CHAPTER 1
INTRODUCTION

1.1 Introduction:

Heat Exchanger is a device in which the exchange of energy takes place between two fluids or between fluid and wall of heat exchanger maintained at different temperatures. A heat exchanger utilizes the fact that, where ever there is a temperature difference, flow of energy occurs. The flowing fluids provide the necessary temperature difference and thus force the energy to flow between them. The energy flowing in a heat exchanger may be either sensible energy or latent heat of flowing fluids. The fluid which gives its energy is known as hot fluid, while the fluid which receives energy is known as cold fluid. It is but obvious that, temperature of hot fluid decreases while the temperature of cold fluid increases in heat exchanger. Accordingly, the purpose of heat exchanger is either to heat or cool the desired fluid. In a special case, when one of the fluids undergoes change in its phase, its temperature remains unchanged. These types of heat exchanger are known as condensers or evaporators.

Heat exchangers have been widely employed in several industrial and engineering applications. Heat exchangers are used in different processes ranging from conversion, utilization and recovery of thermal energy in various industrial, commercial and domestic applications. Some common examples include steam generation and condensation in power and cogeneration plants; sensible heating and cooling in thermal processing of chemical, pharmaceutical and agricultural products; fluid heating in manufacturing and waste heat recovery etc.

1.2 Need of Heat Transfer Enhancement:

Increase in Heat exchanger’s performance can lead to more economical design of heat exchanger which can help to make energy, material and cost savings related to a heat exchange process.The design of heat exchangers needs exact analysis of heat transfer rate and pressure drop estimations. The major challenge in designing a heat exchanger is to make the equipment compact and achieve a high heat transfer rate using minimum pumping power. In recent years, the high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment. The heat transfer rate can be improved by introducing a disturbance in the fluid flow, but in the process pumping power may increase significantly and ultimately the pumping cost becomes high.

The need to increase the thermal performance of heat exchangers, thereby effecting energy, material and cost savings have led to development and use of many techniques termed as-Heat transfer Augmentation. These techniques are also referred as-Heat transfer Enhancement or Intensification. Augmentation techniques increase convective heat transfer by reducing the thermal resistance in a heat exchanger.

In general, some kind of inserts is placed in the flow passage to augment the heat transfer rate, and this reduces the hydraulic diameter of the flow passage. Heat transfer enhancement in a heat exchanger by inserts such as twisted tapes, wire coils, ribs and dimples is mainly due to flow blockage, partitioning of the flow and secondary flow. Flow blockage increases the pressure drop and leads to increased viscous effects because of a reduced free flow area. Blockage also increases the flow velocity and in some situations leads to a significant secondary flow. Secondary flow further provides a better thermal contact between the surface and the fluid because secondary flow creates swirl and the resulting mixing of fluid improves the temperature gradient, which ultimately leads to a high heat transfer coefficient.

CHAPTER 2
LITERATURE REVIEW

2.1 Introduction:

The effects of the various inserts, rib arrangement and configuration on the heat transfer performance have been investigated by many researchers. Recent work that has been carried out related to study of convective heat transfer from rectangular and square duct provided with different types of inserts is reviewed as follows:-

Mr. S. J. Thikane [1] Heat transfer enhancement technology is the process of improving the performance of a heat transfer by increasing the convection heat transfer coefficient. Generally the main objective is to reduce the size and costs of heat exchangers. General techniques for enhancing heat transfer can be divided in two categories. One is passive method such as twisted tapes, helical screw tape inserts, rough surfaces, extended surfaces, additives for liquid and gases. The other is active method, which requires extra external power, for example mechanical aids, surface fluid vibration, use of electrostatic fields. Passive methods are found more inexpensive as compared to active methods**.**

Aharwal, K.R [2] Artificial roughness in the form of repeated ribs has been proposed as a convenient method for enhancement of thermal performance of solar air heaters. This paper presents the experimental investigation of heat transfer and friction factor characteristics of a rectangular duct roughened with repeated square cross-section split-rib with a gap, on one broad wall arranged at an inclination with respect to the flow direction. The duct has a width to height ratio (W/H) of 5.84, relative roughness pitch (P/e) of 10, relative roughness height (e/Dh) of 0.0377, and angle of attack (α) of 60°. The gap width (g/e) and gap position (d/W) were varied in the range of 0.5–2 and 0.1667–0.667, respectively. The heat transfer and friction characteristics of this roughened duct have been compared with those of the smooth duct under similar flow condition. The effect of gap position and gap width has been investigated for the range of flow Reynolds numbers from 3000 to 18,000. The maximum enhancement in Nusselt number and friction factor is observed to be 2.59 and 2.87 times of that of the smooth duct, respectively. The thermo-hydraulic performance parameter is found to be the maximum for the relative gap width of 1.0 and the relative gap position of 0.25**.**

Ravi Teja [3] This study comprehensively simulates the use of laminar and k-ε model for predicting flow and heat transfer with measured flow field data in a stationary duct which sheds light on the detailed physics encountered in the fully developed flow region, and the sharp 180° bend region. Among the major flow features predicted with accuracy are flow transition at the entrance of the duct, the distribution of mean and turbulent quantities in the developing, fully developed, and sharp 180° bend, the development of secondary flows in the duct cross-section and the sharp 180° bend, and heat transfer augmentation. Flow intensities in the sharp 180° bend are found to reach high values and local heat transfer comparisons show that the heat transfer augmentation shifts towards the wall and along the duct. Therefore, understanding of the unsteady heat transfer in sharp 180° bends is important

M.Udaya Kumar[4] This paper presents an experimental study of heat transfer and friction factor of plain square duct with inserts under turbulent flow condition constant heat flux. To conduct the experiments in plain square ducts, with and without inserts air is considered as the working fluid. In order to estimate the heat transfer coefficient and friction factor an experimental set up is fabricated. Experiments are first conducted in plain straight square duct with and without inserts and compared the data with existing literature values. The heat transfer characteristics are predicted under axially constant wall heat flux condition As such, the flow and heat transfer are periodically fully developed in axial direction turbulent heat convection in a square duct is one of the fundamental problem in a thermal science and Engineering. The enhancement of heat transfer in a duct is often achieved by forming some swirling or secondary flow is usually accompanied with high turbulent intensity, which promotes the mixing of different parts of fluids, hence enhances the heat transfer.

Mustafa J. AL-DULAIMI [5] This paper investigates numerically the influence of detached square vortex generator (VGs) on the heat transfer and pressure drop inside a square duct. Reynolds number is fixed at 5000. The geometrical parameters in this investigation are: i) The blocking ratios are 0.1, 0.15 and 0.2), ii) Vortex generator numbers are 1, 2, and 3), iii) Attack angles are 0, 30, and 45, iv) The aspect ratios are 1, 1.5 and 2. The numerical simulation is carried out using ANSYS FLUENT 15 . The results show that the rectangular vortex generators have a positive influence on heat transfer as a result of the augmentation in turbulence level. The maximum enhancement in average heat transfer could reach 40%. The heat transfer is found to increase with the blocking ratio. The heat transfer enhanced by 17% for one VG and 28% for 3 VGs for blocking ratio = 0.2. The VGs at angle value of 45° produce the highest heat transfer enhancement. The aspect ratio is found to have an adverse effect on heat transfer rate.

Sagar S. Desai[6] Several cooling technique have been developed to enhance heat transfer in square duct. Different rib arrays inside square channel are widely used to enhance heat transfer rate. The reason that ribs increase the fluid flow turbulence near the wall, disrupt the boundary layer and also increase the heat transfer area. In this paper, numerical analysis is carried out three different angles of turbulators were placed in square duct. All turbulators located on bottom side wall of duct. The numerical simulation are carried on square duct having hydraulic diameter (Dh) of 0.05m. Air is working fluid with the flow rate in terms of Reynolds number ranging from 15,000 to 20,000. Details for rib height (e), pitch distance between turbulators (P) and turbulators angle are similar to experimental reference. The model is creating using Ansys ICEM software. Numerical simulations were performed using the CFD software package ANSYS 14.5 FLUENT. Turbulence closure was achieved using k-ε turbulence model, with enhance wall treatment for the simulation were used. In this, the heat transfer characteristics of square duct with internal w-shaped ribs with different angles and pitch ratio 0.3 were plotted.

