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Investigation of gas turbine blades cooling using triangular cross-section channel provided with ribs

دراسة تبريد ريش التربينة الغازية باستخدام مجرى مثلث المقطع مزود بعوارض

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KEYWORDS:
Gas turbine blade cooling, Heat transfer, Equilateral triangular ribbed channel

***الملخص العربي:*-تم عمل دراسة عددية لإنتقال الحرارة خلال قناة مقطعها مثلث متساوى الأضلاع طول ضلعه 2 سم وسطحها الداخلى مزود بعوارض وذلك لمحاكاة التبريد الداخلى لريشة التربينة الغازية وهذه القناة مواجهة لخليط الإحتراق مباشرة وتم عمل هذه المحاكاة للقناة مع نوعين مختلفين من العوارض.جدار واحد مزود بعوارض عمودية على اتجاه سريان المائع ،جدارين مزودين بعوارض عمودية على اتجاه سريان المائع مع اختلاف النسب الباعية بين العوارض بقيم (5,7.5,10,12.5). تمت الدراسة فى مدى رقم رينولدز من 20000 إلى 70000. تم استخدام حزم برنامج (Fluent)6.3.26 والذى يعتمد على طريقة الحجوم المحدودة لحساب السريان وإنتقال الحراره وأثبتت النتائج أنه فى حالة السكون فإن معامل انتقال الحرارة فى حالة استخدام جدارين ملصق بهما عوارض أعلى من معامل انتقال الحرارة فى حالة استخدام جدار واحد ملصق به عوارض وكانت النتيجة أن أفضل مسافة باعية بين العوارض هى 10 فى حالة الجدار الواحد الملصق به عوارض فى حالة السكون والنسبة الباعية 7.5 فى حالة الجدارين الملصق بهم عوارض فى حالة السكون والدوران بسرعه 3000 لفة فى الدقيقة.**

***Abstract*—Heat transfer has been investigated in an equilateral triangular channel with slide length 2 cm, hydraulic diameter D_h = 1.1547 cm with ribs fixed to the internal surface of the channel to simulate the internal cooling channel near the leading edge of the gas turbine blade. The simulation has been done for two different ribs configurations (one wall ribbed and two walls ribbed) with different pitch ratios (p/e) of 5, 7.5, 10, 12.5, Reynolds numbers varied from 20,000 to 70,000 at stationary and rotating channel with speed of 3000 rpm, with rotational number range from 0.39 to1.37. A numerical investigation is conducted using a commercial package (Fluent 6.3.26) where the flow and heat transfer characteristics were investigated. The results show that, the heat transfer coefficient for two walls ribbed is better than for one wall ribbed channel at stationary condition. Also, from the results it is concluded that the best heat transfer occurs at pitch ratio of 10 for one wall ribbed and stationary channel, while the two walls ribbed gives the best heat transfer at pitch ratio of 7.5 either for stationary or rotating channel.**

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I. INTRODUCTION

THE gas turbine blade/vane internal cooling is achieved by circulating the compressed air through the cooling passages inside the turbine blade. Leading edge of the turbine blade is critical region which needs to be properly cooled. Leading edge region receives extremely hot mainstream flow and high heat transfer enhancement is required. Reviews on heat transfer augmentation techniques applied in gas turbine blade internal cooling are given in Ligrani (2013) and Gupta et al. (2012).

Han (1988) studied the effect of the channel aspect ratio on heat transfer for five different channel aspect ratios (AR=1:4, 1:2, 1:1, 2:1, 4:1). He concluded that the local heat transfer in the narrow aspect ratio channel was higher than the wide aspect ratio channel.

Han et al. (2000) provided in-depth information about updated cooling techniques. The internal cooling techniques of the gas turbine blade includes: jet impingement, rib turbulated cooling, and pin-fin cooling. Their study of the gas turbine blade internal cooling begins from the stationary, rectangular cooling channels with the ribs placed on the walls of the cooling passage. Ribs trip the boundary layer of the coolant flow and enhance heat transfer at the higher pressure drop, the rib effects in the stationary channel as well as in the rotating channel have been done by several groups. Rib spacing, rib height, rib angle, and the shape of the ribs all affect the heat transfer enhancement.

Han et al. (2001) conducted experimental and numerical investigation of heat transfer of internal cooling channels near the trailing and leading edges for a stationary and rotating blade. Aspect ratios of 4:1 and 8:1, Reynolds number ranged from 5000 to 40000 and rotation number from 0.04 to 0.3 were considered. A 45 º rib turbulators were attached to the trailing and leading surfaces while the channel orientation angle was kept at 90 º and 135 º with respect to the plane of rotation. The results declared that the heat transfer differences for the smooth tilted channel was up to 25% while for the ribbed channel was 50-75%. Also, the channel orientation significantly affects the leading, the outer, and the inner surfaces and did not have much effect on the trailing surfaces.

Fu et al. (2004) studied the heat transfer in two-pass rotating rectangular channel with 45º rib walls for two aspect ratios of 1:2 and 1:4. The Reynolds numbers ranged from 5000 to 40000 while the rotation numbers varied from 0.0 to 0.3. Their studies considered two channel orientations with angles of 90 º and 45 º with the plane of rotation. Their results showed that the heat transfer on the trailing wall increased in the first pass due to the rotation effect, while on the leading wall the heat transfer reduced and the minimum recorded heat transfer coefficient was 25% of that of the stationary channel.

Liu et al. (2006) studied the rib spacing in a 1:2 aspect ratio channel under rotating conditions for the considered four p/e ratios: 3, 5, 7.5 and 10. The rotating speed was fixed at 550 RPM with the channel orientation at β =90°.The Nusselt number increased as p/e decreased. However, the friction factor also increases with decreasing p/e ratio until a p/e ratio of 5. As the p/e ratio further decreased to 3, the friction factor was reduced. Thus, the p/e = 3 case showed the best thermal performance. The results also showed that the highest thermal performance was p/e = 5 for the stationary case and p/e = 7.5 for the rotating case.

Liu (2008) experimentally studied heat transfer and pressure drop in an equilateral triangular channel. Three different rib configurations (45°, inverted 45°, and 90°) were tested at four different Reynolds numbers (10,000-40,000), each with different rotational speeds (0-400 rpm). He concluded that 45° angled ribs showed the highest thermal performance at stationary condition and 90° ribs have the highest thermal performance at the highest rotation number of 0.58.

Lee et al. (2014) performed experiments on heat transfer and friction factors of fully developed turbulent flows in the stationary rectangular divergent channel with different parallel angled ribs (α=30º,45º, 60º, and 90º) are placed to the channel's two opposite walls as well as to the channel's one sided wall only, respectively. The ribbed rectangular divergent channel has the inclination angle of 0.72º at the left and right walls, corresponding to D_{h_o}/D_{hi} = 1.16. The divergent channel has the cross sections of 100 x 75 mm^2 at inlet and 100 x 100 mm^2 at exit. The ribbed walls are manufactured with a fixed rib height (e) = 10 mm and the ratio of rib spacing (p) to height (e) = 10. The Reynolds numbers are from 15,000 to 89,000.They concluded that the total friction factors in the 90º angled ribbed divergent channels are somewhat lower than in the straight ribbed cross-sectional channels; however, the Nusselt numbers are a little greater than in the ribbed straight cross-sectional channels.

The objective of the present study is to get the best configuration of an equilateral triangular channel on the leading edge with ribs at different pitch ratios to achieve the best thermal performance for the cooling of gas turbine blade.

