Numerical And Experimental Investigation Of Heat Transfer Of Triangular Ribs In Divergent Duct

Ms. Renu Yeotikar, Prof. S. Y. Bhosale, Prof. H. N. Deshpande

Abstract: In this work, numerical and experimental investigation has been carried out to obtain detailed heat transfer distributions for a diverging channel with and without enhancement features. The cooling configurations considered include rib turbulators of triangular cross section on the main heat transfer surface. All of the measurements are presented at a range of Reynolds number from 6560 to 13100. Pressure drop measurements for the overall channel are also presented to evaluate the heat transfer enhancement geometry with respect to the pumping power requirements. Various geometrical parameters such as angle of attacks of ribs (90°, 45° & 30°), rib spacing i.e.(P/e=6,8,10,12), rib heights (e=3, 4,5,6mm), and comparison of triangular shaped rib with square shaped rib for the same boundary conditions have been studied numerically for the different heater inputs. This numerical analysis of the results shows that the triangular ribs oriented at 45° angle of attack, with P/e=8 and with rib height of 3mm provides significantly higher heat transfer performance than plain plate. It is concluded that there is increase in thermal performance of 20% to 30% by the use of triangular ribs.

Keywords: Divergent duct, Triangular ribs, Blockage ratio, thermal performance enhancement

I. INTRODUCTION

Various attempts have been made during the recent years in order to apply different technological methods to improve heat transfer in different thermal systems. Moreover saving a substantial amount of energy, achieving greater rates of heat transfer can lead to the development of more compact heat transfer systems with higher thermal efficiencies. Among various methods studied in the past research, creating secondary flows, flow mixing, flow rotation, and causing disturbance in the boundary layer have been especially taken into account for increasing heat transfer rate in channels. Further mixing in the flow and, thus, more uniform profiles of temperature and velocity has led to greater heat transfer rate between the flow and the channel walls. Use of these methods can be proficient to achieve higher rates of heat transfer. The researchers have exploited various techniques thus far such as different types of fins, baffles, protruding surfaces and most commonly and famous method of use of ribs to create mixing in the flow and improve heat transfer.

Several investigations have been carried out to study the effect of these parameters of ribs on heat transfer and friction factor for roughened surfaces. Justin Lamont et al. [1] worked on heat transfer enhancement in narrow diverging channels with obstacles and obtained detailed heat transfer coefficient distributions and pressure drop measurements at a Reynolds number of 28,000. His analysis showed that the ribbed ducts provided significantly higher heat transfer coefficients and a higher overall pressure drop than the plain duct. K. R. Chavan et al. [2] experimentally investigated heat transfer enhancement in ribbed divergent and convergent ducts with different arrangements of ribs placed on inner surface of duct for Reynolds’s numbers ranging 5000-25000. They found that the thermal performance of divergent duct with ribs was higher than the plain divergent duct and it increased by 34% due to Rib tabulator. M. S. Lee et al.[3] have been experimentally investigated the local heat transfer and pressure drop of developed turbulent flows in the stationary ribbed rectangular convergent/divergent channels with a fixed rib height e=10 mm and the ratio of rib spacing (p) to height P/e=10 at Reynolds numbers from 15,000 to 89,000. The comparison showed that among the four channels of C1 (Dho/Dhi=0.67), C2 (Dho/Dhi=0.86), D1(Dho/Dhi=1.16), and
D2(Dho/Dhi=1.49), the divergent channel of D2 was the highest thermal performance at the identical mass flow rate. Dr. Natarajan et al. [4] analysed the rib height comparison for three different sized square ribbed divergent rectangular ducts (e/D=0.035, 0.0697 and 0.1046) and compared the results with smooth duct and concluded that the enhanced heat transfer rate for the 3 mm height rib divergent rectangular duct was found to be more than 6, 9 mm rib height rectangular divergent duct and smooth duct respectively. Oronzio Manca et al. [5] carried out a numerical investigation on turbulent water forced convection in a ribbed channel with differently shaped ribs such as square, rectangular, trapezoidal and triangular to analyse the flow and the heat transfer characteristics of the ribbed channel by means of Fluent code v6.3 for Reynolds numbers in the range between 20000 and 60000 and concluded that the triangular ribs with w/e= 2.0 provides significant thermal performances. Ahmed M. Babagir et al. [6] investigated numerically to examine turbulent flow and heat transfer characteristics in a three-dimensional ribbed square channels for the Reynolds numbers ranging from 10^4 to 4 x10^4. Rib arrays of 45° inclined, 90 ° transverse ribs and 45 ° V-shaped are mounted in inline and staggered arrangements on the lower and upper walls of the channel. Analysis found that 45° V-shaped performed better than other shapes. Some researchers have focussed on the investigation of heat transfer of ribs in tapered or divergent channels. Tuqa Abdulrazzaq et al. [7] dealt with turbulent heat transfer to fluid flow through channel with triangular ribs of different angles using Ansys 14 ICEM and Ansys 14 Fluent at Reynolds number varied from 20000 to 60000. Analysis observed that triangular ribs of angle 60° which provides the highest enhancement of heat transfer compared to the ribs of angel 90° and 45° and at same Reynolds number. Monsak Pimsarn et al.[8] conducted experiment on a rectangular duct with Z-shaped ribs placed at an angle of 30°, 45°, 60° and 90° relative to the air flow direction for the Reynolds numbers ranging from 5000 to 25,000 in the test section and found that the 45° Z-rib provided the highest increase in the heat transfer rate and the best thermal performance. Ponjet Promvonge, et al. [9] conducted the experiment on a constant heat flux channel fitted with different shaped ribs such as triangular (isosceles), wedge (right-triangular) and rectangular shapes to assess turbulent forced convection heat transfer and friction loss behaviours for air flow and found that Isosceles triangular shaped rib had given best heat transfer performance. C. Thianpong et al. [10] dealt with experimental investigation for a constant heat-fluxed straight channel fitted with different heights of (4, 6 7 8mm) of triangular ribs to evaluate turbulent heat transfer and friction loss behaviours of airflow at Reynolds number of 5000 to 22,000 and concluded that the lowest e/Dh rib with staggered array provided the best thermal performance.