T.S. Dhanasekaran [7] Increasing the turbine inlet temperature can increase the gas turbine cycle efficiency. In order to increase the turbine inlet temperature significantly, an advanced cooling system has to be essentially developed. Injection of mist to the coolant fluid is considered a promising technique to protect the hot components such as combustor liners, combustor transition pieces, and turbine vanes and blades. A series of experiments conducted in the past proved the success of mist cooling technology in the laboratory environment. Favorable results from the numerical simulation further encourage continuous exploration of employing mist-cooling technology in the actual gas turbine working environment in various applications. The present study focuses on applying mist cooling to the rotating mist/air internal cooling passage with rib turbulators using numerical simulation.

Sagar et al. [8] carried out a numerical analysis of heat transfer for three different angles of w-shaped turbulators placed at the bottom side wall of square duct. From the numerical analysis of it is found that Nusselt number and friction factor in duct with W-rib insert increases as compare to smooth duct without insert.

Prakash Santosh Patil[9] Modern gas turbines operate at high inlet temperatures to improve thermal efficiency; therefore, it is necessary to cool the turbine blades. Various techniques are used for heat transfer enhancements, such as ribs, protrusions, pin fin, dimples, etc.; the present study focused on compound and rib alone channel. W-shaped, semicircular and multi-semicircular shaped ribs with dimples are studied experimentally to find the optimum configuration for blade cooling. The experiment was carried out at Reynolds numbers 12,600 to 35,000; the ratio of pitch (P) to height (e) of the rib was 8 to 10, the ratio of rib height to channel hydraulic diameter (Dh) was 0.156 and the ratio of dimple depth (δ) to dimple diameter was 0.2. It was observed that the combination of rib and dimple channel (compound channel) performance was higher than the rib channel. The W-shaped rib compound channel shows the highest thermal performance over semicircular and multi-semicircular rib compound channels, and also sees a small rise in friction loss in compound channel. Realizable k–ε turbulence model was used for analysis and observed less difference between experimental and CFD results. In the rib channel, semicircular rib performed better than other tested ribs.

Udaya et al. [9] performed an experimental study of heat transfer enhancement in square duct with inserts under turbulent flow condition and constant heat flux. Experiments are first conducted in plain straight square duct with and without inserts and compared the data with existing literature values. It was observed that enhancement of heat transfer in a duct is achieved by forming some swirling or secondary flow is usually accompanied with high turbulent intensity, which promotes the mixing of different parts of fluids, hence enhances the heat transfer. It is seen that for an increase Reynolds number up to 15%; Nusselt number increased by 30%. It is observed that experimental heat transfer coefficient of air increases by inserting the different inserts in square duct.

Pankaj et al. [10] carried out an experimental analysis to study turbulent flow heat transfer in rectangular duct with and without internal ribs. The effects of internal ribs on the heat transfer coefficient and friction factor are compared with the result of smooth duct under similar flow conditions. Experimental results show that the local Nusselt number distribution is strongly depended on the position, orientation, and geometry of the ribs. Also the discrete V- shaped ribs produce overall less heat transfer enhancement than the continuous V- shaped ribs. However, the increased heat transfer enhancement in the continuous V- shaped ribs came at the cost of an increased pressure penalty.

Yongsiri et al. [11] carried out a numerical study of turbulent flow and heat transfer in a channel with inclined detached-ribs. The effects of the inclined detached ribs with different attack angles on the heat transfer, friction factor and thermal performance behaviours have been investigated numerically for Reynolds numbers from 4000 to 24,000. Among the ribs examined, the one with Ɵ=60° yield comparable heat transfer rate 1.74 times of those in the smooth channel and Ɵ=120° yield thermal performance factor 1.21 which are higher than those given by the others.

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Priyank et al. [12] carried out a two-dimensional CFD investigation to study forced convection of fully developed turbulent flow in a rectangular duct having ribs on the underside of the top wall. The effect of Reynolds number and relative roughness pitch on the heat transfer coefficient and friction factor have been studied. It is observed that the roughened duct having circular and square rib with highest relative roughness height provides the highest Nusselt number at a higher value of Reynolds number. Square sectioned rib provides higher value of enhancement as compared to circular rib at a higher value of Reynolds number.

Heeyoon et al. [13] carried out an experimental investigation of the effect of an intersecting rib on heat/mass transfer performance in rectangular channels with angled ribs and different aspect ratios. In a rib-roughened channel with angled ribs, heat/mass transfer performance deteriorates as the channel aspect ratio increases, since the vortices induced by angled ribs diminish with increasing aspect ratio. A longitudinal rib that bisects the angled ribs is suggested to overcome this disadvantage. The heat transfer performance of angled rib configurations with a 60_ attack angle were tested with and without an intersecting rib using naphthalene sublimation method. The channel aspect ratio is varied from 1 to 4. When the intersecting rib was present, additional vortices were generated at every point of intersection with the angled ribs. Thus the heat/mass transfer performance was significantly enhanced for all channel aspect ratios when an intersecting rib was added to an ordinary angled rib configuration.

.2 Concluding Remarks:

From the previous researches following conclusions are drawn:

- Heat transfer in square ducts was found considerably higher than the circular tube. This is mainly because the square duct has high surface to volume ratio as compared to circular tube.
- Heat transfer rate is increased by using square ducts with inserts, because inserts increase the turbulence of the flow.
- The heat transfer characteristics are mainly dominated by turbulent transport and secondary flow induced by geometrical features.
- Ribs installed at an acute angle provide better heat transfer performance than orthogonally installed ribs. These angled ribs generate strong secondary vortices adjacent to the channel surface as the flow moves in the direction of the ribs. The vortices sweep the surface, and locally intensified heat transfer occurs in these regions. Moreover, parallel ribs installed on opposite channel surfaces generate a rotational flow, which promotes mixing of the fluid.

2.3 Problem Definition:

"Experimental Investigation of Forced Convection Heat Transfer in a Square Duct Provided with Different Configurations of V-shaped Rib Turbulators"

2.4 Objectives of the study:

The objectives of the present dissertation work are-

1. To investigate the effect of different configurations of rib turbulators on the heat transfer coefficient and friction factor characteristics
2. To determine the enhancement in convective heat transfer due to provision of rib turbulators inside the square duct
3. To compare the values of heat transfer coefficient of air obtained for square duct without rib turbulators and for square duct with rib turbulators
4. To determine the value of overall enhancement ratio for different configurations of the rib turbulators

CHAPTER 3
THEORY

3.1 Objectives of Heat Transfer Enhancement:

In most practical applications of enhancement techniques, the following performance objectives, along with a set of operating constraints and conditions, are usually considered for optimizing the use of a heat exchanger:

i. Increase the heat duty of an existing heat exchanger without altering the pumping power (or pressure drop) or flow rate requirements.
ii. Reduce the approach temperature difference between the two heat-exchanging fluid streams for a specified heat load and size of exchanger.
iii. Reduce the size or heat transfer surface area requirements for a specified heat duty and pressure drop or pumping power.
iv. Reduce the process stream's pumping power requirements for a given heat load and exchanger surface area.

It may be noted that objective 1 accounts for increase in heat transfer rate, objective 2 and 4 yield savings in operating (or energy) costs, and objective 3 leads to material savings and reduced capital costs.

3.2 Classification of Heat transfer enhancement or augmentation techniques:

Heat transfer enhancement or augmentation techniques refer to the improvement of thermo-hydraulic performance of heat exchangers. Existing enhancement techniques can be broadly classified into three different categories:

1. Passive Techniques
2. Active Techniques
3. Compound Techniques

3.2.1 Passive Techniques:

These techniques generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behavior (except for extended surfaces) which also leads to increase in the pressure drop. In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over the active techniques as they do not require any direct input of external power.