II. 2. COMPUTATIONAL MODEL AND BOUNDARY CONDITIONS

Figure (1) summarizes the dimensions and boundary conditions used in the model. A schematic diagram shows a channel of an equilateral triangular section of side equal to 20 mm, the total length of the channel, L, equals to 374.65 mm and the channel hydraulic diameter D_h is 11.54 mm. The distance from the inlet of the channel to the axis of rotation (X-axis), is Rr = 336.55 mm. The inlet mass flow rate for the periodic segment is corresponding to the Reynolds number which varied from 20,000 to 70,000; with inlet temperature of 773 K. Turbulence intensity varied from 3.97 to 4.64% is used at the inlet. This is calculated by the relationship provided by ANSYS Fluent 6.3.26. All walls were kept at a constant temperature of 1273 K. The ribs are placed on one wall and two walls. The flow is considered incompressible, three-dimensional, turbulent and steady with constant thermodynamic properties. The working fluid is dry air with Prandtl number equals to 0.71. Ideal gas law is used to calculate density variation with temperature. The grid independence study and validations of numerical results for both types of channels are presented separately.

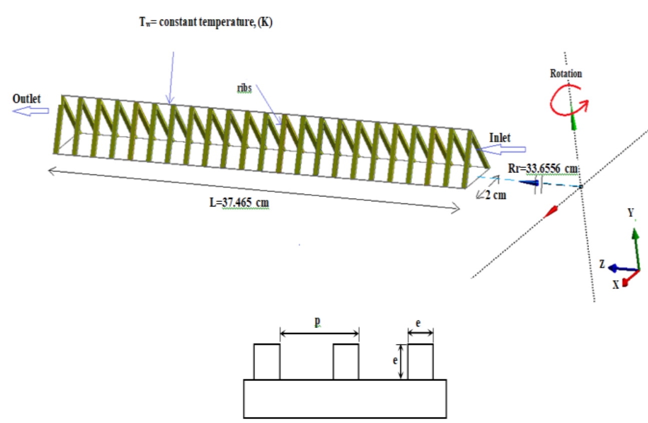


Figure (1) Cooling channel configuration

III. Data Reduction

The present study investigates the regionally averaged heat transfer coefficient at various locations within the rotating and stationary triangular ducts. The heat transfer coefficient is determined by the net heat transferred from the heated wall, the surface area of the walls, the regionally averaged temperature of the wall and the mean temperature of the channel. Therefore, the heat transfer coefficient is given as:

$$h = \frac{Q/A_{sur}}{(T_w - T_f)} \quad (1)$$

Where:

$$Q = \dot{m} C_p (T_o - T_i), T_f = \frac{T_o + T_i}{2}$$

A_{sur} Smooth = (Triangle perimeter (Pr) *Total internal Channel Length)

A_{sur} Ribbed = Triangle Perimeter (Pr)* (Channel Length + (Rib side length*2* Number of Ribs)) = (Pr)*(L+ 2nL$_1$)

T_f = cooling fluid average temperature, T_w = wall mean temperature.

In the present study, Leung et al. (1998) correlation is used to provide a basis of comparison. This relation used to calculate the Nusselt number (Nu_o) for stationary triangular duct with smooth internal surfaces, and is given as:

$$\overline{Nu} = 0.015 \mathrm{Re}^{0.82} \quad (2)$$

Therefore, the heat transfer enhancement ($\overline{Nu}/Nu_o$) is given as:

$$\frac{\overline{Nu}}{Nu_{\circ}} = \frac{(\mathrm{hD_h/k})}{0.015\ \mathrm{Re}^{0.82}} \quad (3)$$

Where ($\overline{Nu}$) is the regionally averaged Nusselt number and $D_h = \frac{4\ A}{\mathrm{Pr}}$

The friction factor is calculated using the values of the inlet (P_i) and outlet (P_o) pressures as, Fu (2005):

$$f = \frac{P_i - P_{\circ}}{4(L/D_h)(\frac{1}{2}\rho v^2)} \quad (4)$$

The turbulent friction factor given by Blasius correlation is given as:

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

The friction factor ratio has been defined by Eq. (6).

$$\frac{f}{f_o} = \frac{P_i - P_{\circ}}{4(L/D_h)(\frac{1}{2}\rho v^2)} \left(\frac{1}{0.079} \mathrm{Re}^{0.25}\right) \quad (6)$$

One way to evaluate the performance of different ribbed channels is to calculate the thermal performance for each ribbed channel configuration where the performance ratio is defined as, Fu (2005):

$$\eta = (Nu/Nu_{\circ})(f/f_{\circ})^{\left(-\frac{1}{3}\right)} \quad (7)$$

The strength of Coriolis force depends on the velocity of the coolant (v), the channel geometry (D_h), and the rotational speed of the channel (Ω). These factors can becombined to define the rotation number, Ro, as follows:

$$Ro = \frac{\Omega * D_h}{v} \quad (8)$$

IV. Numerical Modeling

FLUENT6.3.26 code is used to simulate and solve the heat transfer problem. The first step of solving this problem is the geometrical configuration which is established by the fluent drawing tool known as GAMBIT which is used to create the model. The sequences of steps involved in GAMBIT are shown in Figure 2.

Figure 3 shows the duct geometry and the numerical grid generated using GAMBIT grid generator for this study. After the geometry exported to FLUENT 6.3.26 then the problem specification scale, boundary conditions, solution, and errors are entered to Fluent according to the present problem. Figure (4) shows the steps involved in solving the problem in Fluent.

V. Grid Independency Test

To choose a suitable computational grid, a grid independency test is performed using different grid sizes in the solutions. The numbers of cells used in the present analysis are 18,767 & 30,865 & 60,909 & 466,299 & 720,419 & 939,014 and 1,349,797 cells in stationary smooth channel. The parameter used to check the grid independency is the mean Nusselt number. Comparisons of the mean Nusselt number are shown in Figures 5 and 6. The maximum difference in Nusselt number is 5.52% between (18,767) and (30,865) grid points and the minimum difference in Nusselt number is 0.35% between 939,014 and 1,349,797 grid points. Therefore, all the results presented in the following discussions are based on the (1,349,797) grid points.

