Numerical and Experimental Investigation of Heat Transfer Using Discrete Ribs

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ABSTRACT— A numerical investigation has been carried out to study the performance of V-discrete ribs. The effect of V-discrete rib arrangement on flat plate with forward and backward flow orientation for heat transfer and pressure drop characteristics is studied numerically and compared with performance of smooth plate. The range of parameters for this study is decided on the basis of available present work. The study encompasses Reynolds no. (Re) ranging from 7000 to 30,000, relative roughness pitch (P/e) 10 and angle of attack (α) 45^0. Themaximum enhancement occurs with backward flow orientation with increase in Nusselt no. (Nu) of about 31% compared with smooth plate and that of 16% with forward flow orientation over V-discrete rib plate.

Keywords: V-discrete rib, Forward flow, Backward flow, Relative roughness pitch(P/e).

I. INTRODUCTION

Heat transfer enhancement is directly proportional to turbulence of the fluid flowing over the heated surface. Rate of heat transfer can be increased by reducing thickness of thermal boundary layer over heat exchanger surface. Heat transfer enhancement can be achieved by different means as disruption of laminar sublayer in the turbulent boundary layer, introducing secondary flows, use of secondary heat transfer surface, promoting boundary layer separation, increasing fluid flow rate etc. The heat transfer enhancement techniques mainly classified into two categories as passive techniques and active techniques. The active heat transfer enhancement comes at cost of increased pumping power. For passive heat transfer enhancement there is no need of external power source. Heat exchangers are used for different applications as power generation, refrigeration, heating ventilating and air conditioning systems, process industries, manufacturing industries, aerospace industries, electronic devices cooling systems etc. For different applications there are space constraints for transfer of heat. So there is need to enhance the rate of heat transfer from available space. Consider the cooling passage in gas turbines, solar applications etc. where it is not effective to implement the active techniques for heat transfer enhancement. Also the different vortex generators cannot be placed due to space constraints and the complex designs.

The use of rib roughness is an effective way for the enhancement of heat transfer rate. The roughness rib element breaks up the boundary layer and induces turbulence which results in heat transfer enhancement. These rib elements are being smaller in height as compared to duct size causes turbulence in laminar sublayer adjacent to heated surface i.e. heat transfer is achieved by destroying laminar sublayer. But the increased rib roughness element causes high frictional loss resulting in increased pumping power. Artificial rib roughness is a passive technique of heat transfer enhancement by which thermo-hydraulic performance of heat exchanger can be improved. It is desirable that turbulence is created along the region very close to heat transfer surface. Thus reducing the pumping power required. And this can be done by keeping the height of rib element small as compared to duct dimensions. Formation of two secondary flow cells in case of V-ribs instead of one cell in case of angled rib results in better performance ofV-ribs.DiscreteV-shapedribarrangement can yield to better performance as compared to continuous rib arrangement. The most important effect of rib on flow pattern is generation of two flow separation region on each side of rib. These generated vortices are cause of turbulence as well as friction losses. Discrete ribs can create more secondary flow cell and produce more local turbulence than continuous ribs. Some of the important geometrical parameters for ribs are relative roughness pitch(P/e), relative roughness height(e/D), angle of attack(α), shape of element etc.

Nomenclature

English symbols
A heat transfer area (m^2)
AR Aspect ratio
D_h Hydraulic diameter(m)
e Rib height(m)
h heat transfer coefficient ($W/(m^2 K)$)
K Thermal conductivity of air ($W/(m^{-1}K^{-1})$)
Nu Nusselt no.
Re Reynolds no.

