Rib Spacing effect on Heat Transfer and pressure loss in a Rectangular channel with semicircular ribs

DOI : 10.36909/jer.10795

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ABSTRACT

Ribs of various shapes are used for heat transfer enhancement but its performance is significantly depends on geometrical features and flow conditions. This study experimentally find out the influence of rib spacing, semicircular shape ribs with three rib spacing ratio (P/e) = 8, 10 and 12 are studied and located on lower wall of the rectangular channel. Reynolds numbers varied from 10000 to 29,000 and the blockage ratio of the channel (e/D_h) was 0.151.Result show that semicircular rib performed better than plain plate but found more friction. semicircular rib with rib spacing of 50 mm (P/e =10) shows highest thermal performance, enhanced avg. 39 % heat transfer than rib spacing of 40 and 60 mm (P/e=8 & 12). Friction losses observed highest in rib spacing ratio of 8,found average 10 % more friction compared to rib spacing ratio of 10 & 12. Semicircular rib with spacing ratio 8 shows least thermal performances compared to other configurations.

Key words: semicircular rib, heat transfer enhancement, rib spacing, rectangular channel.

Nomenclature		
Symbol	**Description**	**Units**
A	rectangular duct cross section area	m^2
h	heat transfer coefficient	$W/m^2 K$
D_h	Rect. duct hydraulic diameter	m
W	Width of rect. duct	m
H	height of rect. duct	m
L	Test section length	m
p	Rib pitch	m
e	Rib height	m
$\dot{m}$	Air mass flow rate	kg/s
AR	Channel aspect ratio (W/H)	
p/e	Ratio of rib pitch to rib height	
Q_{net}	Net Heat gain	W
T_i	Inlet temperature of air	0^C or K
T_o	outlet temperature of air	0^C or K
Dimensionless number		
Re	Reynolds number	
Nu	Nusselt number	
Nu_o	smooth duct Nusselt number	
f	Friction factor	
fo	Friction factor for smooth duct	

INTRODUCTION

The main goal of every industry is to achieve the maximum efficiency by using latest techniques, same case in gas turbine engines to get more efficiency and output it is required to operate at maximum inlet temperature. This high inlet temperature was not sustained by blade material; to save life of turbine blade cooling is necessary. Internal cooling method consists of pin fin cooling, rib turbulators, dimple, protrusions, etc. generally ribs ae preferred for cooling, the performance of rib turbulators depends on various factors like rib angle, blockage ratio, pitch ratio, rib shape, an inclination of rib, stationary or rotating condition. In past decades many researchers reported on various kinds of rib turbulators for heat transfer enhancement. Luca Baggetta studied 45 deg angled rib configuration with intersecting ribs, concluded that due to presence of intersecting ribs friction increased compared with heat transfer coefficient, as interesting ribs increased friction found more (Luca Baggetta et al.2018).Kaewchoothong reported inclined ribs using the thermal liquid crystal sheet, concluded that Nusselt number for 60^0, 45^0 angled ribs and 60^0 V-shaped ribs have increased by 20 to 30% than 90^0 angled ribs (Kaewchoothong et al.2017).Truncation ribs was studied by Liu at Reynolds number 80,000, concluded that truncated ribs reduced the friction without decreasing the heat transfer rate (Jian Liu et al. 2018).

Some investigators focused on numerical techniques. Moon numerically studied sixteen ribs shapes at Reynolds numbers of 5000 -50,000. The Reynolds stress model was used and concluded that boot-shaped rib shows highest performance (Mi-Ae Moon et al.2014). Zheng using simulation find out the influence of rib arrangements on the thermal performance with V-type and parallel ribs, result showed that thermal performance of V-type ribs was highest compared to P-type ribs (Zheng et al. 2016). Shukla presented 90^0continuous, V -broken attached thick and thin ribs. show that that 90^0 attached ribs shows highest enhancement at rib spacing ratio (P/e) 10, also performance of thin ribs was improved by thick ribs (Anuj Shukla and Anupam Dewan 2016). Xie numerically reported various offset mid-truncated ribs at Reynolds number ranging from 10,000 to 50,000. Truncation ratio kept 12% in the middle and concluded that due to truncated ribs the heat transfer improved with reduced pressure loss penalty (Gongnan Xie et al.2014). Wandong Bai numerically studied the entrance effect of rib induced vortices in pin fin array. The SST k-ω turbulence model used and show that the 90 deg. rib had the best overall performance (Wandong Bai et. al. 2019).

Few researchers focused on rib spacing, Ansari reported the rib spacing effect at entry length of solar air heater. Found that the non-uniform ribs with an increased pitch shows better thermal performances (Mohammad Ansari et al.2020). Tanda experimentally studied the effect of rib spacing using 45 angled rib, reported different rib spacing ratio (P/e = 6.66, 10, 13.33, 20) at Reynolds number from 9000 to 35000.Concluded that rib pitch ratio at 13.33 performed best for one ribbed wall channel and 6.66 -10 for double ribbed wall (Govanni Tanda 2011, 2009). Yang experimentally studied three rib spacing ratio 5, 10 &15 at Reynolds number from 1400 to 9000. Found rib spacing ratio 10 performed well compared to 5 & 15 spacing ratio, also symmetric ribs performed better compared to staggered ribs (Yang et al. 2017). Guoqiang studied effect of rib spacing at the pitch ratio 3.8 to 14.4 on the pressure side and 10 on the suction side in a rotating channel. Results show that in positive direction at rib spacing ratio 3.8 achieved more enhancements and in negative direction at rib spacing ratio 10 indicates the good performance (Guoqiang Xu et al.2015). Chaube experimentally studied gap effect in inclined ribs, show that highest performance achieved when rib has relative gap width of 1.0 at relative gap position of 1/3 (Chaube et al.2014).