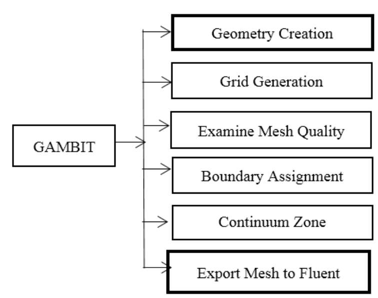


Figure (2) Steps involved in exporting mesh file from GAMBIT

Figure (3) Computational grid of an equilateral triangular channel with ribs

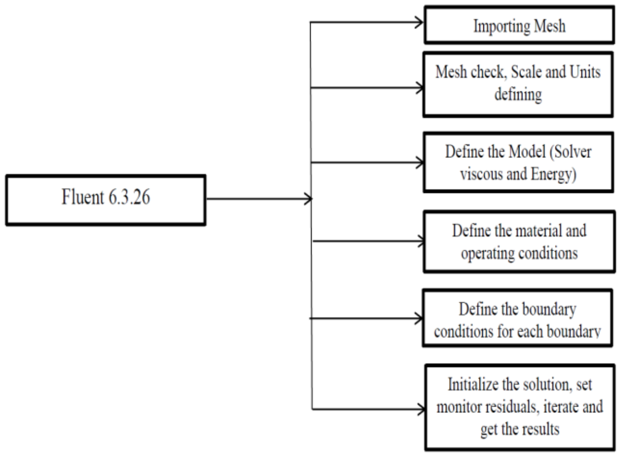


Figure (4) Steps involved in solving the problem in Fluent

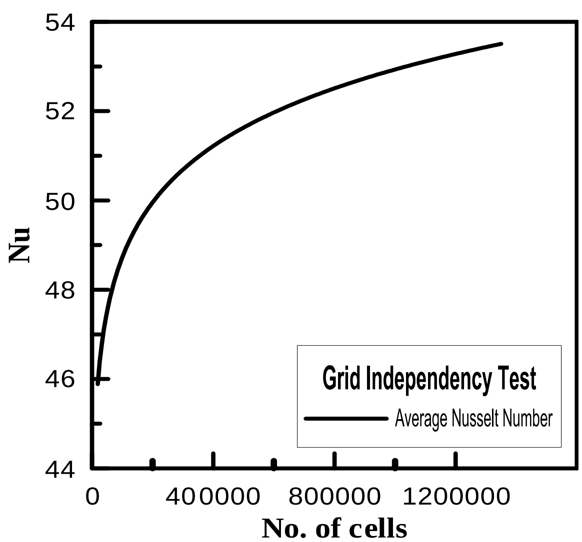


Figure (5) Averaged Nusselt number for different grids

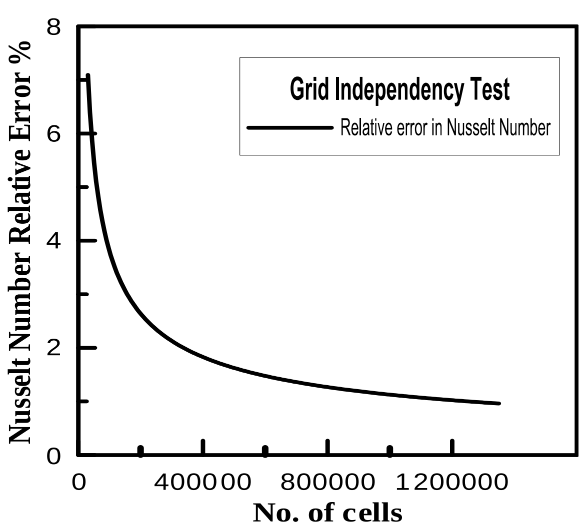


Figure (6) Number of cells versus percentage of error in averaged Nusselt number

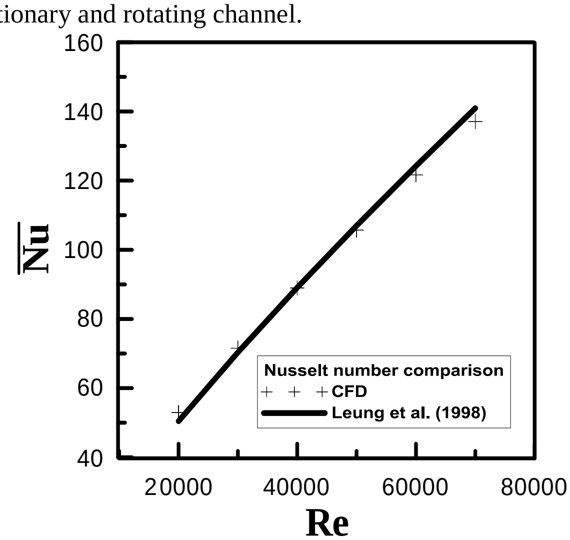
VI. Results and Discussion

6.1 Results Validation

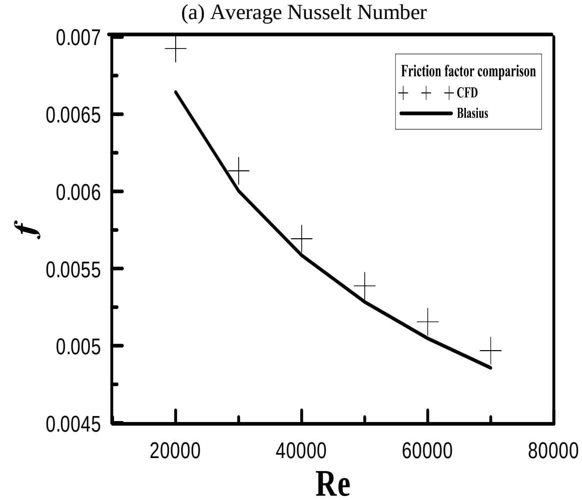
In order to validate the results of Computational Fluid Dynamics (CFD) model used in the present study, comparisons were performed for the average Nusselt number and friction factor. The previous works are performed in case of smooth equilateral triangular channel. Figure (7) shows a reasonably good agreement of these comparisons with the previous work of Leung et al. (1998). The comparison of the average Nusselt number results as shown in Figure (7-a), shows that the maximum difference in average Nusselt number is (4.9%) at low Reynolds number and (2.7%) at high Reynolds number. Also, the present study friction factor results are compared with the Blasius correlation (Cengel, 2006), as shown in Figure (7-b) and the maximum deviation of friction factor is 4% at low Reynolds number and 2.2% at high Reynolds number.

In the present study, the results are considered for one-wall and two-wall ribbed equilateral channel, respectively, at different pitch ratios of (5, 7.5, 10, and 12.5) and Reynolds number ranged from 20,000 to 70,000.

The heat transfer and pressure losses are investigated for stationary and rotating channel.



(a) Average Nusselt Number



(b) Friction Factor

Figure (7) Comparison between present study and previous studies of stationary smooth equilateral triangular channel at different Reynolds numbers

6.2 Stationary channel

6.2.1 The case of one wall ribbed channel

Figure 8 depicts the effect of pitch ratio (p/e) on average Nusselt number ($\overline{Nu}$) for one wall ribbed. As shown, theNusselt number increases with the increasing of pitch ratio till its maximum at p/e = 10 and then decreases with further increasing of the pitch ratio. It may be due to the effect of the pitch ratio on the turbulent flow formation between ribs. A flow separation appears at the tips of ribs causing a circulation zone between ribs. For the low pitch ratio, the circulation zones are created without reattaching points between ribs. As the pitch ratio increases the area of circulated flow increases which enhance the heat transfer. By increasing the pitch ratio, the opportunity of formation a reattaching point between ribs increases which leads to further enhancement till a formation of dead zones appears. By increasing the dead zone formation, the heat transfer decreases with increasing pitch ratio as shown in Figure 9 which agrees with Sundberg (2006) flow behavior.

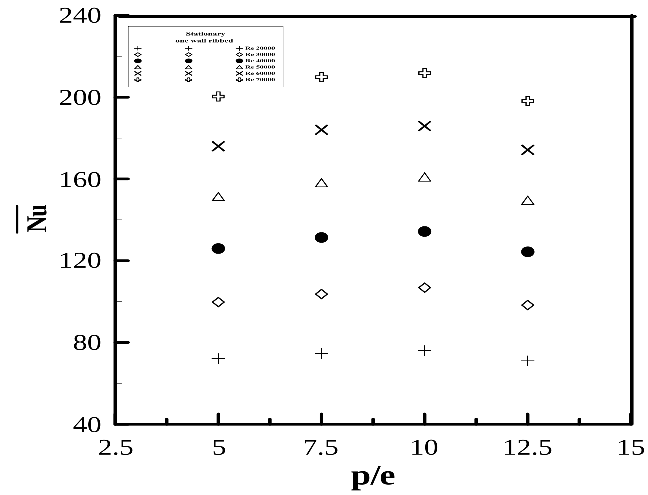


Figure (8) Nusselt number vs. rib pitch ratio for one wall ribbed channel at different Reynolds numbers

Figure 10 represents the variation of friction factor ratio ($f/f_{\circ}$) with the pitch ratio variation. The figure indicates that the friction factor ratio increases with the pitch ratio increasing until it reaches its maximum value at p/e = 10 and then decreases with further increasing in rib pitch ratio. Also, it is clear that the friction factor ratio increases with the increasing of Reynolds number.