II. NUMERICAL STUDY
A computational fluid-dynamic analysis of a three-dimensional channel model, provided with triangular ribs, as reported in Figure 1, is considered in order to evaluate its thermal and fluid-dynamic behaviours and study the temperature and velocity fields. A constant uniform heat flux is applied on the external channel walls. Different inlet velocities are considered in the ranges of turbulent regime and the working fluid is air with constant properties.

The working fluid is air and the assigned boundary conditions are the following:

- **Inlet:** Velocity inlet boundary condition [velocity, static temperature].
- **Outlet:** Pressure Outlet [atmospheric pressure].
- **Heat source:** constant heat flux
- **Walls:** constant Adiabatic.

III. GEOMETRICAL CONFIGURATION
The three-dimensional model, depicted in Figure 1, represents a divergent channel with a length, L, equal to 500 mm while its height H is set equal to 8 mm at inlet & 23 mm at outlet, width W is 13 mm at inlet and 27m at outlet; the hydraulic diameter is equal to 18.37 mm. The 5mm thick baseplate is made up by aluminium and a constant heat flux equal to 2000 W/m² has been applied on the external surfaces. Triangular ribs are mounted on the internal principal wall as shown in fig. 1 and
they are characterized by different geometry parameters as height (e), pitch (P), width (w) and shape of turbulators. In this paper, triangular ribs of aspect ratio of \((w/e= 2)\), \((e=3\text{mm})\) are considered. The range of dimensionless roughness parameters and Reynolds numbers employed in this investigation are given below:

- Reynolds number, \(\text{Re}\) (from 6560 to 13100)
- Angle of attack of ribs \((\alpha = 30, 45 & 90)\)
- Dimensionless roughness pitch, \(P/e (6, 8, 10 & 12)\)
- Dimensionless height, \(e/Dh (0.17, 0.22 & 0.26)\)

IV. NUMERICAL MODEL

The governing equations are solved by means of the finite volume method. The working fluid is air and its properties are considered constant. A steady-state solution and a segregated method are chosen to solve the governing equations, which are linearized implicitly with respect to dependent variables of the equation. A second-order upwind scheme is chosen for energy and momentum equations. The SIMPLE coupling scheme is chosen to couple pressure and velocity. The convergence criterion of \(10^{-5}\) is assumed for the residuals of the velocity components and energy, respectively. It is assumed that the incoming flow is turbulent at ambient temperature, \(T_a\) (300 K), and pressure. Different inlet uniform velocities, \(V_{in}\), equal to 6m/s, 9m/s and 12m/s ,corresponding to Reynolds numbers ranging from 6560 to 13100, were considered. Furthermore, the inlet turbulence intensity value is set to 1%. Along the solid walls no slip condition is employed whereas a velocity inlet and pressure outflow conditions are given for the inlet and outlet surfaces.