Recently some authors focused on newly developed ribs. structured miniature and conventional ribs presented by Yanlin and show that performance of miniature structured ribs was higher than the conventional ribs; also W shape rib performed better compared to other tested ribs (Yanlin Li et al. 2019).Sharma focused on pentagonal ribs and measured surface temperature distribution using liquid crystal thermography. Concluded that the pentagonal ribs performed well than square ribs, also reported solid and converging slit ribs using liquid crystal thermography. Show that converging-slit ribs considerably enhance the heat transfer rate in the downstream vicinity. (Naveen Sharma et al.2017 & 2018). Alfarawi focused on hybrid ribs and show that hybrid ribs performed well compared with the rectangular and semi-circular ribs.(S. Alfarawi et al. 2017). From the literature review it was observed that less research work found on rib spacing using semicircular shape ribs. Previous researchers are mostly focused on parallel, angled and V shape ribs with spacing only few authors focused on circular type ribs but they can't focused more on rib spacing so there is still scope to work on it. The objective of the present study is to find new cooling method for gas turbine blade, which is essentially the optimum rib spacing of semicircular rib. In the proposed research experimentally find out the heat transfer characteristics and friction factor of semicircular shape ribs with three rib spacing ratio (P/e) 8, 10, and 12.

EXPERIMENTATION

Experimental Facility

To collect experimental data experimental setup is designed, fabricated and validated.Fig.1 shows schematic of experimental setup, it consists of centrifugal blower, developing section, an orifice plate with U tube manometer, inlet section, rectangular channel with test section, exit section, k-type thermocouples and data acquisition system.

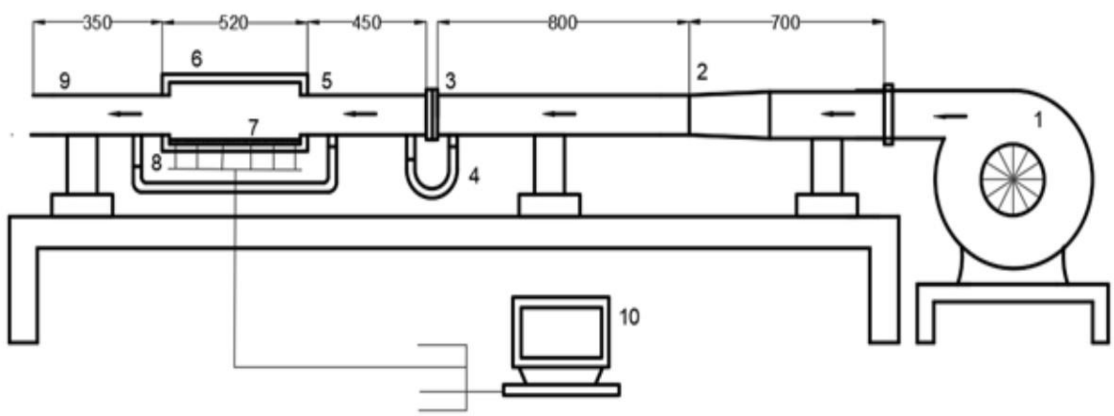

Figure 1. Schematic of experimental test facility.

1-blower,	5- Inlet section	8- Thermo-couples
2- developing section	6- Rectangular channel	9 - Exit section
3 & 4 - orifice plate & U tube manometer	7- Test section	10- data acquisition system

In experimental set up blower was followed by developing section, total length of this section is 1500 mm, length kept more to ensure fully developed and regularize the flow. Developing section followed by inlet section of length more than10 times the hydraulic dia. of channel (D_h) where again flow becomes uniform and regularizes. Mass flow rate of air was controlled by dimmer-stat and measured using orifice plate assembly. Rectangular channel was placed in between the inlet and the exit section and made from Bakelite material to reduce heat conduction loss. Aspect ratio of rectangular channel was 4:1, hydraulic diameter (D_h) 30 mm and length of channel was 540 mm. In rectangular channel test section with base plate and heater was inserted, exit section of length more than 6 times the hydraulic dia. of channel (D_h) was attached at the end to reduce the end effects of flow. Fig.2 shows actual photograph of orifice plate and blower.

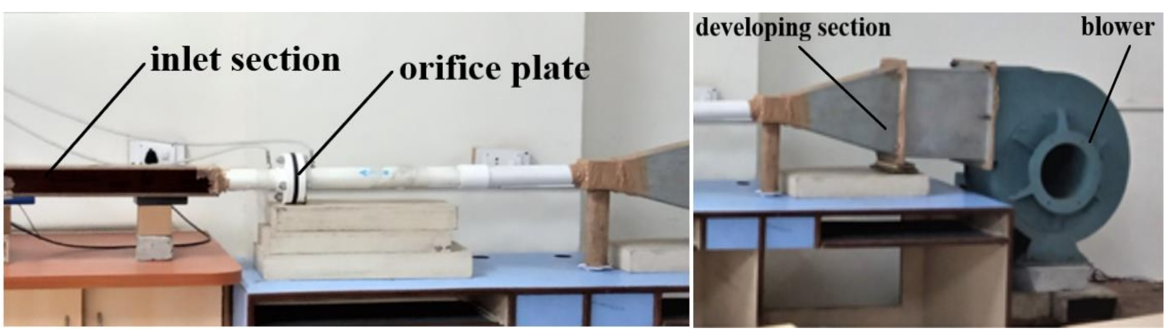

Figure 2. Photographs of experimental set up.