The effect of pitch ratio (p/e) on the thermal performance (η) of the channel cooling is illustrated in Figure 11. It can be noticed that, the thermal performance increases as the pitch ratio increases until it reaches its maximum value at p/e = 10 and then decreases with further increasing of the pitch ratio. Also, it’s seen from the figure that, the thermal performance decreases with the increase of Reynolds number due to the increasing of friction losses over the increasing of heat transfer enhancement as the Reynolds number increases.

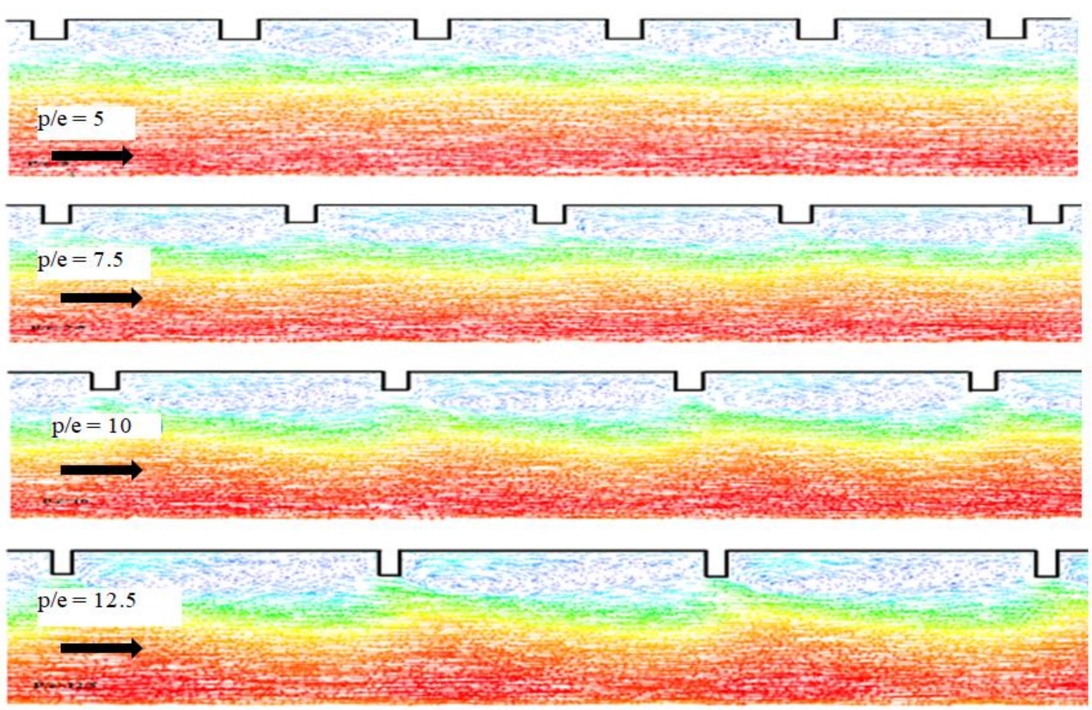


Figure (9) Velocity vectors for an equilateral triangular channel with one wall ribbed at different pitch ratios

6.2.2 The case of two walls ribbed channel

Figure (12) shows the relation between average Nusselt numbers ($\overline{Nu}$) and pitch ratio (p/e). The figure indicates that the average Nusselt number increases with the increasing of pitch ratio until it reaches its maximum value at p/e = 7.5 and then decreases with further increases in rib pitch to height ratio. It is noticed that, for the two walls ribbed, the maximum Nusselt number is obtained at pitch ratio lower than that of one wall ribbed, which reflects the effect of turbulence intensity over the pitch ratio variation.

Figure (13) depicts the effect of pitch ratio (p/e) on the friction factor ratio ($f/f_{\circ}$). As observed from the figure, the friction factor ratio increases with the pitch ratio increasing until it reaches its maximum at

p/e =7.5 and then decreases with further increasing of the pitch ratio.

Figure (14) shows the relation between the thermal performance (η) and pitch ratio (p/e). The figure indicates that the thermal performance increases with the increasing of the pitch ratio and the maximum thermal performance value occurs at p/e = 12.5.

6.2.3 Comparison between smooth, one wall and two walls ribbed channel

As shown in the previous results, the best Nusselt number occurs at pitch ratios of 10, 7.5 for one and two walls ribbed channel, respectively. Figure (15) shows a comparison between the best pitch ratios for one-wall, and two-wall ribbed channel with smooth channel at different Reynolds numbers. The figure indicates that the one wall ribbed channel at pitch ratio of 10 gives an enhancement of approximately 42% when compared with the smooth channel. The two walls ribbed channel enhances the heat transfer in the range of 72.3% to 82.4% according to Reynolds number variation when compared with the smooth channel.

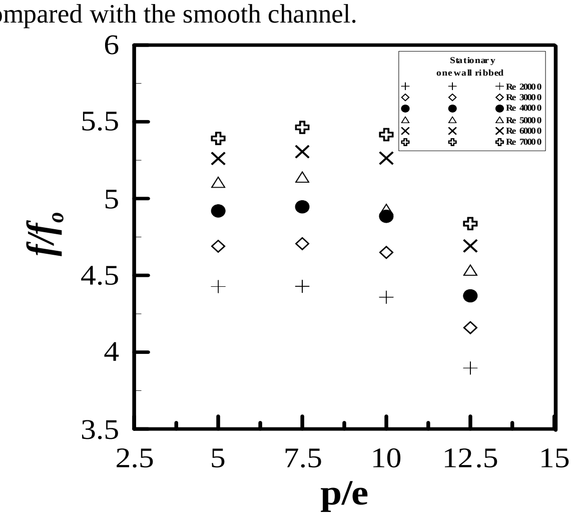


Figure (10) Friction factor ratio versus rib pitch ratio for one wall ribbed channel

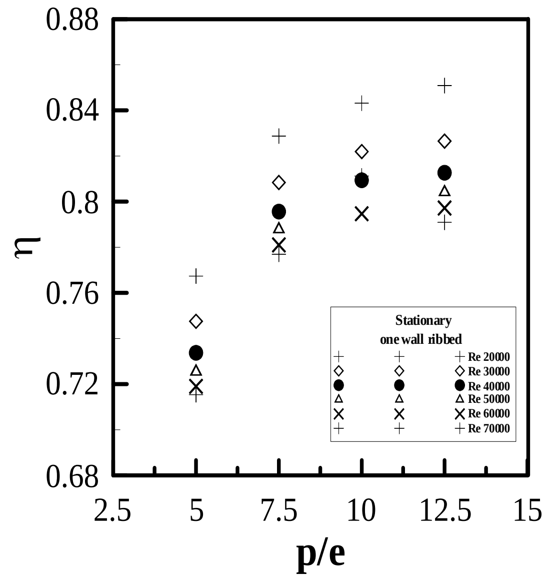


Figure (11) Thermal performance versus rib pitch ratio for one wall ribbed channel