Greek symbols
α Angle of attack
μ Dynamic viscosity of air

Subscript
h Hydraulic

The experimental and numerical analysis of heat transfer with inclined discrete ribs in rectangular channel with effects of the Reynolds number, the height of the ribs and the number of double-inclined ribs along the mainstream shows that the heat transfer performance is enhanced effectively by the double-inclined ribs. The optimum ribs height is different under different Reynolds numbers. The heat transfer enhancement effect becomes stronger with the increase of the ribs number when the ribs heights are fixed (Wang, et al, 2014) [7].Study for heat transfer enhancement with V-ribs as turbulence promotor with investigation encompassing Reynolds number (Re) ranging from 4000 to 18,000, relative gap width (g/e) values of 1–5, relative roughness pitch (P/e) values of 6–12, angle of attack (a) range of 30–75 degree and relative roughness height (e/D) values of 0.043. For Nusselt number (Nu), the maximum enhancement of the order of 3.6 times that of smooth duct has been obtained (Maithani, et al, 2016) [1]. The V-shaped ribs have better thermal and hydraulic characteristics than transverse rib. For Reynolds number of 4000-40,000 it has found that the use of in-line ribs provides considerable heat transfer augmentations, the thermal efficiency of the roughened duct air heater being 6–26% higher than that of a smooth duct air heater the highest advantage is at the lowest flow rate(Karwa, et al, 2013) [9]. Investigation with turbulent flow and V-shaped rib configurations with three different inclinations (60, 45 and 30 degree) were studied and compared to the perpendicular (90 degree) rib case. The better results are obtained with inclination of 45(Fang, et al, 2015) [3]. A comparative experimental study on the heat transfer characteristics of steam and air flow in rectangular channels roughened with parallel ribs was conducted. Reynolds number for both coolants ranges from 3000 to 15,000, the rib spacing ratios were 8, 10 and 12, and rib angles were 90, 75, 60, and 45 degrees respectively. The heat transfer enhancement of both steam and air increased with decreasing the rib angle from 90 to 45 degree (Chao Ma, et al,2015) [4]. The performance of V and W ribs studied for different pitch to rib height ratios.The Reynolds number was varied from 5000 to 35,000 The rib heightto mean duct hydraulic diameter ratio (e/Dh,m) was kept constant at 0.08, Study for three pitch to height ratios (P/e) equal to 6, 10 and 17.5 were reported for straight and V ribs. The optimum P/e ratio based on constant pumping power thermal performance criterion was observed to be equal to 10 (Abraham, et al,2016) [2]. The effect of duct aspect ratio on heat transfer and friction characteristics was investigated experimentally and numerically. Four different duct aspect ratios (AR) were studied: AR = 1/4, 1/2, 1/1 and 2/1, the corresponding hydraulic diameter (D) was 32.0 mm, 53.33 mm, 40.0 mm and 53.33 mm, and the rib height-to-hydraulic diameter ratio (e/D) was 0.078, 0.047, 0.0475 and 0.047, respectively. The rib pitch-to-rib height ratio (p/e) was kept 10 for all the ducts. The investigated Re number ranges from 10,000 to 80,000 The averaged heat transfer coefficient ratio of steam was higher by 12–25% than air at the same test conditions(Shui, et al, 2013) [11].

Experimental and numerical study for discrete rib geometry with different flow orientations not observed from literature survey. Study on heat transfer investigation with discrete ribs by changing the rib geometry parameters such as relative roughness pitch, angle of attack, and rib height has done. But there is scope for work to investigate heat transfer with varying flow orientation and keeping the other geometrical parameters constant.

Objective of study

From the literature review presented above it can be observed that the investigation of heat transfer has been carried out with different angle of attack(α) and varying the relative roughness pitch(P/e). The presented work focuses on thermal-hydraulic performance of V-discrete ribs with forward and backward flow orientations to compare the results with smooth plate. A numerical investigation has been carried out for V-discrete ribs with 45^0 angle of attack and relative roughness pitch (P/e) of 10. The obtained results are to be validated experimentally.

Rib Geometry with flow orientation

Fig. 1::V-Discrete rib flow orientation

The flow orientations (i.e. forward flow and backward flow) are shown in Fig.1.
There are different parameters used to specify the rib geometry as follows,

1. Relative roughness pitch (P/e):It is the ratio of distance between two consecutive ribs(P) and height of the rib(e). For this study relative roughness pitch is 10.
2. Angle of attack(α) is 45^0,
3. Shape of rib element: Rib element of square cross section 2mm×2mm×15mm is selected for this study.
4. Relative gap between two ribs is 10mm

Boundary conditions

Boundary conditions for this study are selected on the basis of literature available. The parameters are selected in order to cover wide range of heat exchanger application as possible. The Reynolds no. (Re) for this study ranges from 7000 to 30000. Constant heat flux condition is used with plate heater provided three step inputs as 15W,25W and 35W. Forward and backward flow orientations are selected for V-discrete rib test plate.