Pressure drop along the rectangular channel was recorded using a micro manometer connected along inlet and outlet of the test plate. Air inlet and outlet temperature are measured using two thermocouples located at inlet and exit side of duct and connected to data logger.

Twelve thermocouples are provided at surface of test plate to measure the temperature of test section, copper constant K-type thermocouples are used for temperature measurement.The uniform heat flux was supplied by the flat plate heater of size 300 × 50 mm and inserted between the test plate and base plate. Heat input was varied with the help of dimmer-stat and All instruments used for experimentation are calibrated from NABL accredited Laboratory. At steady-state conditions all data recorded initially; it takes 2 to 3 hours.

Test Section and Rib configurations

The material used for test section was aluminum, thickness and length of the test plate was 10 mm & 540 mm. The semicircular rib was pasted on the test plate using thermal glue. The test plate along with a heater and the supporting plate was tightly fastened with nut and bolts and placed in a rectangular channel.in order to reduce the heat losses, the bottom surface of heater and side wall are insulated with the insulating material.

Rib Configurations - The ribs are pasted to a test section using bonding material. Semi-circular shape ribs are manufactured from 75 mm diameter aluminum pipe. Aluminum pipe is first made hollow using a boring tool and then prepared outside diameter of the pipe in the correct size on lathe machine after that on special purpose machine required size round ribs are cut. The height and width of the rib are 5 mm. Height of rib to channel hydraulic dia. ratio (e/D_h) =0.156, and ratio of rib pitch to rib height (P/e) was 8, 10 & 12. Fig.3 shows actual photograph of rectangular channel with test plate.

Figure 3. a) Rectangular duct with test section assembly (10 ribs & 12 ribs).

Figure 3. b) Semicircular rib (P/e = 10 & 8).

Rib arrangement - three ribbed channels are prepared for testing with different rib spacing. In first set semicircular ribs having rib spacing ratio (P/e) =8 studied, here distance between two ribs are 40 mm. In the second set same ribs are studied but pitch of rib to height of rib ratio (P/e) = 10, means distance between two ribs are 50 mm and in third set rib spacing ratio (P/e) kept 12, means distance between two ribs are 60 mm. Ribs are tested at Reynolds number from 10000 to 29000 and placed on the lower surface of channel and aspect ratio of channel was 4:1.

DATA REDUCTION AND ERROR ANALYSIS

The recorded data of temperatures and pressures at different places in the channel are used to find out various parameters such as, friction factor and its ratio; Nusselt number and its ratio, thermal performance etc. heat transfer coefficient (ha) was calculated by equation (1)

- Q_{net} = ha A (Ts-Tb)
- $$\mathrm{h} = \frac{\mathrm{Q_{net}}}{\mathrm{A(T_s - T_b)}} \text{ -------------} \qquad (1)$$

$$Ts = \frac{T_1 + T_2 + ---- + T_{12}}{12} \ , \quad T_b = \frac{T_{air\,in} + T_{air\,out}}{2}$$

Where Ts is average surface temperature of test plate and T_b is bulk temperature of air. Net heat input was calculated considering in account heat losses from the test plate (conduction, convection), the maximum heat loss from the test section was calculated to be less than 12 %. Nusselt number was calculated using equation (2), where h- average heat transfer coefficient, D_h - hydraulic diameter of channel, A - cross section area of channel & p – perimeter of channel

- $\mathrm{Nu} = \frac{\mathrm{h\,D_h}}{\mathrm{k}}$------------------------------ (2)

$$D_h = \frac{4\ \mathrm{A}}{\mathrm{p}}$$

The Dittus–Boelter co-relation was applied to validate the Nusselt number of plain plate. The Nusselt number ratio is given as

- $Nu/Nu_o = (hD_h / k) / (0.023\ Re^{\mathbf{0.8}}\ Pr^{\mathbf{0.4}})$ ------------ ------- (3)

Frictional Losses- friction factor was calculated by measuring pressure drop along the test channel and calculate using equation (4)

- $\Delta P = fL\rho V^2 avg./2\ D$ --------------------------- (4)
- $f = (P_i - Pe) / [\ (4(L/ D_h)\ (\ 1/2\ \boldsymbol{\rho} V^2)]$

The frictional factor ratio was calculated by considering friction factor of smooth plate. Using Blasius co-relation validates friction factor values of smooth plate.

- $f / f_0 = f / (0.079\ Re^{\mathbf{-1/4}})$ -------------------------------- (5)

Thermal Performance- Thermal hydraulic performance was calculated by taking account the Nusselt number ratio (Nu/Nuo) and the frictional factor ratio (f/fo). Equation (6) was used to evaluate the performance (η) of rib configuration.

- $\eta = (Nu/Nu_0) / (\ f / f_0)^{1/3}$ ------------------------- (6)

Error analysis - Kline and McClintock (1953) method was used for uncertainty analysis; the calculated uncertainty in the temperature measurement was found $\pm\ 1^0$ for all cases. In the Nusselt number the average uncertainty was almost 8 -11 % of the experimental values. Initially more heat loss was observed at low Reynold's number due to that more uncertainty observed but as flow velocity increased uncertainty reduced. In the friction factor the overall uncertainty was almost 7-10 % of the experimental values.