Heat transfer augmentation by these techniques can be achieved by using:

− **Treated Surfaces**:

It consists of a variety of structured surfaces (continuous or discontinuous integral surface roughness) and coatings. The roughness created by this treatment do not causes any significant effect in the single phase heat transfer. These are applicable in cases of two phase heat transfer only. This technique involves using pits, cavities or scratches like alteration in the surfaces of heat transfer area which may be continuous or discontinuous. They are primarily used for boiling and condensing duties.

− **Rough surfaces**:

Small scale roughness or surface modification promotes turbulence in the flow field near the wall region by disturbing the viscous laminar sub layer. This disturbance causes higher momentum and heart transfer. This small scale roughness has little effect in laminar flows, but is very effective in turbulent single phase flows. External rough surface can created by grooving the heat transfer surface and can be used in double pipe and shell and tube bundles to enhance annulus or shell side heat transfer. Corrugated tubes, a type of 2-D roughness is shown in Fig. 3.1, rough surfaces have been employed to enhance heat transfer in single phase flows both inside tubes and outside tubes.

Fig. 3.1 Corrugated tubes, Two-Dimensional Roughness

− **Extended surfaces**:

Extended or finned surfaces increase the heat transfer area which could be very effective in case of fluids with low heat transfer coefficients. This technique includes finned tube for shell and tube exchangers, plate fins for compact heat exchanger and finned heat sinks for electronic cooling. Finned surfaces enhance heat transfer which can be used for cooling of electrical and electronic devices. The use of extended surfaces for cooling electronic devices is not restricted to the natural convection heat transfer regime but also can be used for forced convective heat transfer. Segmented or interrupted longitudinal fins, as shown in Fig. 3.2, promote boundary layer separation of the fluids and disturb the whole bulk flow field inside circular tubes.

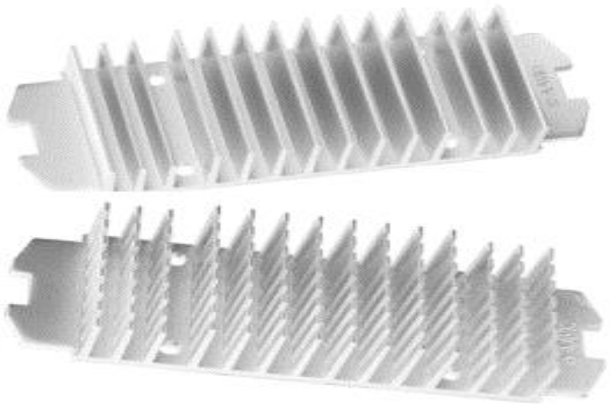
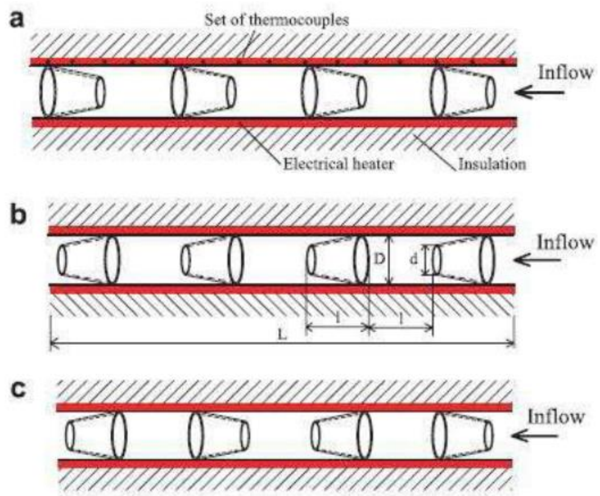


Fig. 3.2 Segmented fin heat sink

− **Displaced enhancement devices**:

Displaced enhancement devices displace the fluid elements from the core of the channel to heated or cooled surfaces and vice versa .Displaced enhancement devices include inserts like static mixer elements, metallic mesh, and discs, wire matrix inserts, rings or balls. Different types of conical ring inserts used in circular tubes are shown in Fig. 3.3. These inserts do not alter heat transfer surface and provide a lot of scope for inter-mixing of the fluid particles.



a- Diverging Ring   b- Converging Ring   c- Converging and Diverging Rings

Fig. 3.3 Conical Ring inserts in circular tubes

− **Swirl flow devices**:

Swirl flow devices causes swirl flow or secondary flow in the fluid .A variety of devices can be employed to cause this effect which includes tube inserts, altered tube flow arrangements, and duct geometry modifications. Dimples, ribs, helically twisted tapes are examples of duct geometry modifications. Tube inserts include twisted-tape inserts, helical strip or cored screw–type inserts and wire coils. Fig. 3.4 to Fig. 3.7 shows different configurations of twisted tape inserts which are used commonly.

Fig. 3.4 Plain Twisted Tape

Fig. 3.5 Baffled Twisted Tape with holes

Fig. 3.6 Wavy twisted tape insert

Fig. 3.7 Parabolic-cut twisted tape insert

Inserts such as twisted tape, wire coils, ribs and dimples mainly obstruct the flow and separate the primary flow from the secondary flows. This causes the enhancement of the heat transfer in the tube flow. Inserts reduce the effective flow area thereby increasing the flow velocity. This also leads to increase in the pressure drop and in some cases causes' significant secondary flow. Secondary flow creates swirl and the mixing of the fluid elements and hence enhances the temperature gradient, which ultimately leads to a high heat transfer coefficient.

− **Coiled tubes**:

In these devices secondary flows or vortices are generated due to curvature of the coils which promotes higher heat transfer coefficient in single phase flows and in most regions of boiling. This leads to relatively more compact heat exchangers. Refer Fig. 3.8.

Fig. 3.8 Coiled Tubes

Coiled tubes are used in domestic water heaters, chemical process reactors, solar heating system, industrial and marine boilers, kidney dialysis devices and blood oxygenators.

− **Additives for liquids**:

Pressure drop in the tube flow is a consequence of the frictional loses with the solid surface. These frictional loses occurs because of the drag force of the fluid. This technique is basically concerned with reducing the drag coefficient using some additives to the fluid in single phase flows. Additives when added to the fluids are found to have operational benefits by lowering the frictional losses. These operational benefits could be fixed pressure or pumping costs. Polymeric additives induce a viscoelastic character to the solution which promotes secondary circulation in the bulk flow. These secondary flows have significant effect on the heat transfer coefficient. Some soluble polymeric additives in water have shear thinning effect on the solutions, which lead to a significant reduction in frictional loss as well as a modest increase in the heat transfer coefficient. Some of the additives used are polystyrene spheres suspension in oil and injection of gas bubbles.

- **Segmental Baffle Heat Exchanger with Continuous Helical Baffle Heat Exchanger:**

The developments for shell and tube exchangers focus on better conversion of pressure drop into heat transfer i.e. higher Heat transfer co-efficient to Pressure drop ratio, by improving the conventional baffle design. With single segmental baffles, most of the overall pressure drop is wasted in changing the direction of flow. This kind of baffle arrangement shown in Fig. 3.9 also leads to more grievous undesirable effects such as dead spots or zones of recirculation which can cause increased fouling, high leakage flow that bypasses the heat transfer surface giving rise to lesser heat transfer co-efficient, and large cross flow. The cross flow not only reduces the mean temperature difference but can also cause potentially damaging tube vibration.

The baffles are of primary importance in improving mixing levels and consequently enhancing heat transfer of shell-and-tube heat exchangers. However, the segmental baffles have some adverse effects such as large back mixing, fouling, high leakage flow, and large cross flow, but the main shortcomings of segmental baffle design remain.