Meshing and grid independence study

The computational model of test plate after meshing is shown in fig. 2. The tetra/mixed type of mesh is used for this computational model. The mesh density is selected at rib element on test plate to get more accurate results. For tetra/mixed mesh there are three methods available as robust, quick (Delaunay) and smooth. Delaunay method is robust and fast for meshing of complex meshing. The no. of terms to be modelled for turbulent flow is large, different turbulence models are available for turbulence modelling as k-ε, k-ω etc. Here

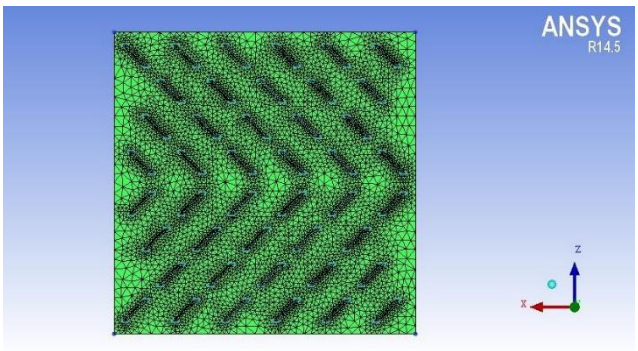

Fig. 2:Computational model of test plate after meshing

the k-ε model is used for simulating flow characteristicsin turbulence modelling as it is widely used and reliable.Grid independence study is done over the different no.of cells. The characteristics of grids 79,128, 1,07,263 and 1,22,600 cells are used for simulation. The variation in Nu and ΔP is less than 0.25% when increasing no. of cells from 1,07,263 to 1,22,600. Hence it is quite disadvantageous to increase no. of cells beyond this value. So grid system of 1,07,263 cells is adopted for this computational model.

II. EXPERIMENTAL SETUP AND PROCEDURE

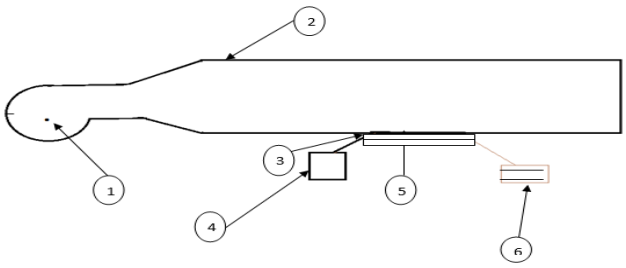

Fig. 3: Block diagram for experimental setup

The block diagram for experimental setup is as shown in fig. 3.The experimental setup consists of 150mm× 150mm× 10mm aluminium test plate with rib elements placed in V shape at an angle of attack 45^0 (3). The plate heater is placed below test plate(5). This test plate assembly with thermocouple as temperature measuring device is placed in acrylic duct of size 2500mm×150mm×100mm(2). Air blower (1) connected at inlet section. Temperature indicator (4) and wattmeter (6) connected to respective connectors. The proper insulation is provided with insulating tape. The instruments used for experimentation areheater plate of nichrom wire with dimensions same as that of test plate is used, variable speed type blower with rpm range of 1600, Pt-100 Simplex type thermocouples with three core cable, vane probe anemometer with velocity measuring range 0to30m/s and wattmeter with capacity to measure 0to750watts is used for the experimentation.

While conducting test constant power input is provided with respective velocity. Temperature of air at inlet and exit of test section and at test plate are recorded at steady state condition. The data is collected for different specified velocities and power input combinations for smooth plate, forward flow orientation and for backward flow orientation.

Data reduction

Values for air and plate temperatures are obtained from temperature plots at different locations. Heat transfer coefficient (h) and Nusselt no. (Nu) is calculated from data. Pressure drop (ΔP) is obtained from pressure plot. These values are used for investigation of various influencing parameters.