Heat transfer in smooth duct

The experimental results of smooth duct are validated using correlation results. Fig. (4) presents experimental result of Nusselt number with result achieved from Dittus-Boelter correlation (7) for plain duct. A better relation was seen when linking the experimental results of Nusselt number with the correlations results, maximum 13% deviation observed in Nusselt number between the experimental and correlation values. Experimental Friction factor value correlates with values obtained from Modified Blasius equation (8) for plain duct. Meantime, the friction factor of smooth duct correlates closely with Blasius's correlation as presented in Fig. (5), observed highest 10 % difference in the theoretical and experimental value of friction factor.

- $Nu_o = 0.023\, Re^{0.8}\, Pr^{0.4}$ ---------------- (7)
- $f_o = 0.079\, Re^{-1/4}$ ---------------------- (8)

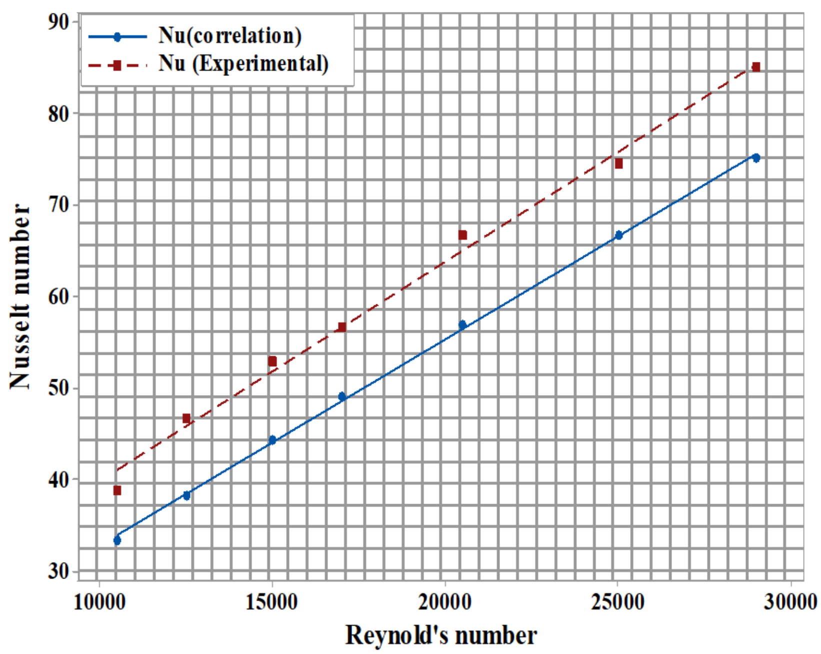

Figure 4. Comparison of smooth duct results of Nusselt number with correlation.

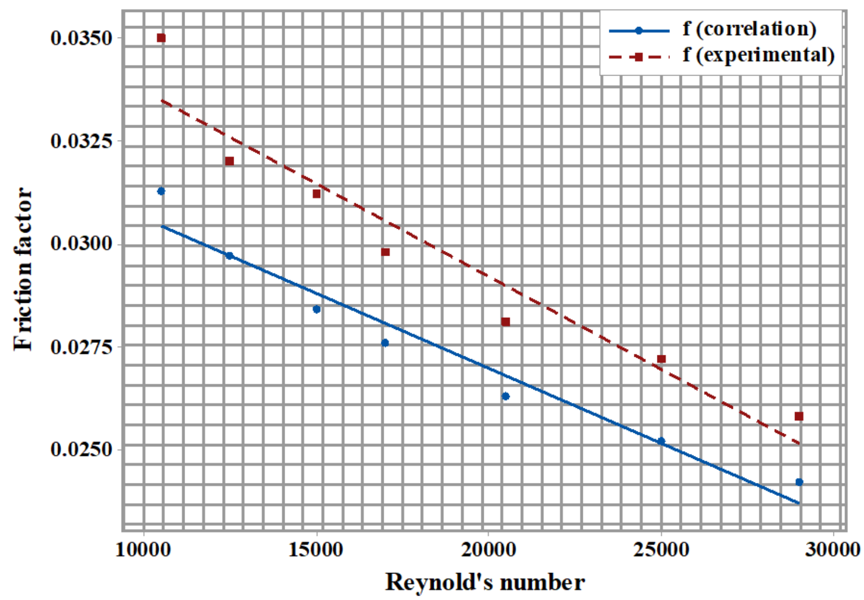

Figure 5. Comparison of smooth duct results with correlation.

RESULTS AND DISCUSSION

Here Nussult number and friction factor of all tested configurations are presented and discussed at a range of Reynolds number 10000 to 29000. The results of different configurations have been studied and compared with each other to find optimum rib spacing for semicircular rib.

Heat transfer in ribbed duct

From experimental data Nusselt number of semicircular rib was calculated at various rib spacing and shown in Fig. (6).Nusselt number for all channels shows increasing trend means as Reynolds number increased Nusselt number also increased. From graph also seen that semicircular rib performs well compared to plain plate, enhanced up to 130 % heat transfer compared to plain plate. Application of ribs on the surface increased the heat transfer area and also increased separation of rib turbulators on flow boundary layer due to this reasons rib increased heat transfer but due to obstruction pressure drop created. Semicircular shape ribs have more frontal area due to curved shape and also create reduced heat transfer area behind the ribs because of this heat transfer increased. In case of semicircular rib high turbulent kinetic energy appears at the middle region between two adjacent ribs and in front of the ribs also observed that the fluid recirculation region with low turbulent kinetic energy and velocity appears behind the ribs.