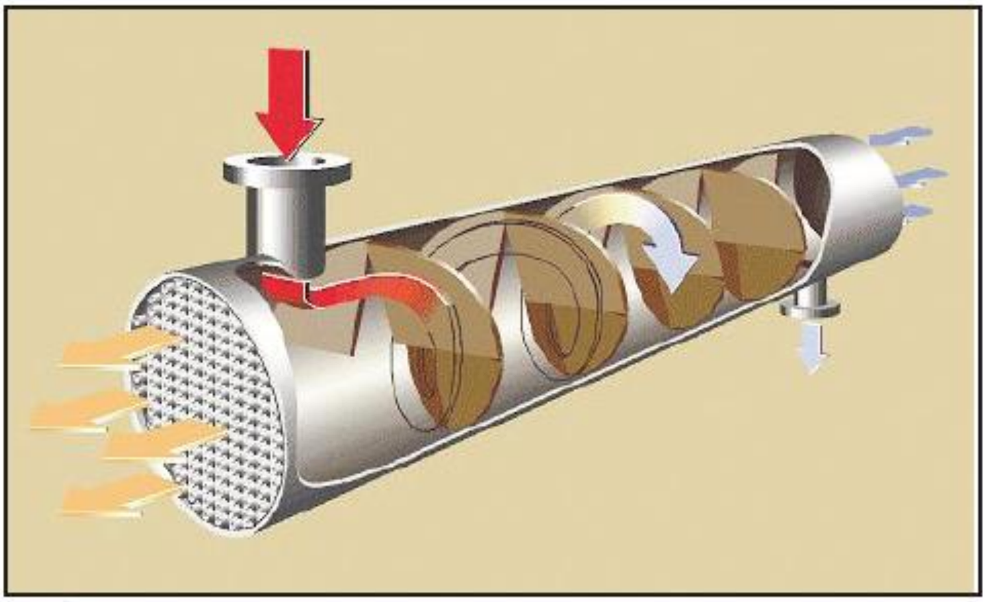


Fig. 3.9 Helical Baffle Heat Exchanger

Compared to the conventional segmental baffled shell and tube exchanger Helixchanger offers the following general advantages:

i. Increased heat transfer rate/ pressure drop ratio
ii. Reduced bypass effects
iii. Reduced shell side fouling
iv. Prevention of flow induced vibration

3.2.2 Active Techniques:

These techniques are more complex from the use and design point of view as the method requires some external power input to cause the desired flow modification and improvement in the rate of heat transfer. It finds limited application because of the need of external power in many practical applications. In comparison to the passive techniques, these techniques have not shown much potential as it is difficult to provide external power input in many cases. Various active techniques are as follows:

− **Mechanical Aids**:

Examples of the mechanical aids include rotating tube exchangers and scrapped surface heat and mass exchangers. These devices stir the fluid by mechanical means or by rotating the surface.

− **Surface vibration**:

They have been used primarily in single phase flows. A low or high frequency is applied to facilitate the surface vibrations which results in higher convective heat transfer coefficients.

− **Fluid vibration**:

Instead of applying vibrations to the surface, pulsations are created in the fluid itself. This kind of vibration enhancement technique is employed for single phase flows.

− **Electrostatic fields**:

Electrostatic field like electric or magnetic fields or a combination of the two from DC or AC sources is applied in heat exchanger systems which induces greater bulk mixing, force convection or electromagnetic pumping to enhance heat transfer. This technique is applicable in heat transfer process involving dielectric fluids.

− **Injection**:

In this technique, same or other fluid is injected into the main bulk fluid through a porous heat transfer interface or upstream of the heat transfer section. This technique is used for single phase heat transfer process.

− **Suction**:

This technique is used for both two phase heat transfer and single phase heat transfer process. Two phase nucleate boiling involves the vapour removal through a porous heated surface whereas in single phase flows fluid is withdrawn through the porous heated surface.

− **Jet impingement:**

This technique is applicable for both two phase and single phase heat transfer processes. In this method, fluid is heated or cooled perpendicularly or obliquely to the heat transfer surface.

3.2.3 Compound Techniques:

A compound augmentation technique is the one where more than one of the above mentioned techniques is used in combination with the purpose of further improving the thermo-hydraulic performance of a heat exchanger. Fig. 3.10 show test section fitted with wire coil and twisted tape insert.

Fig. 3.10 Test section fitted with wire coil and twisted tape inserts

CHAPTER 4
PRESENT EXPERIMENTAL WORK

4.1 Components of Experimental Setup:

- **Centrifugal Blower**

The air is forced through the test section by using a centrifugal blower. It is run by an AC electric motor having following specifications:

Make: Maruti Enterprise (Captain Motorized Blower)

Model: 50	HP: 0.28
Amp: 1	RPM: 2800
Watts: 210	Ins.: Class-A

The blower is connected with the circular to square shape connector which carries the air to test section. The suction side of blower is provided with flow control valve which controls amount of air passing to the test section.

- **Heater**

To provide uniform heat flux over all the surfaces of test section the plate type heaters are used. The heater is mounted on test section with the help of nut and bolt. Four plate type heaters, each of capacity 500 watts are connected in parallel. Heat flux is controlled from the control panel which contains dimmer stat to vary the heat input.

- **Test Section**

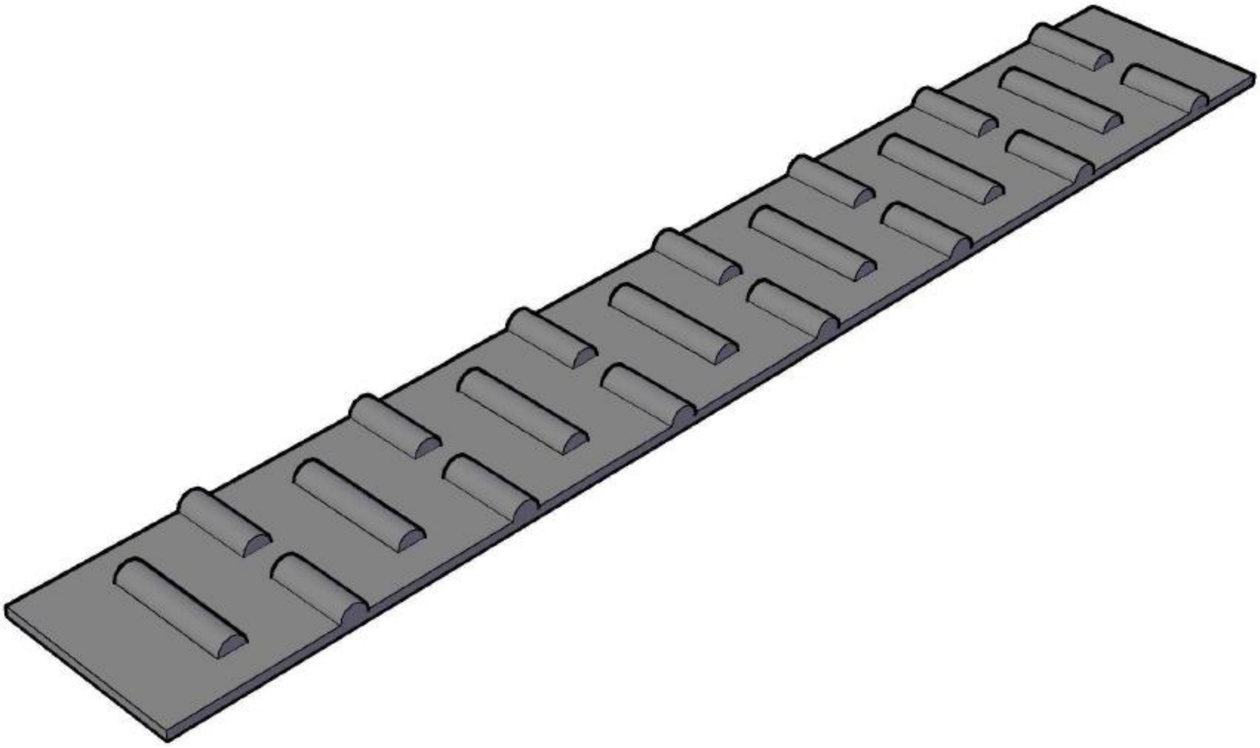
The specifications of test section used are as follows:

- Length of test section: 750 mm
- Inner dimensions of test section: 110 mm X 110 mm
- Outer dimensions of test section: 120 mm X 120 mm
- Thickness of test section: 5 mm
- Material of test section: Aluminum
- Cross-section of test section: Square

4.2 Types of Inserts Used:

During the experimentation Semicircular shaped ribs are used as the inserts inside the square duc. To change the disturbance pattern during the flow for different configurations of Semi circular rib, the rib is again broken into different pieces and these pieces are staggered in the direction of flow of air as shown in below Fig.