Results are validated by comparing the obtained numerical data with the experimental result data in terms of average Nusselt numbers and friction coefficients, in the case of plain divergent duct and ribbed divergent duct.

V. EXPERIMENTAL SETUP

An experimental apparatus has been designed to measure the temperature distribution and pressure drop by the ribs in divergent duct. The duct is divergent in shape made up of Acrylic material with triangular ribs placed on bottom wall. The bottom wall of the duct is heated by a plate type heater place beneath the bottom base plate. The other three walls of the duct are non-heating as made up of Acrylic. Bottom wall with plate heater are insulated by using 5mm thick Asbestos sheet to avoid heat losses. Temperature of the bottom test plate is measured by K- type thermocouple. Air as a working fluid is made to flow by the use of a blower in the duct.

In the present study, following assumptions are made as 1. The thermal conductivity of material is uniform and constant. 2. Heat loss from radiation is neglected. 3. There is a complete contact between the plate surface and the rib. 4. Air velocity from the blower is considered as uniform. 5. Loss through the fine gaps of acrylic duct is neglected.

A schematic diagram of the experimental apparatus is presented in Fig.2 while the details of ribs mounted on an aluminium plate in divergent channel are depicted in Fig. 3. In Fig. 2, a circular pipe was used for connecting a 0.5Hp high-pressure blower to a settling tank, while the channel including the calm section and the test section was employed following the settling tank. The channel geometry is characterized by the channel height, \(H\) with the channel width, \(W_{in}\) of 13 mm at inlet and \(W_{out}\) of 27mm at outlet with total length of test section as 500 mm. The ribbed wall was fabricated from 5 mm thick aluminium plate, 13 mm wide at inlet and 27mm at outlet and 500 mm long (L). The uniform rib dimensions are 3mm high (e), 6mm wide and 15 mm thick (t). The form of ribbed plate was accomplished by means of wire-EDM (electrical discharge machine) machining. The channel test section is composed of the three walls in C section made up of acrylic and one bottom principal wall made up of aluminium. The AC power supply was the source of power for the plate-type heater,
used for heating of the test plate of test section to maintain uniform surface heat flux. A conducting compound was applied to the heater and the principal wall in order to reduce contact resistance. Test plate is covered with asbestos sheet to act as insulating wall. Air as the tested fluid in both the heat transfer and pressure drop experiments, was directed into the systems by a 0.5Hp high-pressure blower. The flow rate of air in the systems was measured by vane-type anemometer (Testo 445). The pressure across the channel was measured using pressure transducer. In order to measure temperature distributions on the principal wall, five thermocouples were fitted to the upper wall only due to assumption of symmetric walls.

VI. DATA REDUCTION

The goal of this experiment is to investigate the Nusselt number in triangular-ribbed channel. The independent parameters are Reynolds number, rib height ratios and e/Dh ratios. The Reynolds number based on the channel hydraulic diameter is given by

\[ Re = \frac{(\rho V D_h)}{\mu} \]  

(1)

The heat transfer coefficients are evaluated from the measured temperatures and given heat inputs. With heat added uniformly to fluid (Qair) and the temperature difference of wall and fluid (Tw−Tb), average heat transfer coefficient will be evaluated from the experimental data via the following equations:

\[ h = \frac{Q_{in}}{A_s(T_s-T_{mf})} \]  

(2)

in which,

\[ T_{mf} = \frac{(T_0 + T_i)}{2} \quad \text{and} \quad T_s = \frac{T_1 + T_2 + T_3 + T_4 + T_5}{5} \]  

(3) & (4)

The term As is the convective heat transfer area of the upper channel walls whereas Ts is the average surface temperature obtained from local surface temperatures along the axial length of the heated Plate. Then, average Nusselt number is written as:

\[ Nu = \frac{h D_h}{K} \]  

(5)

The friction factor is evaluated by:

\[ f = \frac{2 \Delta P}{(\frac{1}{5}) \rho V^5} \]  

where \( \Delta P \) is a pressure drop across the test section and \( V \) is mean air velocity of the channel. All of thermo-physical properties of the air are determined at the mean bulk air temperature from Eq.(3)

The thermal enhancement factor, \( \eta \), defined as the ratio of the heat transfer coefficient of an augmented surface to that of a smooth surface,

\[ \eta = \left( \frac{Nu}{Nu_0} \right) \ast \left( \frac{f}{f_0} \right)^{-1/(3)} \]

VII. RESULTS AND DISCUSSIONS

Results are presented in form of graphs depicting the average Nusselt number, friction factors and required pumping power at different dimensionless pitches, heights and ribs shapes, and Reynolds numbers, and also temperature fields and streamlines contours for some significant cases are given. Ribs are characterized by triangular shape with different values of P/e ratio, e/Dh ratio and different angle of attacks.