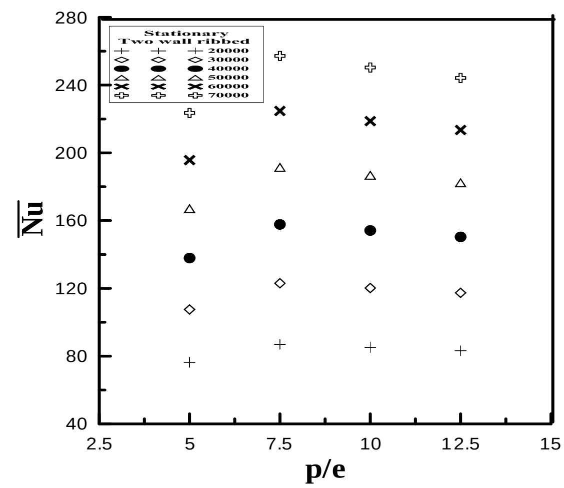


Figure (12) Mean Nusselt number versus rib pitch ratio for two walls ribbed channel

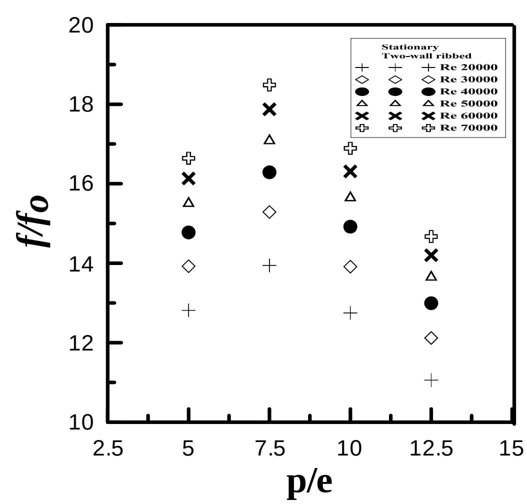


Figure (13) Friction factor ratio versus rib pitch ratio for two walls ribbed channel

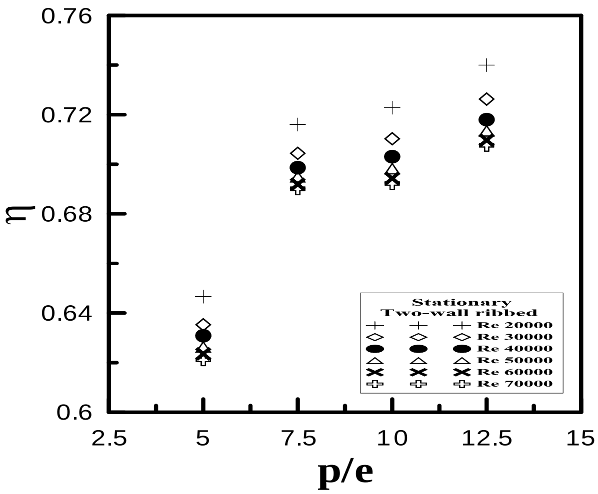


Figure (14) Thermal performance versus rib pitch ratio for two walls ribbed channel

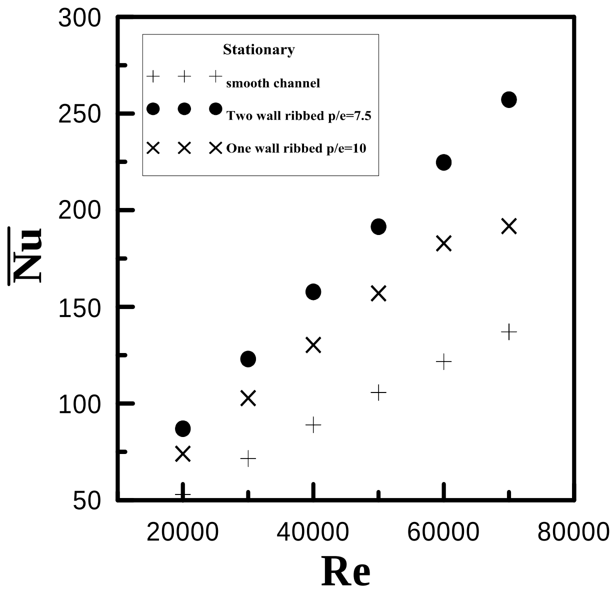


Figure (15) Comparison between averages Nusselt number of smooth, one wall ribbed and two walls ribbed channel at various values of Reynolds number

6.3 Rotating channel

6.3.1 Smooth channel at rotating speed of 3000 rpm

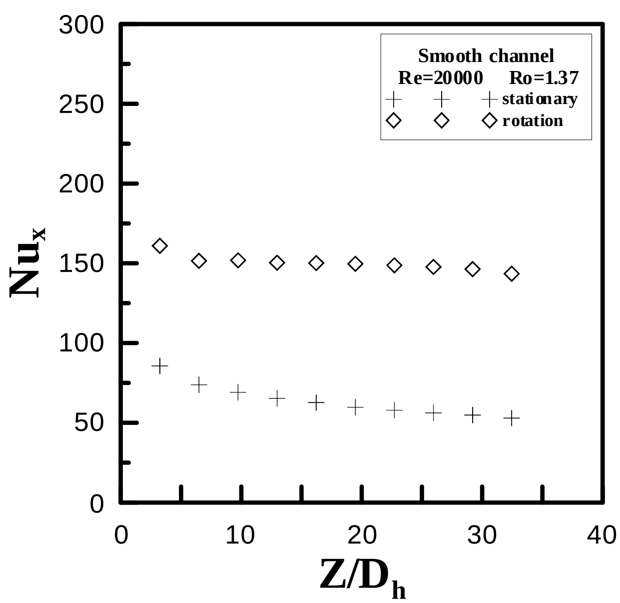
The behavior of flow and heat transfer are investigated and the comparison with the stationary smooth channel at a Reynolds number ranges from 20,000 to 70,000, rotation number ranges from 0.39 to 1.37 and for the rotating channel with a speed of 3000 rpm.

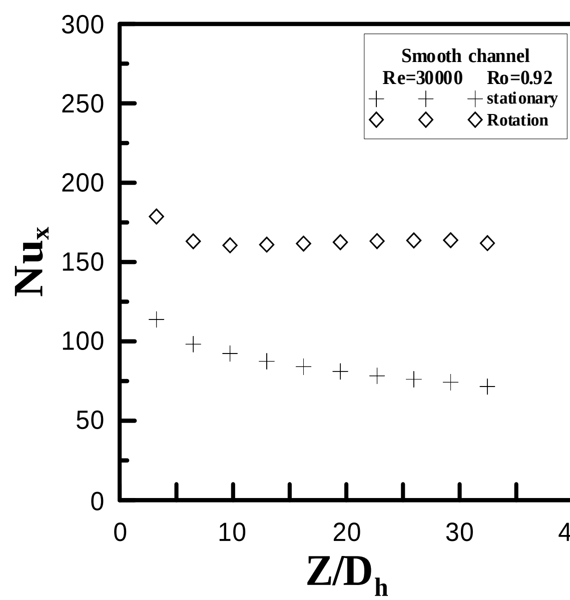
Figure (16) represents the variation of local Nusselt number (Nu_x) along the stream direction (Z/D_h). It's clear that the local Nusselt number decreases along the streamwise direction due to the boundary layer development.

Figure (17) depicts the effect of rotation on heat transfer at different Reynolds numbers (Re). The figure indicates that: in case of rotation, the heat transfer is enhanced through the channel. This enhancement may be due to the effect of generated vortices during rotation. When compared with the stationary to smooth channel, the enhancement is ranged from 47.4% to 184.3% according to Reynolds number variation.