The material used for rib is aluminium. The Circular rod of aluminium having cross section is used in this experimentation.



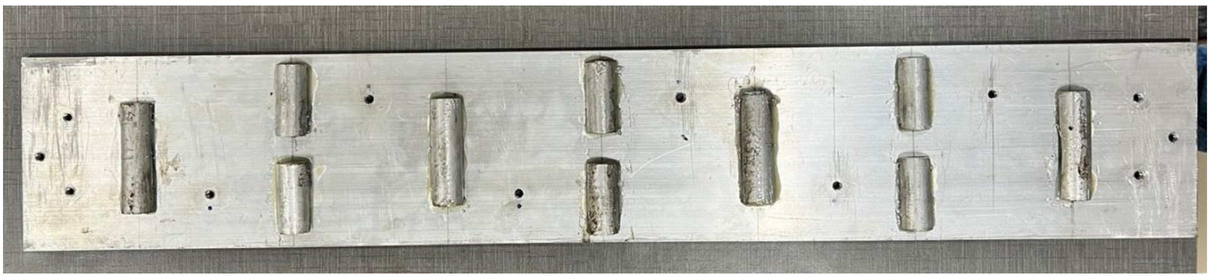
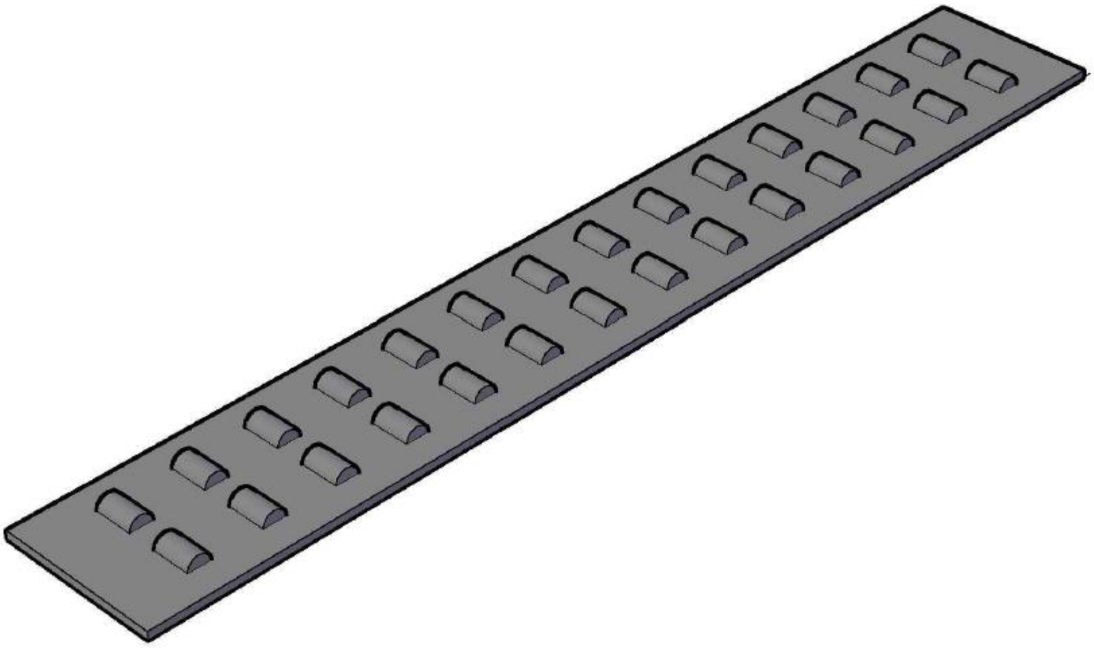


Fig. 4.1 Hybrid Ribs Type Plate



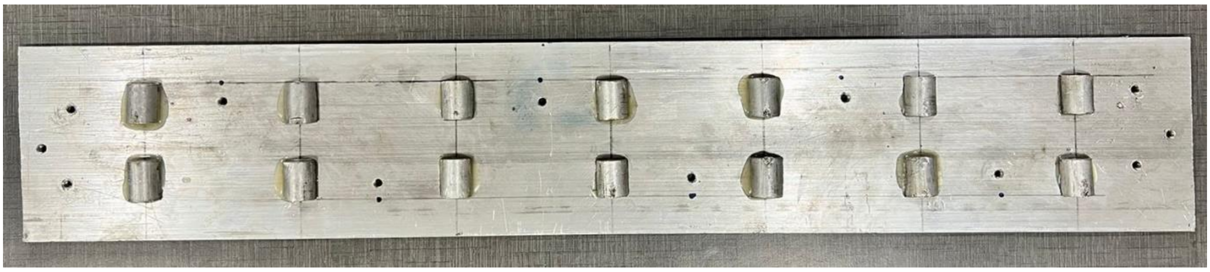
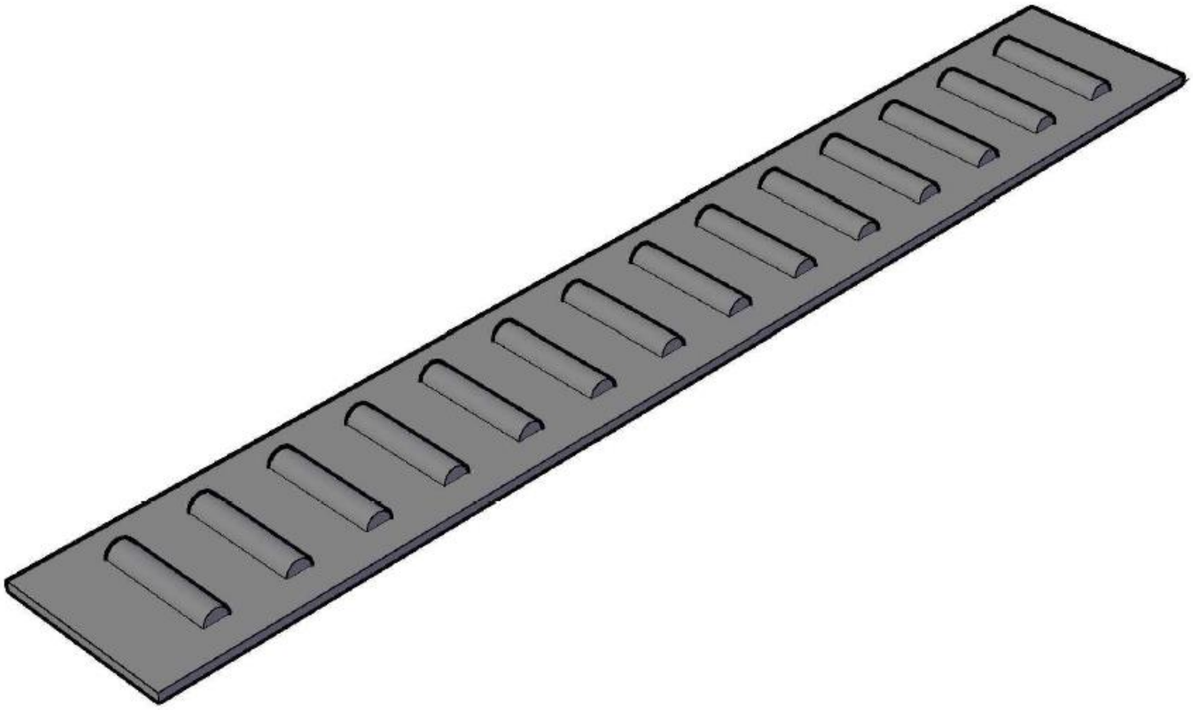