The present work on both heat transfer and pressure loss behaviours in an isosceles triangular ribbed channel with aspect ratio (AR) of rib of 2 and fixed rib height as 3mm is presented. Measurements were conducted for rib array over a range of Reynolds numbers as mentioned earlier
a. **Effect of rib height:**

The experimental results on heat and flow friction characteristics in a uniform heat flux channel fitted with triangular ribs of four uniform heights, e = 3, 4, 5, and 6 mm (e/Dh=0.17, 0.22, 0.26 and 0.35) are presented in the form of Nusselt number and friction factor in the figures. The Nusselt numbers obtained under turbulent flow regimes for all rib turbulators with a single rib pitch, P= 24 mm (P/e=8) are depicted in Fig. In the figure, the use rib turbulators leads to considerable heat transfer enhancements in a similar trend in comparison with the smooth channel. It is found that the Nusselt number increases with the rise of rib height, apart from Reynolds number values. This is because the presence of ribs interrupts the development of the boundary layer of the fluid flow and helps to increase the turbulence intensity of the flow. A close inspection reveals that the Nusselt number of rib with e/Dh=0.23 and 0.28 are seen to be approximately the mean value between the Nusselt numbers of ribs with e/Dh=0.34 and 0.17 for the same Reynolds number. The use of rib with e/Dh=0.17 shows a higher heat transfer rate than that with e/Dh=0.34 at around 65% or with e/Dh=0.28 at about 35%. The variation of the pressure drop is presented in terms of friction factor with Reynolds number as shown in fig.3. In this figure, it is apparent that the use of rib turbulators leads to a substantial increase in friction factor over the smooth channel. The friction factor for rib turbulators is considerably higher than that for the smooth channel. This can be attributed to flow blockage, higher surface area and the act caused by the reverse flow due to the presence of ribs. As expected, the friction factor obtained from the rib with e/Dh=0.34 is substantially higher than that from one with smaller e/Dh values. The mean increase in friction factor of using the ribs is in a range of 20 to 95% the smooth channel. The friction factor value of the rib with e/Dh=0.34 is found to be around 28% higher than that with e/Dh=0.28 or about 70% higher than one with e/Dh=0.17. The losses mainly come from the dissipation of the dynamical pressure of the air due to high viscous losses near the wall, to the extra forces exerted by reverse flow and to the high flow blockage due to the presence of the ribs, especially for rib with e/Dh=0.34.

b. **Effect of angle of attack:**

It is found that if ribs are placed at an inclination angle with respect to the axial direction, secondary flows are generated over the channel which results in the rise in the heat transfer rate towards the upstream region with respect to the downstream one. The effect of attack angle of triangular ribs on Nusselt number at various Reynolds numbers is shown in Fig. In general, Nusselt number increases with increasing Reynolds number owing to higher turbulent intensity imparted to the flow between the ribs. The effect of attack angle of triangular ribs becomes more significant as Reynolds number increases due to the induction of recirculation or rotating eddy. The ribs placed at 45° which induce recirculation give higher Nusselt number than the ones which do not (30° and 90°). This is attributed to a better mixing of the fluid between the core and the tube surface regions caused by turbulent fluctuation or eddy motion between rib elements. This section shows the comparison among the results of triangular ribs with different angle of attack α = 30°, 45° and 90° with the same rib height and pitch ratio, (e/Dh= 0.17, and P/e= 8) and smooth channel. Fig. shows that the triangular ribs placed at an angle of attack of 45° are higher heat transfer enhancement factor than that the ribs with 90° and 30°. The heat transfer enhancement factor obtained from the triangular ribs with α = 45° are, around 30% and 50% more compared to angle of attack of 90° and 30° as can be seen in Fig.