Following equations are used for evaluation of parameters,

Reynolds no. (Re):

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

Heat transfer coefficient (h):

$$Q = hA\Delta T \quad (2)$$

Nusselt no.(Nu):

$$Nu = \frac{hD_h}{K} \quad (3)$$

III. RESULTS AND DISCUSSION

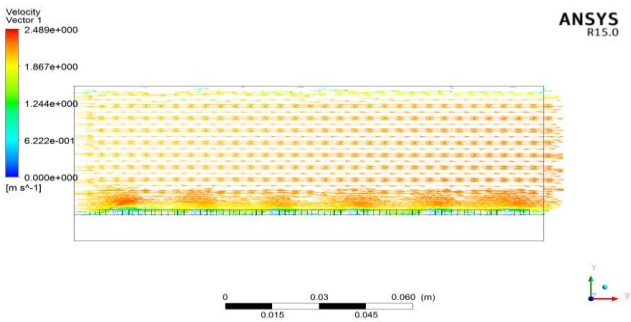

Fig. 4: Velocity vector plot over V-discrete ribs

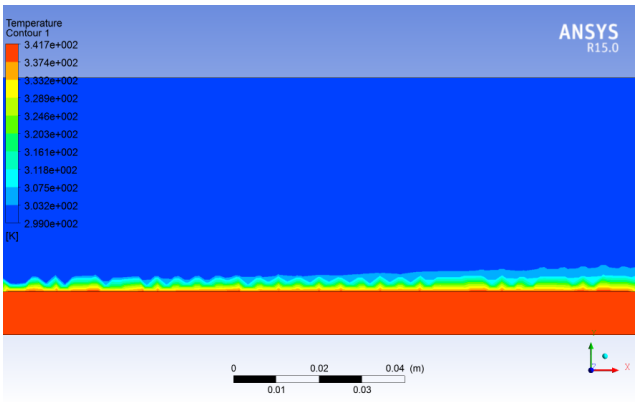

Fig. 5:Temperature contour for flow over smooth plate

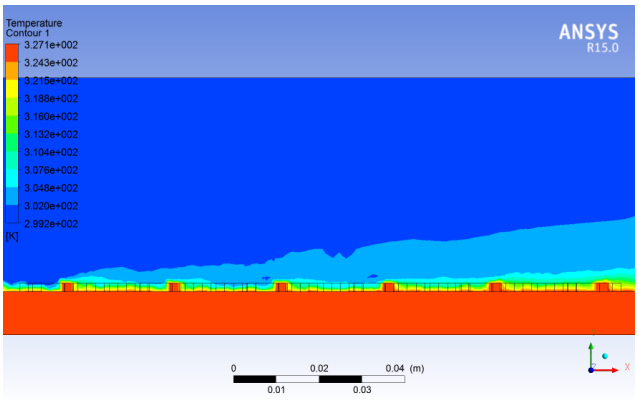

Fig. 6:Temperature contour for forward flow over V-discrete ribs

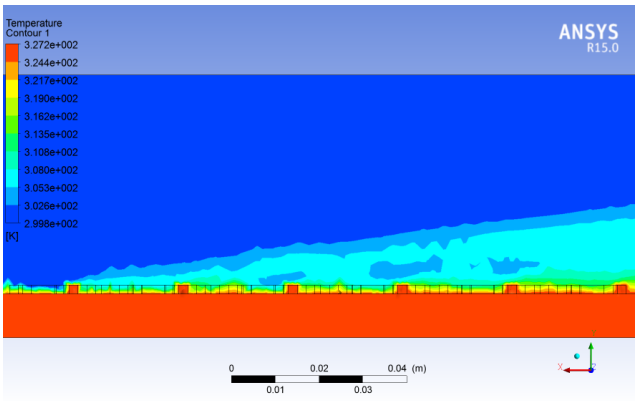

Fig. 7:Temperature contour for backward flow over V-discrete ribs

Fig.4 shows vector plot for the flow over V-discrete ribs. It shows turbulence created by discrete ribs at the region near the heat transfer surface. The effect of turbulence is enhanced heat transfer rate with comparatively less pressure drop than other vortex generation. For smooth plate the temperature profile shows developing boundary layer. With use of discrete ribs, the boundary layer is not developed due flow separation and turbulence caused due to discrete ribs.Due to turbulence the laminar sub-boundary layer breaks causing increased heat transfer rate. Fig. 5, Fig. 6, Fig. 7 shows the temperature contours for smooth plate, forward flow with discrete ribs and backward flow with discrete ribs respectively. From contours it can be observed that the sub-boundary layer is developed for smooth plate causing less heat transfer, contours for forward and backward flow orientation shows the more heat transfer due to increase flow separation and turbulence caused by discrete ribs. Temperature contour with backward flow over discrete ribs show more heat transfer than the other with intense temperature profile.