From fig. it also indicates that for semicircular shape rib the Nusselt number was highest at rib pitch to height ratio (P/e) =10, after P/e = 8 & 12 follows at all Reynold's number means the semicircular rib with spacing ratio 10 performs better compared to other tested configurations. In the case of 40 mm rib pitch (P/e=8), spacing between two ribs observed less due to that the flow reattachment to the rib wall will be slowly reduced as the falling of the rib pitch and the phenomena of flow reattachment was not developed, thus; it reduced the heat transfer rate at the rib pitch ratio (P/e = 8). In the case of 60 mm rib pitch (P/e=12), spacing between two ribs observed more due to that boundary layer grows later reattachment points and also the reattachment point at the wall was reached and a boundary layer starts to develop early, the subsequent rib is come across thus; it reduced the heat transfer rate at the rib pitch ratio (P/e = 12), but more over to 40 mm rib spacing. However, as the increase of the rib spacing, the effects of separation vortex and reattachment on convective heat transfer are weaker and weaker.

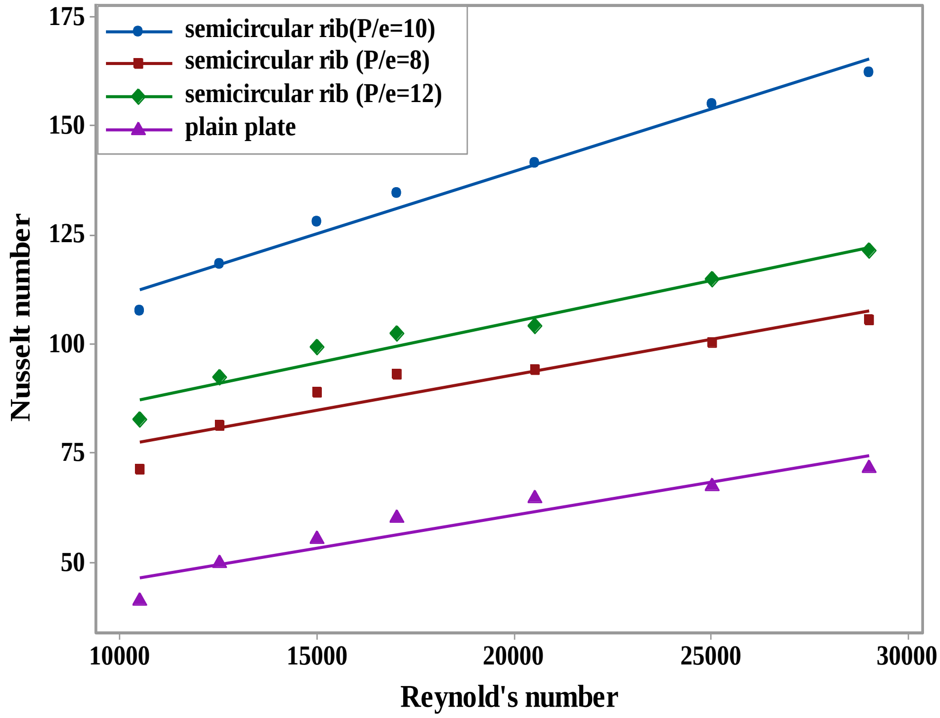

Figure 6. Reynold's number vs Nusselt number for semicircular rib.

In case of rib pitch 50 mm (P/e=10) heat transfer observed highest because the flow does reattach near to the next rib and the flow separation vortex and reattachment to rib wall play important roles in enhancement. Also seen distinct separation region and reattachment region between ribs, the flow strongly reattaches to the ribbed wall and detaches again quickly.in case of 50 mm spacing observed the effective mixing between mainstream and fluids quite near the walls and secondary flow disturbs boundary layer growth on the side wall due to that the surface heat transfer is improved.

Frictional Losses

Ribs have enhanced heat transfer but also increased pressure drop, generally round edge ribs has less friction compared to sharp edge ribs. The friction factor for semicircular rib channel is presented in Fig. (7). Friction factor shows reducing trend as Reynolds number increased, this is because at the starting obstruction to flow is high as velocity increased retardation to flow reduced. The friction factor for the semicircular rib was greater than those of the plain plate because off large curvature shape of rib creates more turbulence and also roughness component delay the flow and creates excess pressure drop. The semicircular rib has an average 200% more friction compared to a plain plate, semicircular rib with rib pitch ratio (P/e) 8 has more friction loss compared to other tested configurations. At rib pitch 40 mm (P/e=8), the distance between two ribs is less and also number of ribs on test plate increased due to that more retardation & disturbance created in test plate, as a result of this friction increased. Also observed larger separation zones and the additional form drag of having thicker boundary layers between the two ribs. When rib pitch is 60 mm (P/e =12) number of ribs on test plate reduced and a distance between two ribs more due to that it offers smaller separation zones as a result found less friction compared to other tested cases. In case of rib pitch 50 mm (P/e =10) channel offers medium friction means less than 40 mm rib spacing and more than 60 mm rib spacing, here medium flow turbulance observed because optimum distance maintain between two succeeding ribs. For friction factor rib height was also important parameters as rib spacing.

