [24]. They have also pointed out that the difference between 30° and 40° ejection is not significant at larger blowing ratios. In addition, 40° ejection was observed to give better film cooling performance farther downstream. In the present study, slot configuration (45° inclination) was observed to give the best performance. The performance of slot ejection at 30° inclination is likely to be better than the predicted performance.

CONCLUSIONS

The performance of purge slot, cylindrical and tripod ejection upstream of the endwall of a first stage nozzle guide vane was investigated experimentally over a range of mass flow ratios (0.5%, 1.0%, 1.5%, 2.0% and 2.5%) using infrared thermography.

- In general, it was observed that at low mass flow ratios the coolant jets cannot overcome the secondary flows inside the passage.
- However, with increasing mass flow ratio, the coolant jets survive the secondary flow and provide much better film coverage even further downstream.
- The convection of coolant towards the suction surface was observed for all the three configurations at all mass flow ratios.
- Slot ejection was observed to give highest effectiveness amongst the configurations studied at all mass flow ratios.
- Very low effectiveness values were observed downstream of cylindrical ejection. This could be attributed to coolant lift-off at high blowing ratios corresponding to the mass flow ratios studied.
- Tripod ejection outperforms cylindrical ejection at all mass flow ratios.
- The performance of tripod ejection was on par with slot ejection at lower mass flow ratios, with an average difference of ~5% in laterally averaged effectiveness at mass flow ratios between 0.5 and 1.5. However, at the highest mass flow ratio, there was ~20% difference in effectiveness between slot and tripod ejection.

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