The Nusselt no.for smooth plate and modified plate under similarpower input condition are shown in fig.8, fig.9, and fig10. The results are plotted for Nu vs. Re for all the three flow orientations i.e. flow over smooth plate, forward flow and backward flow over V-discrete rib plate at constant power input condition.

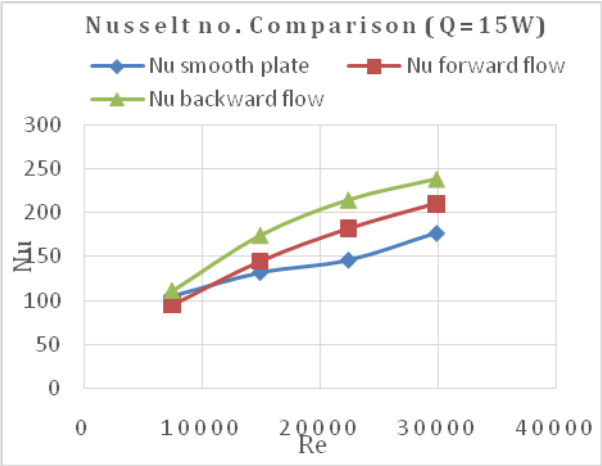

Fig. 8: Comparison of Nu for smooth plate, forward and backward flow for power input of 15W

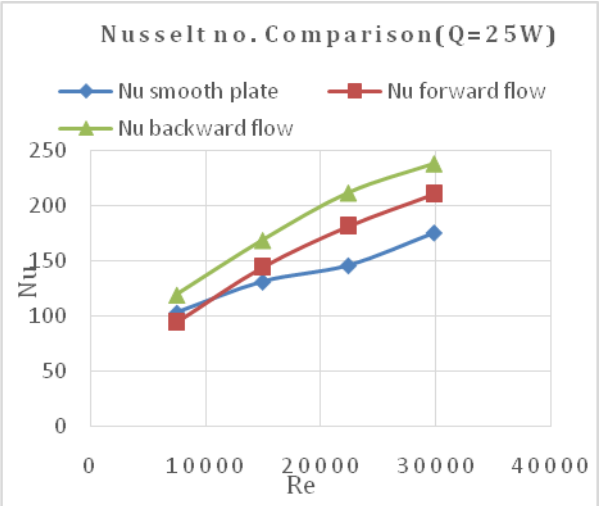

Fig. 9: Comparison of Nu for smooth plate, forward and backward flow for power input of 25W

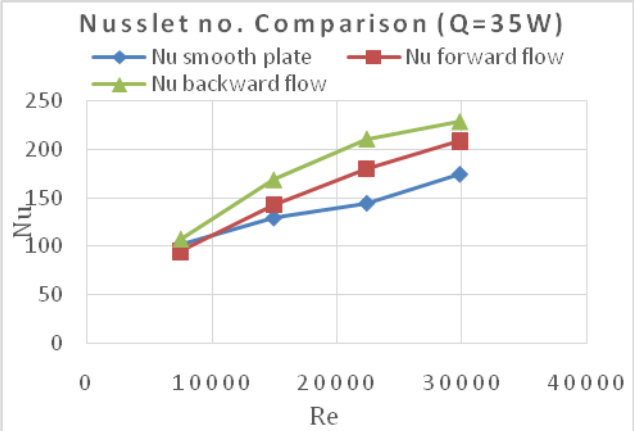

Fig. 10:Comparison of Nu for smooth plate, forward and backward flow for power input of 35W

Fig. 8 shows comparison of Nusselt no. for flow over smooth plate, forward flow and backward flows over V-discrete rib plate with power input of 15W. It is observed that the Nusselt no. (Nu) for backward flow over ribbed plate is considerably more than the forward flow and flow over smooth plate. This is the case for constant heat flux and over entire range of Reynolds no. (Re) i.e. 7000 to 30000 for all three flow orientations. It is also observed that Nu increases with increase in Re. Fig.9 and Fig.10 shows Nu Vs Re for the heater input 25W and 35W respectively. It shows similar pattern as that of result obtained for 15W heater input. With change in heater input the Nu increases with Re in same pattern.