Figure (18) depicts the effect of rotation on friction factor ratio ($f/f_\circ$) with different Reynolds numbers (Re) for smooth channel. It can be seen that, the friction factor ratio decreases more sharply at low Reynolds number than that of stationary channel with the increasing of Reynolds number.

Figure (19) describes the thermal performance of the rotating channel when compared with the stationary channel. The comparison shows an enhancement of the thermal performance from 31.5% to 82.3% according to the Reynolds number variation. Also, it can be noticed that the trend of the thermal performance variation with Reynolds number is affected strongly by the friction factor ratio trend as described by Eq. (7)





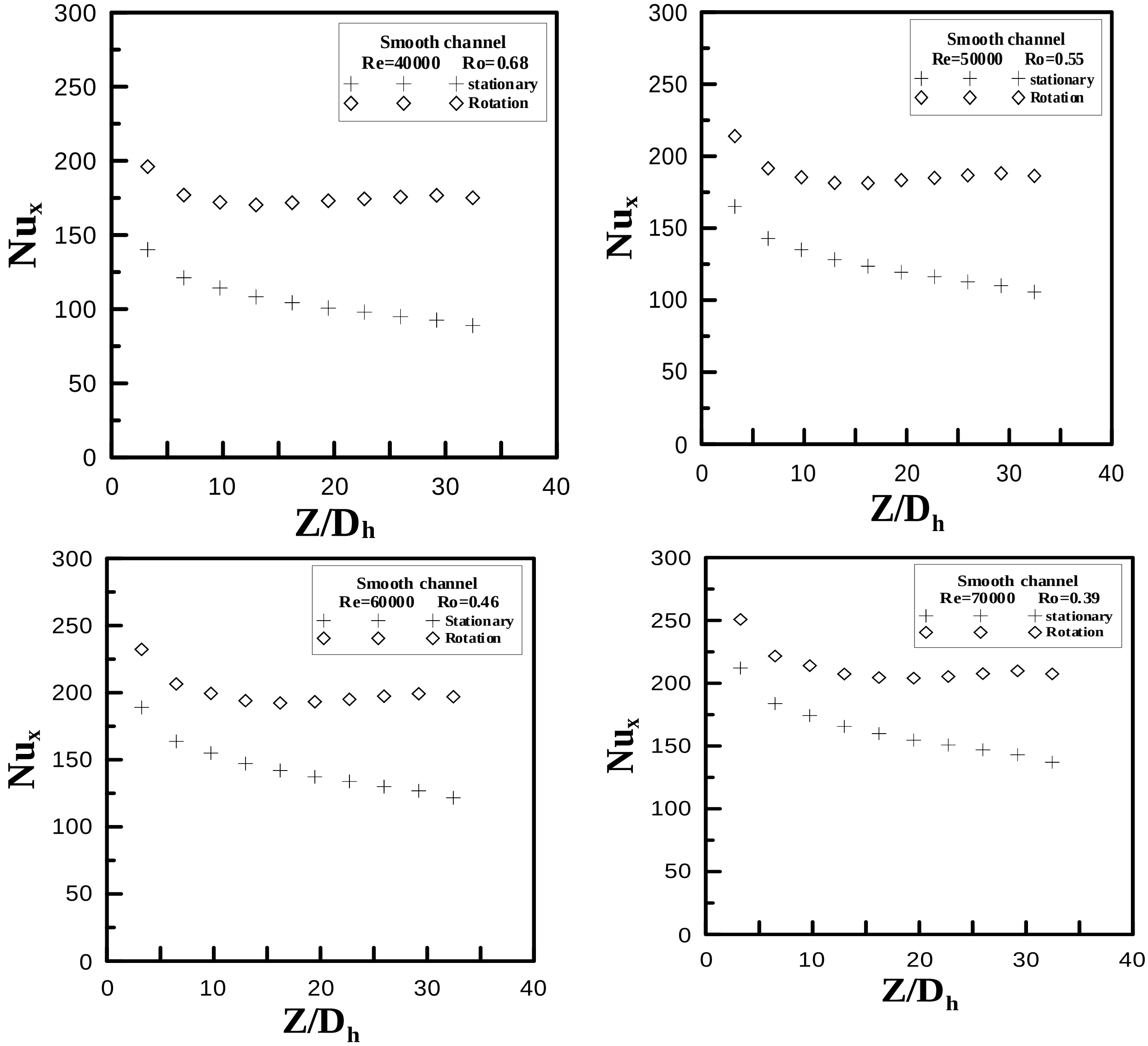


Figure (16) Nusselt number along the stream direction (Z/Dh) for stationary and rotation of smooth channel

6.3.2 Two walls ribbed channel at rotating speed 3000 rpm

From the previous discussion, it can be concluded that the two walls ribbed stationary channel and smooth rotating channel enhance the heat transfer and the thermal performance, so it's important to investigate the case of two walls ribbed and rotational channel.

Figure (20) describes the local Nusselt number (Nu_x) along the stream direction (Z/D_h). It's seen that at different rib pitch ratios the local Nusselt number at entrance has its maximum value at p/e of 7.5 and decreases along the stream direction due to the developed thermal boundary layer.

Figure (21) depicts the effect of pitch ratio (p/e) on average Nusselt number ($\overline{Nu}$) for two-wall ribbed with rotating channel. The figure shows that the average Nusselt number increases with the pitch ratio increasing till it reaches its maximum value at p/e = 7.5 and then decreases with further increase in rib pitch to height ratio.

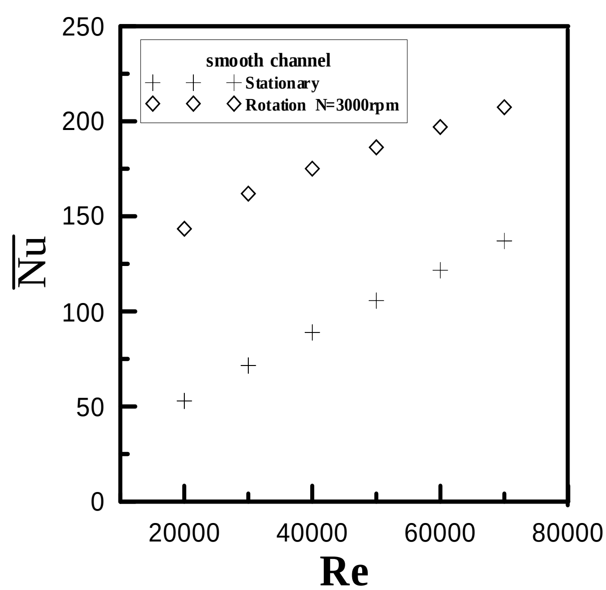


Figure (17) Nusselt number versus Reynolds number for stationary and rotating smooth channel

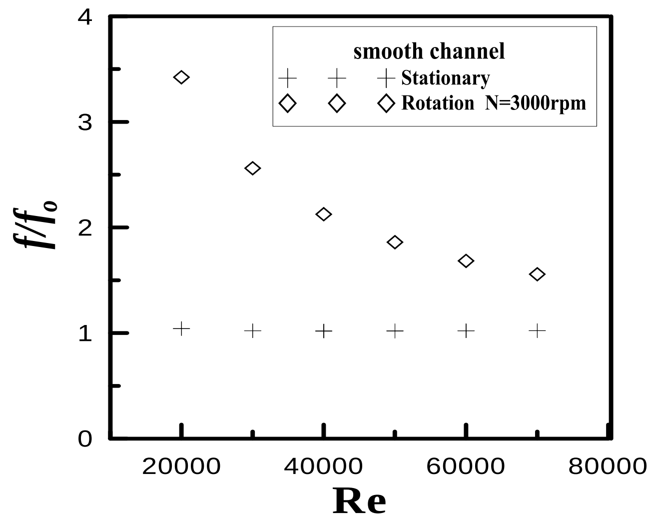


Figure (18) Friction factor ratio versus Reynolds number for stationary and rotating smooth channel