c. **Effect of rib spacing:**

As shown in Fig. eleven ribs were periodically positioned on the heated side of the test section. Four values of the longitudinal rib pitch by rib height e = 3 mm were selected: P = 18, 24, 30 and 36 mm to yield rib pitch-to-rib height ratios P/e = 6, 8, 10, and 12 respectively. The Reynolds number, based on the channel hydraulic diameter, was: 6560, 9850 and 13100. The effect of rib spacing on thermal enhancement factor versus various Reynolds numbers is shown in Fig. The heat transfer
enhancement factor obtained from the triangular ribs with rib spacing of 24mm and pitch/height ratio of 8 is found to be more than that of the other cases of rib spacing are, around 30% and 50% more compared to $P/e = 6, 10 \& 12$ as can be seen in Fig. The optimum value of $P/e$ is found approximately to be 8 for the divergent channel with triangular ribs on bottom side.

d. Effect of rib geometry:
The present experimental results on heat and flow friction characteristics in a uniform heat flux channel equipped with 3 mm rib height of two different cross-sections: triangular and square are presented in the form of thermal enhancement factor and friction factor. The thermal enhancement factors obtained under turbulent flow conditions for both rib-type turbulators with only one rib pitch ($P= 24 \text{ mm}$) are presented in Fig. In the figure, the rib turbulators yield considerable heat transfer enhancements with a similar trend in comparison with the smooth channel and the Nusselt number increases with the rise of Reynolds number. Even though the squarer rib provides the highest value of Nusselt number, while the triangular rib is found to be perform better than square ribs. A close examination reveals that both the square and triangular ribs yield higher heat transfer than the smooth channel for all Reynolds number values, but there is very high increase in pressure drop in case of square ribs, hence square ribs gives lower heat transfer performance as compared to triangular ribs. For the triangular rib, the increase in thermal enhancement factor value is about 15% more over the square rib. The effect of using the rib turbulators on the isothermal pressure drop across the tested channel is presented in Fig. The variation of the pressure drop is shown in terms of friction factor with Reynolds number. In the figure, it is apparent that the use of rib turbulators leads to a substantial increase in friction factor over the smooth channel. The increase in friction factor for rib turbulators is considerably higher than that for the smooth channel. This can be attributed to flow blockage, higher surface area and the act caused by the reverse flow. As the friction factor obtained from the square rib is substantially higher than that from the triangular one. The increase in friction factor of using the square rib is in a range of 19% to 140% times over the triangular ribs. The losses mainly come from the dissipation of the dynamical pressure of the air due to high viscous losses near the wall, to the extra forces exerted by reverse flow and to higher friction of increasing surface area and the blockage because of the presence of the ribs.

e. Performance Evaluation:
This section shows variation of the thermal enhancement factor with the Reynolds number in Fig. 10. In the figure, the thermal enhancement factors of divergent duct with triangular ribs are higher than the plain divergent duct due to the low heat transfer rate from the plate with no ribs. The triangular rib with $45^\circ$ angle of attack, $P/e= 8$ and $e/Dh = 0.17$ shows the highest thermal enhancement factor. This is because the Nusselt number at $45^\circ$ is very high when compare with the other angle ribs shown in Figures 6 and 8. The thermal enhancement factor tends to increase with the rise of Reynolds number values. The triangular rib shows the highest thermal enhancement factor of about 1.34 at $Re = 13100$. This confirms that to obtain higher the thermal performance, triangular ribs should be applied at $e/Dh$ of about 0.1, rib pitch ($P/e$) of about 8 and angle of attack of $45^\circ$. Fig. 20 shows the comparison of current result with various literatures studied in this work in terms of thermal enhancement factor and it is found to be little better values than the other works.

VIII. CONCLUSIONS
The work reported here is a systematic numerical as well as experimental study of heat transfer and friction factor of a divergent rectangular duct with in three different angle of attack of triangular rib(45, 30 and 90), four different height ribs (3, 4, 5 and 6 mm) and spacing of ribs as (24, 30 and 36mm). The Reynolds number variation range was 6500 to 13100. The heat transfer enhancement
Comparisons were made smooth, and ribbed divergent rectangular duct under the same mass flow rate.

a. Optimum geometry is found to be Triangular rib with height of 3mm, 45° angle of attack and with P/e = 8. Nusselt number augmentation tends to be increased with the rise of Reynolds number.

b. Heat transfer enhancement observed to be 17% to 23% and 20% to 26% more experimentally and numerically for triangular ribs as compared to smooth channel.

c. The triangular ribs with higher rib height, higher rib spacing and transverse ribs provide lower heat transfer enhancement compared to corresponding rib height of 3mm, rib spacing of 24mm and 45° angle of attack.

d. Thermal Enhancement Factor (TEF) for triangular ribs is found to be comparatively more with maximum value of 1.22 and 1.34 as per experimental and numerical study.