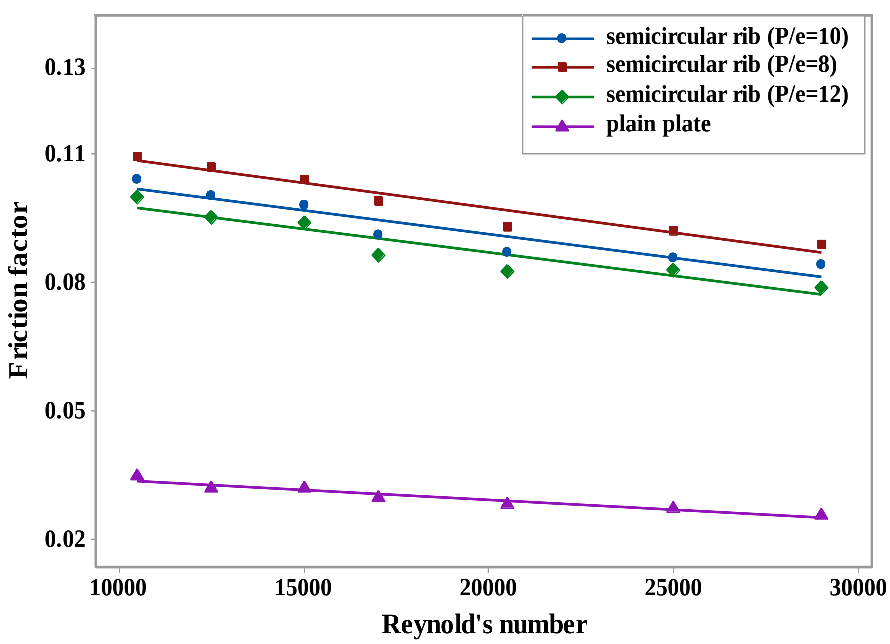

Figure 7. Reynold’s number vs Friction factor for semicircular rib.

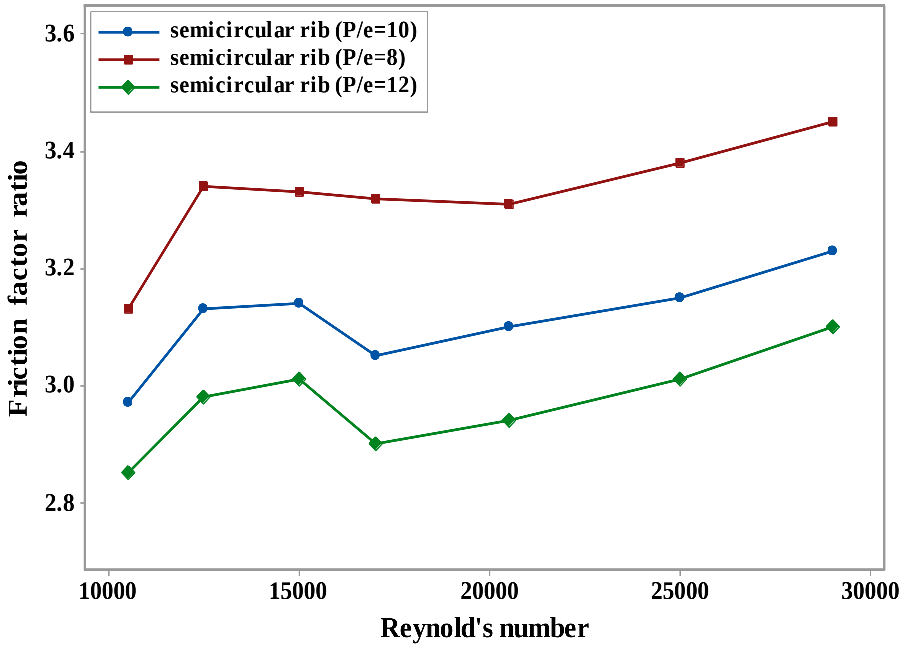

Figure 8. Variation of friction factor ratio against Reynold's number.

Fig. (8) Shows result of friction factor ratio for semicircular ribs. Friction factor ratio from 10000 to 15000 Reynolds number shows increasing trend after that decreasing but at Reynolds number 25000 onwards show increasing trend this is because of elimination of viscous sublayer with a rise in Reynolds number, as Reynolds no. increased boundary layer thickness decreased cause of increase in fluid velocities in the boundary layer. Semicircular rib with pitch ratio (P/e = 8) has more friction compared to other spacing ribs, enhanced 7 to 12 % more friction compared to pitch ratio P/e = 10 &12. Semicircular rib with 60 mm (P/e =12) spacing found less friction over 40 & 50 mm rib spacing because as spacing between rib more retardation to flow becomes weaker due to that friction found less.

Generalized correlations of experimental data

The least square method was used to generalize the current results, also the Nusselt number ratio and friction factor ratios were correlates in terms of rib spacing ratio (P/e) and Reynolds number for the semicircular rib with different rib spacing configurations. The thermal efficiency index criterion was used to estimate the heat transfer enhancement of ribbed surfaces. The correlations for semicircular rib presented by Alfarawi et al.2017 was used to generalize the experimental results. All the correlations are summarized below and they are valid for a range of Reynolds number from 10000 to 29000, within a range of (P/e) from 8 to 12 for semi-circular rib duct. For Nusselt number ratio correlation (9) was used, found max. ±13% deviation between experimental results and correlation results.