Fig. 4.2 2 Slot Ribs Type Plate

Fig.4.3 Center Slot Ribs Type Plate



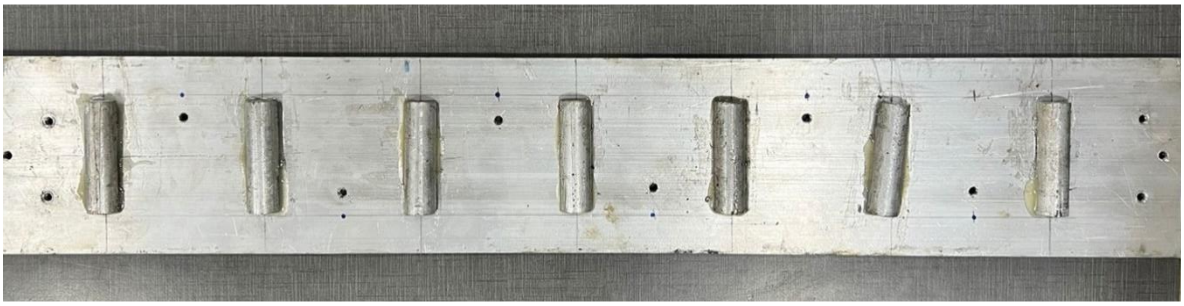


Fig. 4.4 2 Slot Ribs Type Plate

Fig. 4.5 zig zag Slot Ribs Type Plate

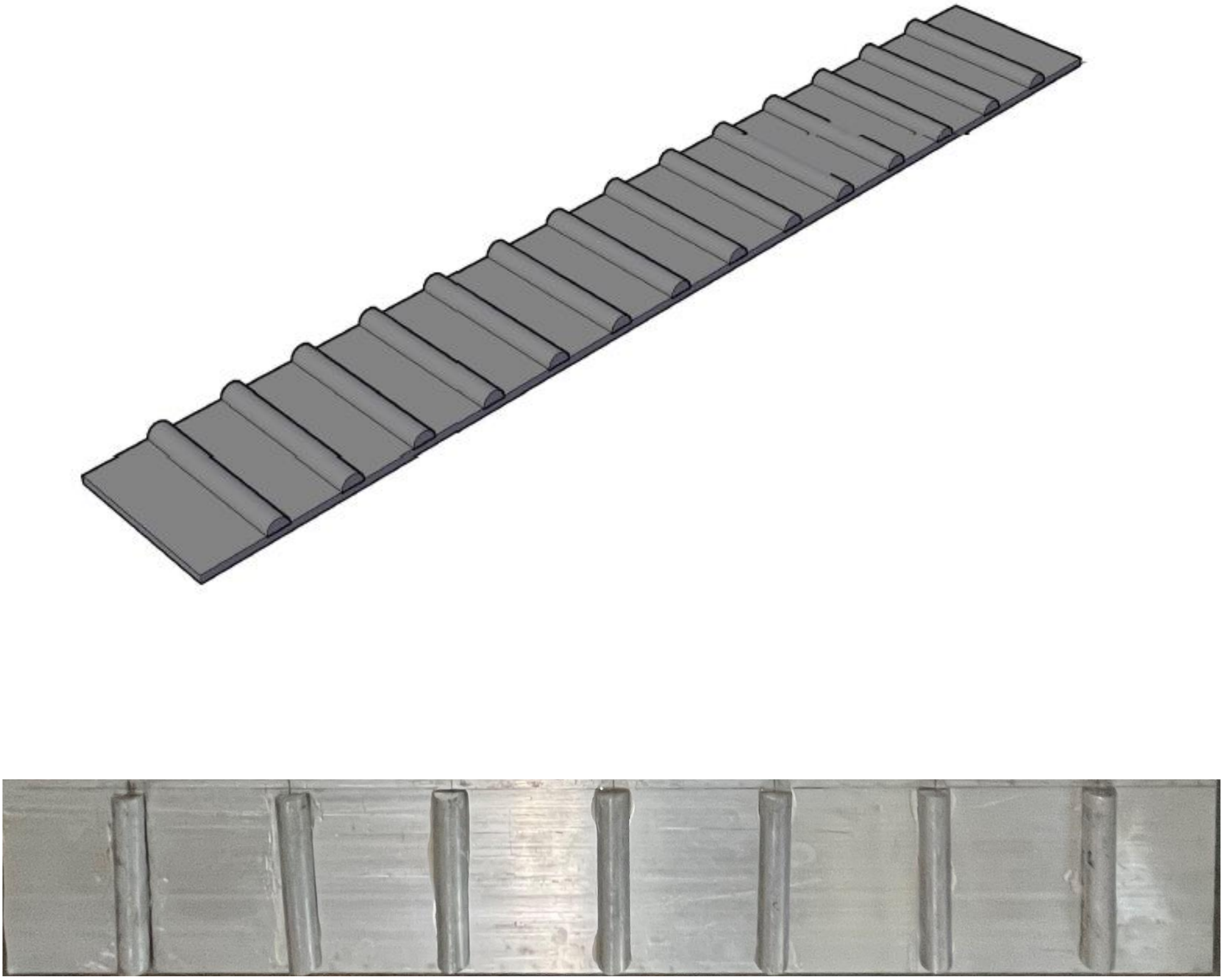


Fig. 4.6 Continuous Slot Ribs Type Plate

4.3 Experimental work

Fig. 4.7 Photograps While Preparing Rib Plates

Fig. 4.8 Photograph While Assembling Duct

Fig. 4.9 Photograph of Actual Assembled Duct

Fig. 4.10- Photograph of Experimental Set-up

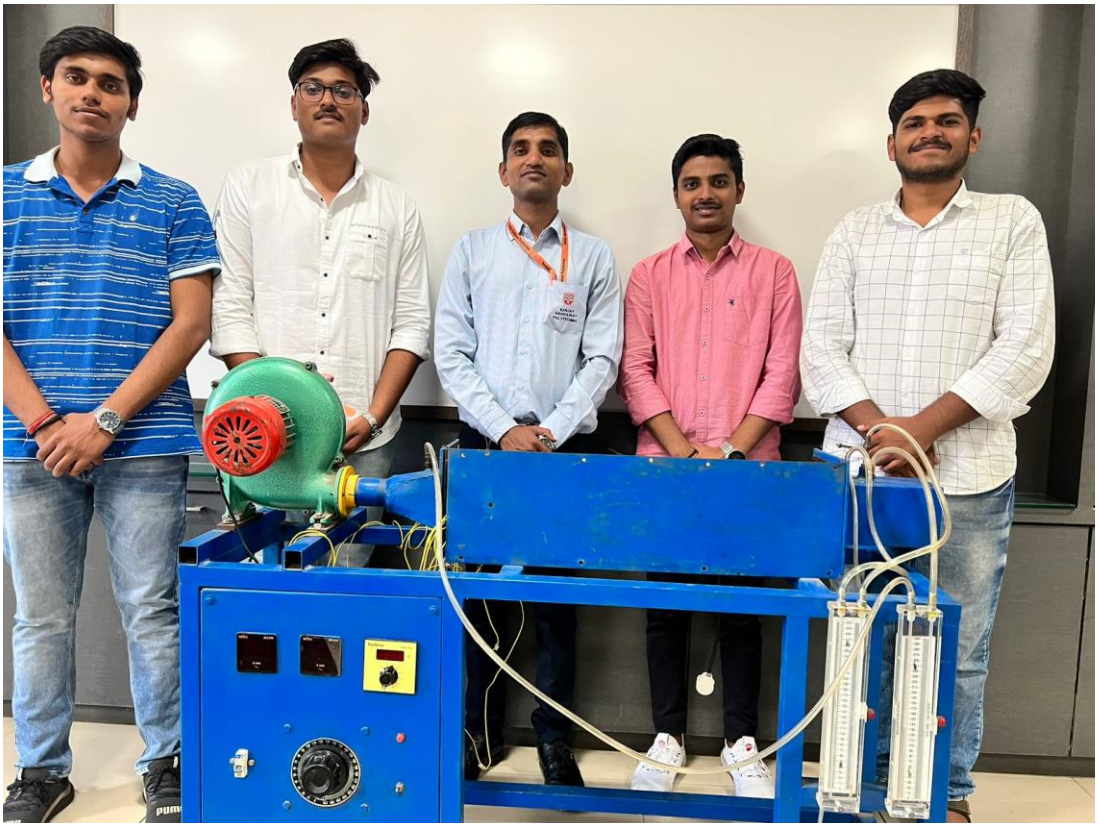


Fig 4.11 Photograph of Project With Guide

4.4 Calculation Procedure:

1. Note down all the parameters displayed on control panel which includes all the temperatures, voltage and current values.
2. Note down manometer readings giving pressure drop across the orifice and test section.
3. Calculate average temperature of duct wall and bulk temperature of air.
4. Obtain properties of air like thermal conductivity, kinematic viscosity, Prandtl number from the air table corresponding to above bulk temperature of air.
5. Calculate mass flow rate and velocity of air flowing through the test section.
6. Calculate convective heat transferred to air.
7. Calculate convective heat transfer coefficient.