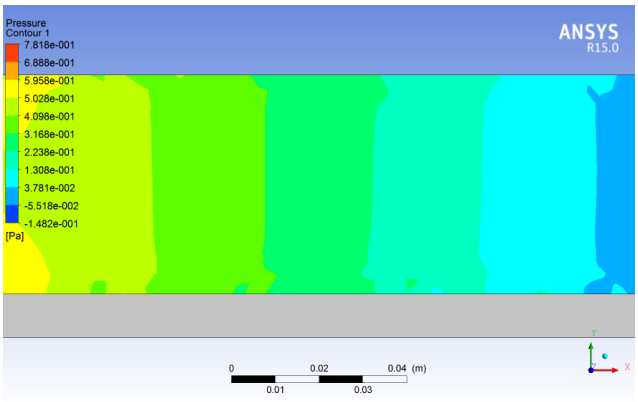

Fig. 11: Pressure contour for flow over smooth plate

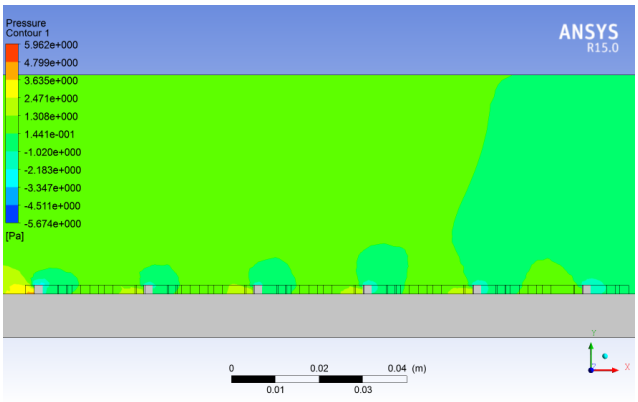

Fig. 12: Pressure contour for forward flow over discrete ribs

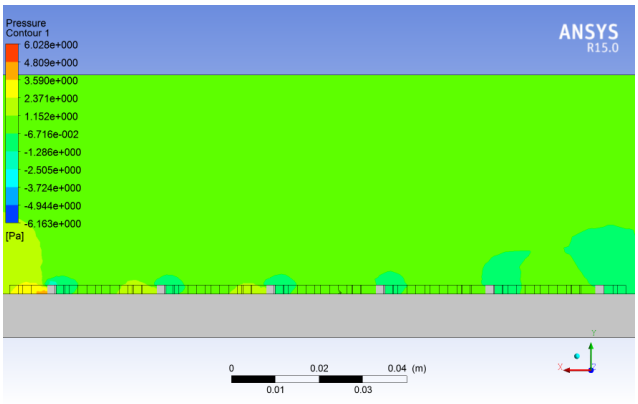

Fig. 13:Pressure contour for forward flow over discrete ribs

Fig.11, Fig.12 and Fig. 13 shows the pressure contours across flow over smooth plate, forward and backward flow orientation over discrete ribs. From this it is observed that pressure drop for V-discrete rib plate is increased than that of the pressure drop over smooth plate. It is also observed that the pressure drop for forward flow and backward flow over V-discrete rib plate as nearly same. So the discrete ribs are an effective means for increasing heat transfer rate with small increase in pressure drop.

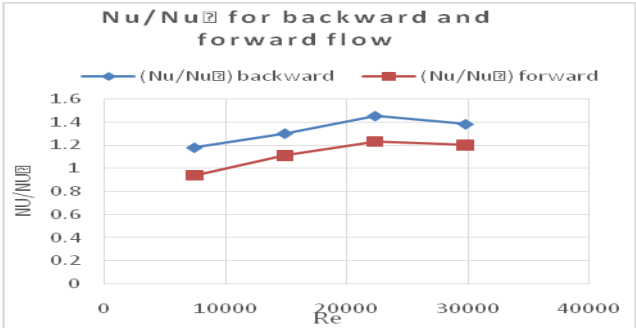

Fig. 14:Nu/Nu□Vs Re for discrete ribs

Fig. 14 shows Nu/Nu□variation with Re for effectiveness of discrete ribs over smooth plate. It is observed that Nusselt no. effectiveness increases gradually with Re up to certain value and the its value decreases slightly. It is also seen that Nu effectiveness is more for backward flow than with forward flow.