e. The triangular rib with e/Dh=0.17 and P/e = 8 with 45° angle of attack should be applied to obtain higher thermal performance, leading to more compact heat exchanger.

f. The best operating regime for all rib turbulators is found at higher Reynolds number values

REFERENCES

I. Justin Lamont, Sridharan Ramesh, Srinath V. Ekkad, “Heat Transfer Enhancement in Narrow Diverging Channels, Journal of Turbomachinery JULY 2013, Vol. 135 / 041017-7

II. R. Chavan, N.D. Dhawale, “Experimental Investigation of Heat Transfer Analysis of Ribbed Duct for Thermal Performance Enhancement”, International Journal of Science, Engineering and Technology Research (IJSETR) Volume 4, Issue 4, April 2015

III. M. S. Lee, S. S. Jeong, S. W. Ahn, “Heat Transfer and Friction in Rectangular Convergent and Divergent Channels with Ribs”, JOURNAL OF THERMOPHYSICS AND HEAT TRANSFER, Vol. 27, No. 4, October–December 2013.

IV. K. Sivakumar, Dr. E. Natarajan, Dr. N. Kulasekharan, “Heat transfer and pressure drop comparison between smooth and different sized rib-roughened rectangular divergent ducts”, International Journal of Engineering and Technology (IJET), ISSN : 0975-4024, Vol 6 No 1 Feb-Mar 2014

V. Oronzio Manca, Sergio Nardini, and Daniele Ricci, “Numerical Analysis of Water Forced Convection in channels with Differently Shaped Transverse Ribs”, Hindawi Publishing Corporation Journal of Applied Mathematics Volume 2011, Article ID 323485.25 pages doi:10.1155/2011/323485.

VI. Ahmed M. Bagabir, Jabril A. Khamajand Ahmed S. Hassan, “Turbulent Periodic Flow and Heat Transfer in a Square Channel with Different Ribs”, CFD Letters, www.cfdl.issres.net, Vol. 5(3) 2013

VII. Tuqa Abdulrazzaq, Hussein Togun, “Numerical Simulation on Heat Transfer Enhancement in Channel by Triangular Ribs”, International Journal of Mechanical, Aerospace, Industrial, Mechatronic and Manufacturing Engineering Vol: 7, No:8, 2013

VIII. Monsak Pimsarn, Parkpoom Sriromreun, and Pongjet Promvonge, “Augmented Heat Transfer in Rectangular Duct with Angled Z-Shaped Ribs”, International Conference on Energy and Sustainable Development: Issues and Strategies (2010)

IX. Pongjet Promvonge, Chinaruk Thianpong, “Thermal performance assessment of turbulent channel flows over different shaped ribs”, International Communications in Heat and Mass Transfer pp. (2008) 1327–1334

X. C. Thianpong , T. Chompookham, “Thermal characterization of turbulent flow in a channel with isosceles triangular ribs”, International Communications in Heat and Mass Transfer (2009), pp. 712–717

XI. Yunus Cengel, John Cimbala, “Fluid Mechanics”, Second edition, 2010
Fig. 1. Geometrical configuration

Fig. 2. Schematic diagram of Experimental Setup
**Fig.3** Actual Setup image

**Fig.4** Nusselt number comparison for different Rib height (Nu Vs Re)
Fig. 5. Friction factor comparison for rib heights (e=3 m to 6 mm)

Fig 6: Nusselt Number comparison for different angle of attack
Fig 7: Thermal enhancement factor comparison for different angle of attack

Fig 8: Thermal enhancement factor comparison for different P/e ratio
**Fig. 9: Nusselt number comparison for different Rib shapes**

![Nusselt Number Comparison Graph]

**Fig. 10: Friction factor comparison for different rib shapes**

![Friction Factor Comparison Graph]
Fig. 11 Nusselt number comparison for Experimental & Numerical analysis

Fig. 12. Friction factor comparison for Experimental & Numerical analysis
Fig. 13. Thermal enhancement factor comparison for Experimental & Numerical analysis

Fig 15. Vector velocity
Fig. 16 temperature contour for Plain plate

Fig. 17 Temperature Contour for Plate with triangular ribs
Fig. 18 Pressure Contour for Plate without ribs.

Fig. 19 Pressure Contour for Plate with ribs
Fig. 20 Comparison of Thermal Enhancement factor with various literatures

Comparison of Thermal Enhancement Factor

Authors
- Straight Channel-Prongjet
- Straight Channel-Prongjet
- Straight channel-Oronzio Manca
- Divergent channel-R. Yeotikar

TEF