CHAPTER 5

OBSERVATIONS FOR PLAIN SQUARE DUCT AND SQUARE DUCT PROVIDED WITH DIFFERENT INSERTS

Table- 5.1: Observations for Plane Duct

Sr no	Vtg.	Current	Air Velocity	Air Inlet temp.	Air Outlet temp.	Plate Surface Temp.						Manometer reading		
						T_1	T_2	T_3	T_4	T_5	T_6	P1	P2	P1-P2
1	200	1.875	2.1	30	33.4	51	51	50	51	52	52	147	145	2
2	200	1.875	3	29.9	33.1	45	46	46	47	46	46	145	142	3
3	200	1.875	4	29.8	32.5	43	43	42	42	43	42	145	141	4
4	200	1.875	4.8	30.1	32	37	38	38	38	37	38	146	141	5

Table- 5.2: Observations for Hybrid Type Ribs

Sr no	Vtg.	Current	Air Velocity	Air Inlet temp.	Air Outlet temp.	Plate Surface Temp.						Manometer reading		
						T_1	T_2	T_3	T_4	T_5	T_6	P1	P2	P1-P2
1	200	1.875	2.1	29.8	35.8	56	56	55	56	57	57	147	145	2
2	200	1.875	3	30	34.5	50	51	51	52	51	51	145	140	5
3	200	1.875	4	29.9	33.9	48	48	47	47	48	47	145	139	6
4	200	1.875	4.8	30.1	33	42	43	43	43	42	43	146	137	11

Table- 5.3: Observations for 3 Slot Type Ribs

Sr no	Vtg.	Current	Air Velocity	Air Inlet temp.	Air Outlet temp.	Plate Surface Temp.						Manometer reading		
						T_1	T_2	T_3	T_4	T_5	T_6	P1	P2	P1-P2
1	200	1.875	2.1	30	36.6	56	54	53	54	55	55	147	145	2
2	200	1.875	3	29.9	35.2	52	53	53	52	53	53	148	143	5
3	200	1.875	4	29.8	33.8	47	47	46	45	45	46	147	139	8
4	200	1.875	4.8	29.8	32.8	43	42	43	43	42	43	146	136	10

Table- 5.4: Observations for Center Slot Type Ribs

Sr no	Vtg.	Current	Air Velocity	Air Inlet temp.	Air Outlet temp.	Plate Surface Temp.						Manometer reading		
						T_1	T_2	T_3	T_4	T_5	T_6	P1	P2	P1-P2
1	200	1.875	2.1	29.8	35.9	58	59	58	60	58	59	143	140	3
2	200	1.875	3	29.9	34.4	51	52	52	51	52	52	144	140	4
3	200	1.875	4	29.5	33.9	46	46	47	47	48	47	145	138	7
4	200	1.875	4.8	29.6	33.3	43	44	43	44	43	43	148	139	9

Table- 5.5: Observations for 2Type Ribs

Sr no	Vtg.	Current	Air Velocity	Air Inlet temp.	Air Outlet temp.	Plate Surface Temp.						Manometer reading		
						T_1	T_2	T_3	T_4	T_5	T_6	P1	P2	P1-P2
1	200	1.875	2.1	30	36.6	58	56	57	57	58	57	146	143	3
2	200	1.875	3	30.2	35.8	53	52	53	53	52	52	145	141	4
3	200	1.875	4	30	34.6	47	48	49	47	47	48	144	138	6
4	200	1.875	4.8	29.8	33.6	43	44	43	46	44	45	148	140	8

Table- 5.6: Observations for zig zag Slot Type Ribs

Sr no	Vtg.	Current	Air Velocity	Air Inlet temp.	Air Outlet temp.	Plate Surface Temp.						Manometer reading		
						T_1	T_2	T_3	T_4	T_5	T_6	P1	P2	P1-P2
1	200	1.875	2.1	30.3	37.3	58	59	58	58	59	58	144	141	3
2	200	1.875	3	30	36.2	52	53	53	53	52	53	144	139	4
3	200	1.875	4	30.1	35.2	49	48	48	47	48	47	143	137	6
4	200	1.875	4.8	30	34.6	44	43	44	43	43	44	147	139	8

Table- 5.7: Observations for Continuous Slot Type Ribs

Sr no	**Vtg.**	**Current**	**Air Velocity**	**Air Inlet temp.**	**Air Outlet temp.**	**Plate Surface Temp.**						**Manometer reading**		
						T_1	**T_2**	**T_3**	**T_4**	**T_5**	**T_6**	P1	P2	P1-P2
1	200	1.875	2.1	30	37.8	54	53	54	54	53	53	143	140	3
2	200	1.875	3	30	37.2	53	53	54	53	54	53	143	138	5
3	200	1.875	4	29.9	35.6	49	50	48	47	48	48	143	135	8
4	200	1.875	4.8	30.2	34.4	56	45	44	44	43	44	147	137	10

Table-5.8: Air Properties for Hybrid Ribs configuration

Sr no.	Average Temp. of Wall of Duct	Bulk Temp. 0f Air	Density of air (ρa)	Specific heat of air (Cp)	Kinematic viscosity of air (υ)	Prandtl no. (Pr)	Thermal conductivity (K)
1	56.17	32.8	1.154	1007	0.0000163	0.7133	0.02655
2	51.00	32.25	1.156	1007	0.00001625	0.7134	0.02651
3	47.50	31.9	1.158	1007	0.00001621	0.7134	0.02648
4	42.67	31.55	1.159	1007	0.00001618	0.7135	0.02646

Table- 5.9: Air Properties for 3 Slot Type Ribs

Sr no.	Average Temp. of Wall of Duct	Bulk Temp. 0f Air	Density of air (ρa)	Specific heat of air (Cp)	Kinematic viscosity of air (υ)	Prandtl no. (Pr)	Thermal conductivity (K)
1	54.50	33.3	1.152	1007	0.00001635	0.7132	0.02658
2	52.67	32.55	1.155	1007	0.00001628	0.7133	0.02653
3	46.00	31.8	1.158	1007	0.00001621	0.7134	0.02647
4	42.67	31.3	1.16	1007	0.00001616	0.7135	0.02644

3 slots

Table- 5.10: Air Properties for Center Slot Type Ribs

Sr no.	Average Temp. of Wall of Duct	Bulk Temp. 0f Air	Density of air (ρa)	Specific heat of air (Cp)	Kinematic viscosity of air (υ)	Prandtl no. (Pr)	Thermal conductivity (K)
1	58.67	32.85	1.154	1007	1.63E-05	0.7133	0.02655
2	51.67	31.95	1.157	1007	1.62E-05	0.7134	0.02648
3	46.83	31.75	1.158	1007	1.62E-05	0.7134	0.02647
4	43.33	31.45	1.159	1007	1.62E-05	0.7135	0.02645

Table- 5.11: Air Properties for 2Type Ribs

Sr no.	Average Temp. of Wall of Duct	Bulk Temp. 0f Air	Density of air (ρa)	Specific heat of air (Cp)	Kinematic viscosity of air (υ)	Prandtl no. (Pr)	Thermal conductivity (K)
1	57.17	33.3	1.152	1007	1.64E-05	0.7132	0.02658
2	52.50	33	1.153	1007	1.63E-05	0.7132	0.02656
3	47.67	32.3	1.156	1007	1.63E-05	0.7134	0.02651
4	44.17	31.7	1.158	1007	1.62E-05	0.7135	0.02647