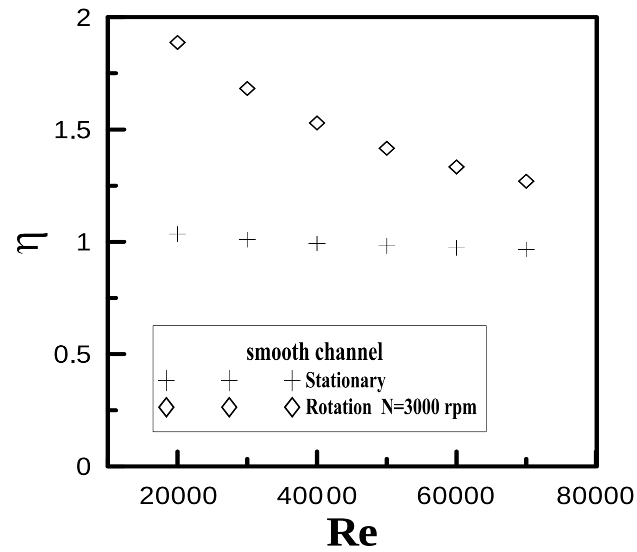
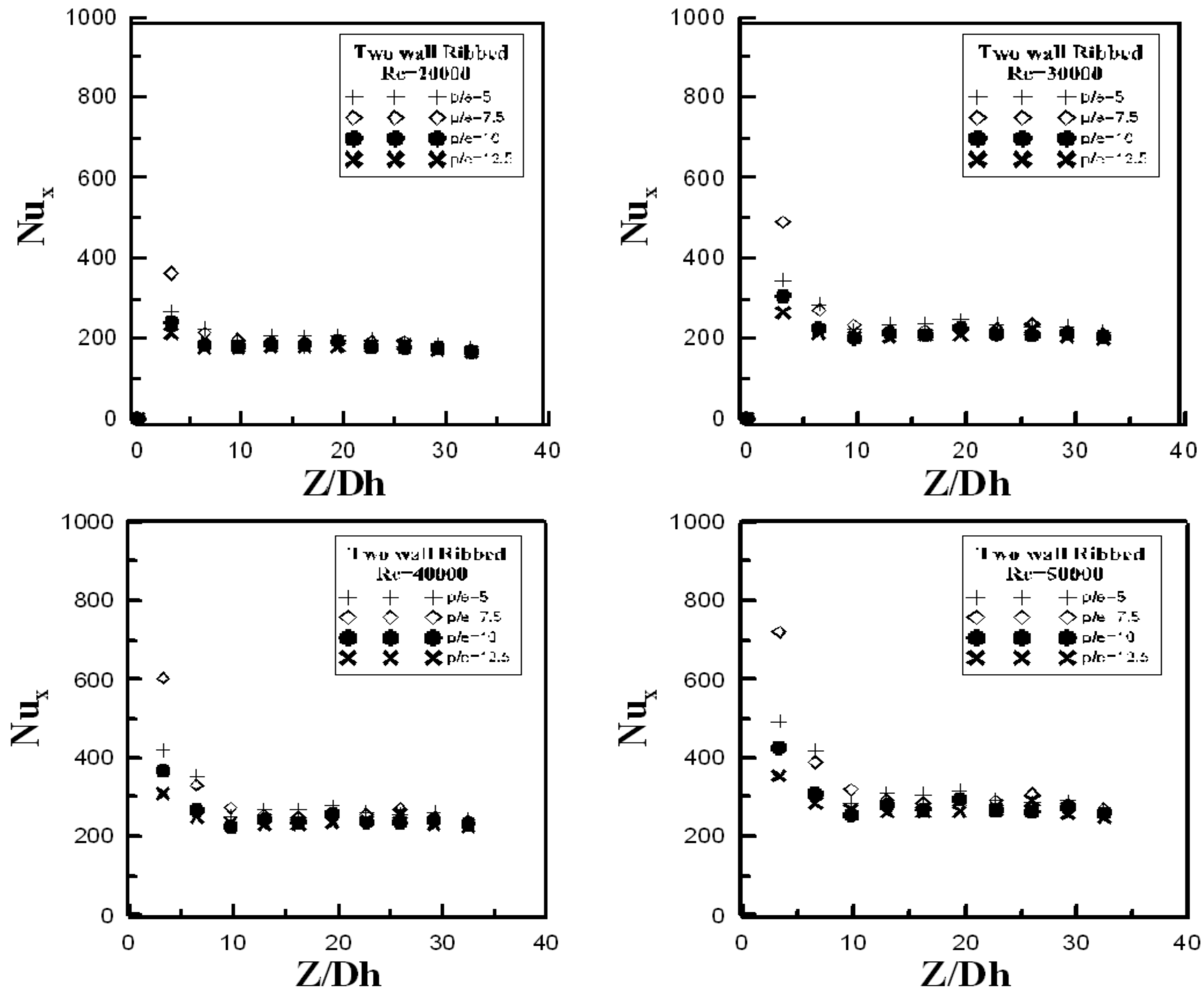


Figure (19) Thermal performance versus Reynolds number for stationary and rotating smooth channel



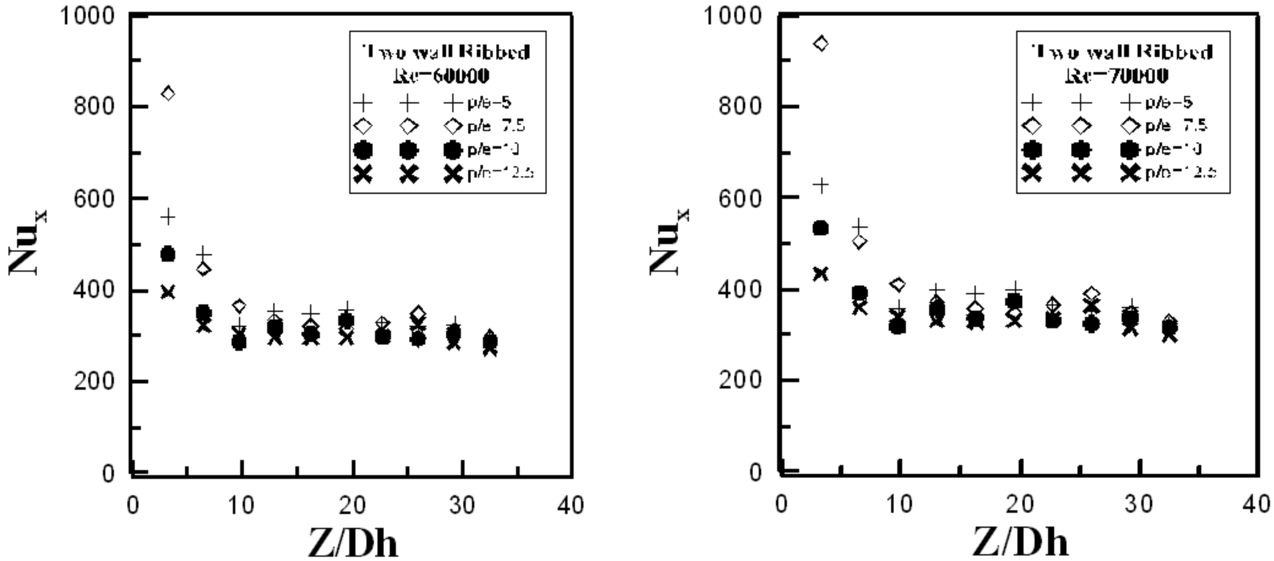


Figure (20) Nusselt number variation for rotating two walls ribbed channel at different pitch ratios