- $Nu/Nus = 10{:}07\ Re^{-0.164}[P/e]^{-0.0595}$ --------------------- (9)

Same way for friction factor ratio, correlation (10) was used, found max. ± 10% deviation, also for thermal performance or efficiency index equation (11) was used, found max. ± 9.65% deviation.

- $f/fo = 9.33\ Re^{-0.041}\ [P/e]^{-0.28}$ ------------------------ (10)
- $\eta_i = 4.73\ Re^{-0.15}\ [P/e]^{0.034}$ ------------------------------ (11)

Validation of ribbed duct results through CFD

Experimental results of semicircular rib are validated through numerical results, for simulation STAR-CCM version 2019 software was used. In this study the realizable k–ε turbulence model selected and results of turbulence model are validated by comparing its results with previous paper results. Figure 9 presents the Top and side wall velocity contours of the semicircular rib, found high turbulent kinetic energy appears at the middle region between two adjacent ribs and in front of the ribs.

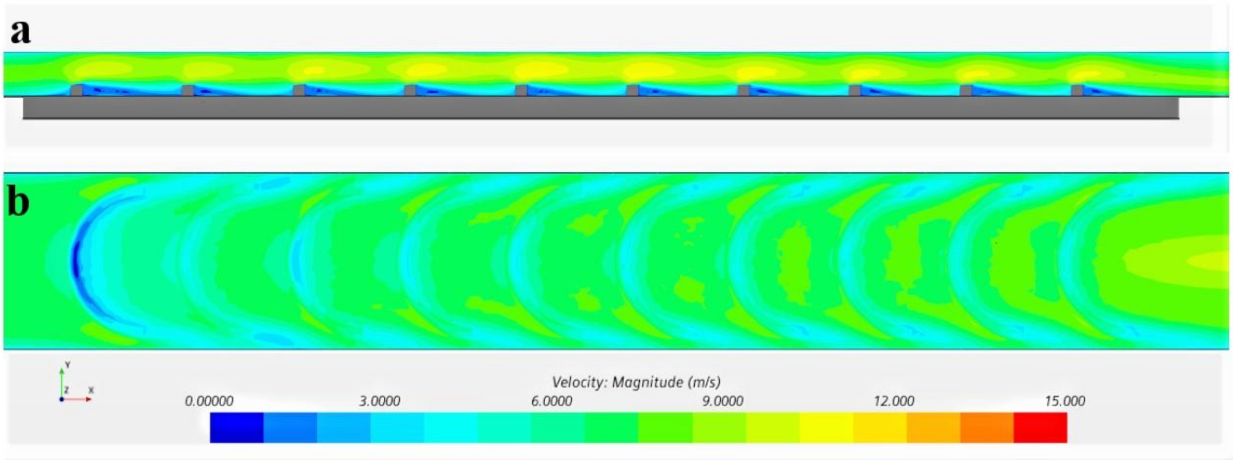

Figure 9. Side and top velocity contours of semicircular rib.

In experimental results semicircular rib with pitch ratio (P/e=10) performed best same results are observed in simulation.Fig.10 &11 shows variation in experimental and numerical results of semicircular rib (P/e = 10). From fig. found that less variation between the numerical and the experimental results, the differences for the Nusselt number and friction factor values between the numerical and the experimental results are less than ± 12%.

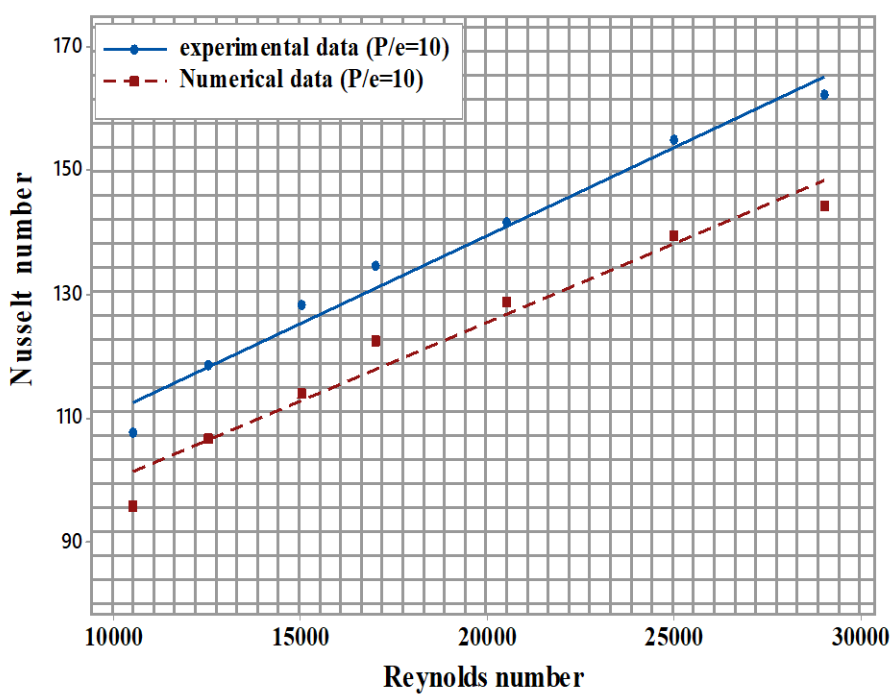

Figure 10.variation of expt. and numerical results of Nusselt number.