After comparing Nusselt no. (Nu) and pressure variation for smooth plate and forward and backward flow over V-discrete plate vs Reynolds no. (Re), we can say that heat transfer rate increases with increase in Reynolds no.

IV. CONCLUSIONS

The heat transfer characteristics for V-Discrete ribs with forward flow and backward flow has been investigated numerically. Based on the results following are the conclusions,

1. Use of V-Discrete ribs is an effective technique for heat transfer enhancement by promoting boundary layer separation.
2. The Nusselt no. (Nu) is strongly dependent on flow orientation (i.e. Forward flow and Backward flow) over the V-Discrete rib surface.
3. For backward flow the Nusselt no. is increased by about 31% as compared to smooth plate and about 16% increase in Nu than forward flow orientation with small change in pressure drop
4. The increase in Nusselt no. (Nu) for backward flow than with forward flow is considerable with nearly same pressure drop.
5. The Nusselt no. effectiveness is maximum with Reynolds no. 22300 for backward flow.

REFERENCES

1. Rajesh Maithani, J.S. Saini, (2016) Heat transfer and friction factor correlations for a solar air heater duct roughened artificially with V-ribs with symmetrical gaps, Experimental Thermal and Fluid Science, 70, 220–227

2. Satyanand Abraham, Rajendra P. Vedula, (2016), Heat transfer and pressure drop measurements in a square cross-section converging channel with V and W rib turbulators, Experimental Thermal and Fluid Science, 70, 208–219

3. Xingjun Fang, Zixuan Yang, Bing-Chen Wang, Mark F. Tachie, Donald J.Bergstrom, (2015) Highly-disturbed turbulent flow in a square channel with V-shaped ribs on one wall, International Journal of Heat and Fluid Flow, 56, 182–197

4. Chao Ma, Xiaoling Chen, Jianfei Wang, ShushengZang, YongbinJi, (2015), An experimental investigation of heat transfer characteristics for steam cooling and air cooling in a rectangular channel roughened with parallel ribs, Experimental Thermal and Fluid Science 64, 142–151

5. Heeyoon Chung a, Jun Su Park a, Sehjin Park, (2015) Augmented heat transfer with intersecting rib in rectangular channels having different aspect ratios, International Journal of Heat and Mass Transfer ,88, 357–367

6. Chao Ma, Xiaoling Chen, (2015), An experimental investigation of heat transfer characteristics for steam cooling and air cooling in a rectangular channel roughened with parallel ribs, Experimental Thermal and Fluid Science, 64, 142–151

7. Yingshuang Wang, Bing Zhou, Zhichun Liu, ZhengkaiTu, Wei Liu, (2014) Numerical study and performance analyses of the mini-channel with discrete double-inclined ribs, International Journal of Heat and MassTransfer, 78 498–505

8. Abhilash Kumar and S. Prabhakar, (2014), Numerical investigation of heat transfer characteristics of square duct with internal ribs, IJIRSET, volume 3,1179-1185

9. RajendraKarwa and V. Srivastava, (2013), Thermal Performance of Solar Air Heater Having Absorber Plate with V-Down Discrete Rib Roughness for Space-Heating Applications,Journal of Renewable Energy Volume, Article ID 151578, 113-126.

10. S. S. Pawar, D. Hingolia ,(2013), Experimental study of Nusselt no. in solar air heater with diamond shape rib roughness on plate, AJER, volume 2, 60-68

11. LinqiShui, Jianmin Gao, XiaojunShi,Jiazeng Liu, (2013) Effect of duct aspect ratio on heat transfer and friction in steam-cooled ducts with 60 degree angled rib turbulators, Experimental Thermal and Fluid Science,49, 123–134