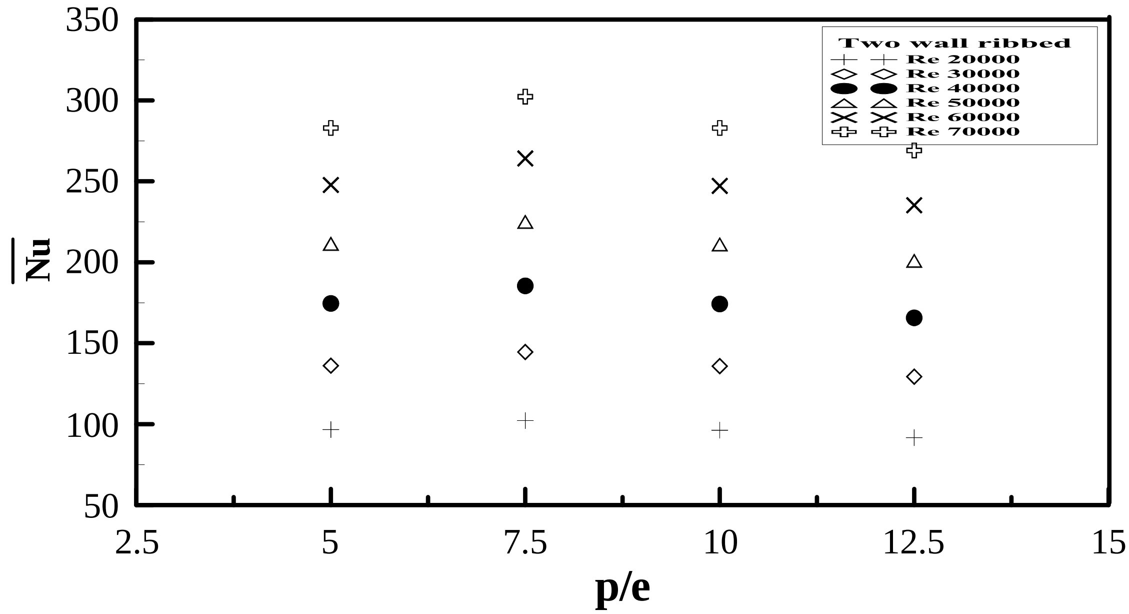


Figure (21) Average Nusselt number versus pitch ratio for rotating two walls ribbed channel at different Reynolds numbers

VII. Conclusion

Investigation of flow and heat transfer in a smooth and ribbed equilateral triangular channels under rotational and stationary conditions are conducted. Computations are performed at Reynolds number (Re) ranging from 20000 to 70000, rotation number (Ro) from 0 to 1.37, pitch ratio of 5, 7.5, 10, and 12.5. It can be concluded that:

1. For one wall ribbed channel, there is an enhancement in the heat transfer by about 42% when compared with the smooth channel.
2. For two walls ribbed channel, the enhancement in the heat transfer is ranged from 72.3% to 82.4% according to the Reynolds number variation, when compared with the smooth channel.
3. The average Nusselt number for one wall ribbed channel at stationary condition increases with increasing the pitch ratio up to 10 and then decreases with further increasing in pitch ratio. Also, the friction factor ratio increases with increasing the pitch ratio up to 10 and then decreases with the increasing of pitch ratio.
4. The average Nusselt number for two walls ribbed channels at stationary and rotating conditions takes its maximum increasing at pitch ratio of 7.5.
5. The thermal performance for one wall ribbed channel at stationary condition increase with the pitch ratio increasing until it reaches its maximum value at p/e = 10 and then decreases with further increase in the rib pitch ratio.
6. Thermal performance decreases with the Reynolds number increase.

For smooth channel with rotation, the results show an enhancement ranged from 47.4% to 184.3% and 31.5% to 82.3% in heat transfer and the thermal performance respectively, when compared with the stationary smooth channel.

REFERENCES

[1] Ansys–Fluent User's Guide, available http://www.arc.vt.edu/ansys_help/flu_ug/flu_ug.html; accessed on 31/01/2015.

[2] Cengel, Y.A., Heat and Mass Transfer, 3rd edition, McGraw-Hill, 2006.

[3] Fu, W.L., "Aspect Ratio Effect on Heat Transfer in Rotating Two-Pass Rectangular Channels with Smooth Walls and Ribbed Walls", Ph.D. Thesis, Texas A&M University, May 2005.

[4] Fu, W.L., Wright, L.M., and Han, J.C., "Heat Transfer in Two-Pass Rotating Rectangular Channels (AR=1:2 and AR=1:4) with 45º Angled Rib Turbulators," ASME Paper No. GT 2004-53261, 2004.

[5] Gupta, S., Chaube, A., and Verma, P., "Review on Heat Transfer Augmentation Techniques: Application in Gas Turbine Blade Cooling", Journal of Engineering Science and Technology Review, Vol. 5 (1), pp. 57-62, 2012.

[6] Han, J.-C., "Heat Transfer and Friction Characteristics in Rectangular Channels with Rib Turbulators," ASME J. of Heat Transfer, Vol .110, pp. 321-328, 1988.

[7] Han, J.C., Dutta, S., and Ekkad, S.V., Gas Turbine Heat Transfer and Cooling Technology, Taylor and Francis, New York, 2000.

[8] Han, J.C., Ligrani, P., and Chen, H.C., "Rotating and Stationary Rectangular Cooling Passage Heat Transfer and Friction with Ribs, Pins and Dimples", Annual Progress Report for: AGTSR Project No. SR-082, June 2001.

[9] Lee, M.S., Jeong, S.S., Ahn, S.W., and Han, J.C., "Effects of Angled Ribs on Turbulent Heat Transfer and Friction Factors in a Rectangular Divergent Channel", International Journal of Thermal Sciences, Vol. 84, pp. 1-8, 2014.

[10] Leung, C.W., Wong, T., and Kang, H.J., "Forced Convection of Turbulent Flow in Triangular Ducts with Different Angles and Surface Roughnesses", Heat and Mass Transfer, Vol. 34, pp. 63-68, 1998.

[11] Ligrani, P., "Heat Transfer Augmentation Techniques for Internal cooling of Turbine Components of Gas Turbine Engine", International Journal of Rotating Machinery, Vol. 2013, ID275653.

[12] Liu, Y.H., Wright, L.M., Fu, W.L., and Han, J.C., "Rib Spacing Effect on Heat Transfer and Pressure Loss in a Rotating Two-Pass Rectangular Channel (AR=1:2) with 45-Degree Angled Ribs," ASME Paper No. GT2006-90368, 2006.

[13] Liu,Y.-H."Heat Transfer in Leading and Trailing Edge Cooling Channels of the Gas Turbine Blade under High Rotation Numbers", Ph.D. Thesis, Texas A&M University, December 2008.

[14] Sundberg, J., "Heat Transfer Correlations for Gas Turbine Cooling", M.Sc. Thesis, Department of Mechanical Engineering, Linköping University, Linköping, Sweden, 2006.