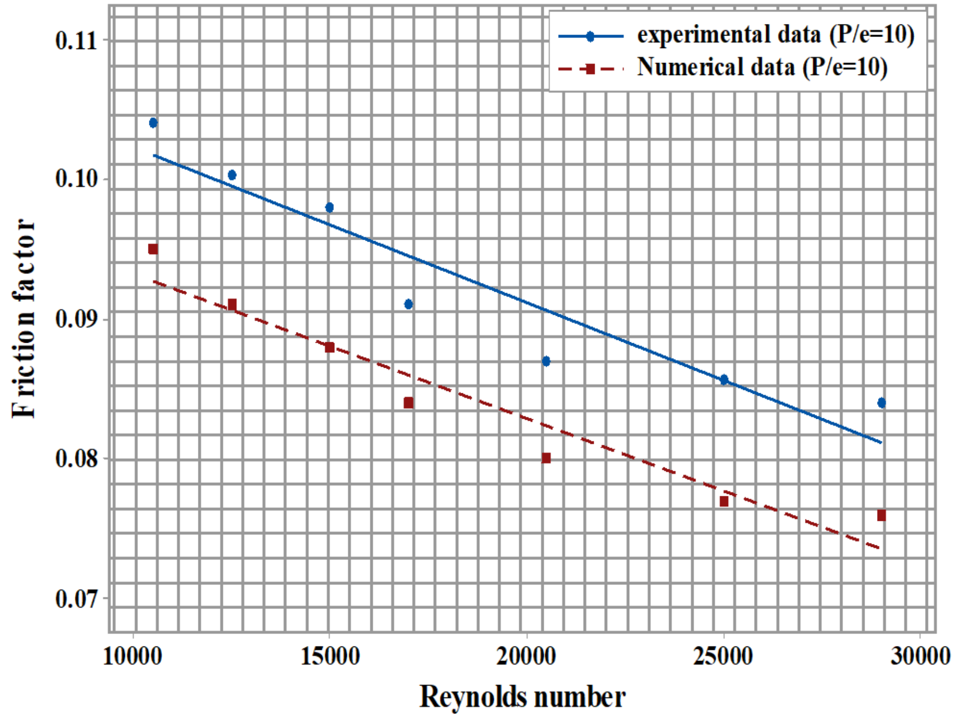

Figure 11. Variation of expt. and numerical results of friction factor.

Thermal Performance (T.P.)

For calculation of thermal performance frictional losses are united with Nusselt numbers. Fig. (12) Shows the thermal performance for all tested configurations. The semicircular rib with pitch spacing ratio (P/e) 10 offered the greatest thermal performance compared to all tested configurations. The semicircular rib with P/e =10 shows increased heat transfer enhancement and decreased frictional losses over the other rib spacing cases. Semicircular rib with pitch ratio (P/e) of 8 offered lowest thermal performance compared to other rib spacing configurations.

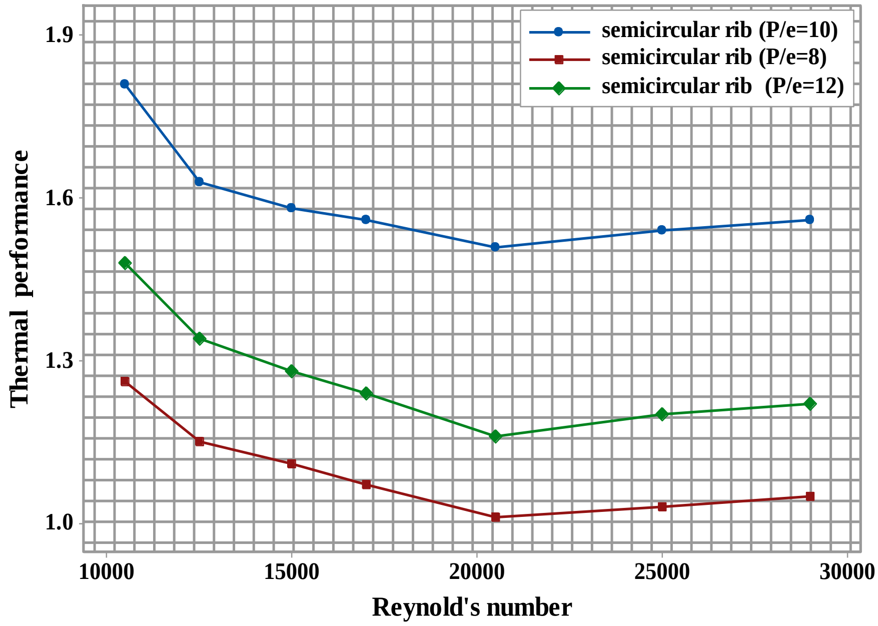

Figure 12. Reynold's number vs thermal performance.

CONCLUSIONS

Heat transfer and flow friction characteristics in a rectangular duct with three rib spacing (P/e = 8, 10, 12) of semicircular rib was experimentally investigated. The experimental results propose the following conclusions:

- Semicircular rib performed better compared to plain duct, enhanced up to 130% heat transfer compared to plain duct.
- Semicircular rib with pitch to height ratio (P/e) 10 has highest thermal performance, found average 40% heat transfer enhancement compared to rib spacing ratio of 8 &12.
- Friction in semicircular rib was average 200 % more than plain plate, found highest in semicircular rib with pitch to height ratio (P/e) of 8 compared to rib spacing ratio of 10 & 12.
- Semicircular shape rib with pitch to height ratio (P/e) 8 has least thermal performance compared to other tested rib spacing.

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