Table- 5.12: Air Properties for zig zag Slot Type Ribs

Sr no.	Average Temp. of Wall of Duct	Bulk Temp. 0f Air	Density of air (ρa)	Specific heat of air (Cp)	Kinematic viscosity of air (υ)	Prandtl no. (Pr)	Thermal conductivity (K)
1	58.33	33.8	1.15	1007	1.64E-05	0.7131	0.02662
2	52.67	33.1	1.153	1007	1.63E-05	0.7132	0.02657
3	47.83	32.65	1.155	1007	1.63E-05	0.7133	0.02654
4	43.50	32.3	1.156	1007	1.63E-05	0.7134	0.02651

Table- 5.13: Air Properties for Continuous Slot Type Ribs

Sr no.	Average Temp. of Wall of Duct	Bulk Temp. 0f Air	Density of air (ρa)	Specific heat of air (Cp)	Kinematic viscosity of air (υ)	Prandtl no. (Pr)	Thermal conductivity (K)
1	53.50	33.9	1.15	1007	1.64E-05	0.7131	0.02663
2	53.33	33.6	1.151	1007	1.64E-05	0.7131	0.02661
3	48.33	32.75	1.154	1007	1.63E-05	0.7133	0.02654
4	46.00	32.3	1.156	1007	1.63E-05	0.7134	0.02651

Table- 5.14: Result of Hybrid Type Ribs

Sr no.	Manometric Head of Air (hm)	Equivalent height of air column (hair)	Mass flow rate of air (m)	Convective heat transferred to air (Q)	Convective heat transfer coefficient (h)
1	0.002	1.73	0.0293	177.1704	45.9526
2	0.005	4.33	0.0420	190.1544	61.4641
3	0.006	5.18	0.0560	225.7581	87.7071
4	0.009	7.77	0.0673	196.5792	107.1714

Table- 5.15: Result of for 3 Slot Type Ribs

Sr no.	Manometric Head of Air (hm)	Equivalent height of air column (hair)	Mass flow rate of air (m)	Convective heat transferred to air (Q)	Convective heat transfer coefficient (h)
1	0.002	1.74	0.0293	194.5497	55.6174
2	0.005	4.33	0.0419	223.7659	67.4146
3	0.008	6.91	0.0560	225.7581	96.3543
4	0.01	8.62	0.0674	203.5332	108.5221

Table- 5.16: Result of for Center Slot Type Ribs

Sr no.	Manometric Head of Air (hm)	Equivalent height of air column (hair)	Mass flow rate of air (m)	Convective heat transferred to air (Q)	Convective heat transfer coefficient (h)
1	0.003	2.60	0.0293	180.1233	42.2849
2	0.004	3.46	0.0420	207.2362	63.7013
3	0.007	6.04	0.0560	242.6900	97.5148
4	0.009	7.77	0.0673	250.8079	127.9143

Table- 5.17: Result of for 2Type Ribs

Sr no.	Manometric Head of Air (hm)	Equivalent height of air column (hair)	Mass flow rate of air (m)	Convective heat transferred to air (Q)	Convective heat transfer coefficient (h)
1	0.003	2.60	0.0293	194.5497	49.4032
2	0.004	3.47	0.0419	236.0225	73.3559
3	0.006	5.19	0.0560	259.1734	102.2179
4	0.008	6.91	0.0673	257.3643	125.1163

Table- 5.18: Result of for zig zag Slot Type Ribs

Sr no.	Manometric Head of Air (hm)	Equivalent height of air column (hair)	Mass flow rate of air (m)	Convective heat transferred to air (Q)	Convective heat transfer coefficient (h)
1	0.003	2.61	0.0292	205.9824	50.8850
2	0.005	4.34	0.0419	261.3106	80.9387
3	0.006	5.19	0.0559	287.0959	114.5977
4	0.008	6.92	0.0671	311.0081	168.2944

Table- 5.19: Result of for Continuous Slot Type Ribs

Sr no.	Manometric Head of Air (hm)	Equivalent height of air column (hair)	Mass flow rate of air (m)	Convective heat transferred to air (Q)	Convective heat transfer coefficient (h)
1	0.02663	0.003	0.0292	229.5232	70.9719
2	0.02661	0.005	0.0418	302.9311	93.0378
3	0.02654	0.008	0.0559	320.5941	124.6841
4	0.02651	0.01	0.0671	283.9639	125.6200

CHAPTER 6

SAMPLE CALCULATIONS

Sample Calculation for Table 5.4 Reading No.:01

Average temperature of duct wall (T_s):

$$Ts = \frac{T1 + T2 + T3 + T4 + T5 + T6}{6}$$

$$= \frac{56+56+55+56+57+57}{6}$$

$$Ts = 56.17$$

Bulk temperature of air (T_b):

$$Tb = \frac{\text{Tair in } + \text{Tair out}}{2}$$

$$= \frac{29.8+35.8}{2}$$

$$Tb = 32.8°C$$

Properties of air from the air table corresponding to above bulk temperature of air are:

- Density of air (ρ_a)= 1.154 kg/m^3
- Specific heat of air (C_p)= 1007 J/kg.K
- Kinematic viscosity of air (υ)= 1.630 x 10^{-5} m^2/s
- Prandtl no. (Pr)= 0.7133
- Thermal conductivity (k)= 0.02655 W/m.K

Area:

Convective heat transfer area (A_c):

$$A_c = 4W.L$$

$$= 4 \times 0.11 \times 0.75$$

$$A_c = 0.33 \text{ m}^2$$

Test section duct flow area (A):

$$A = W.H$$

$$= 0.11 \times 0.11$$

$$A = 0.0121 \text{ m}^2$$

Convective heat transferred to air (Q):

$Q = m \times C_p \times (T_{air\ out} - T_{air\ in})$

$= 0.0384 \times 1006.8 \times (35.8 - 29.8)$

$Q = 177.1704$ W

Convective heat transfer coefficient (h):

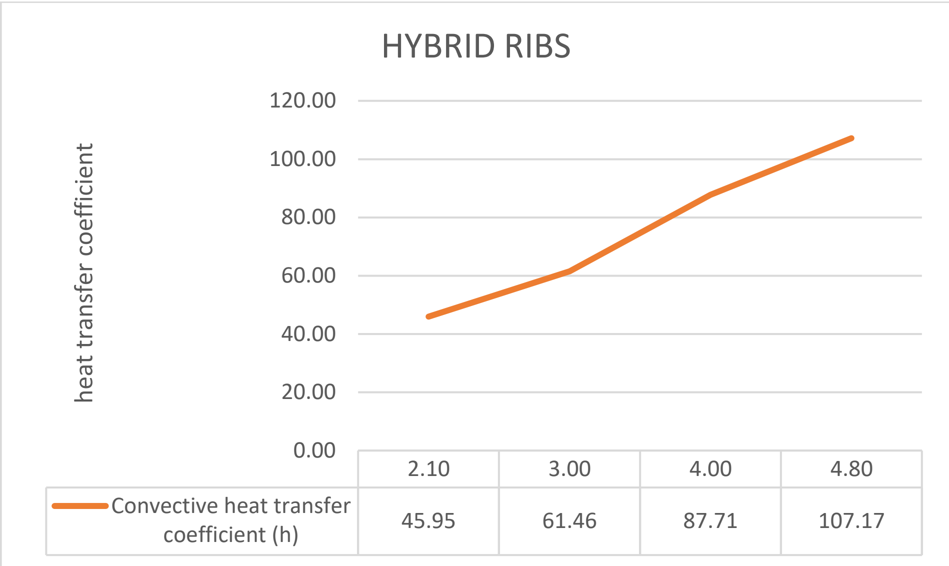
$h = \frac{Q}{Ac \text{ x } (Ts - Tb)}$

$= \frac{247.3264}{0.33 \times (56.17 - 32.8)}$

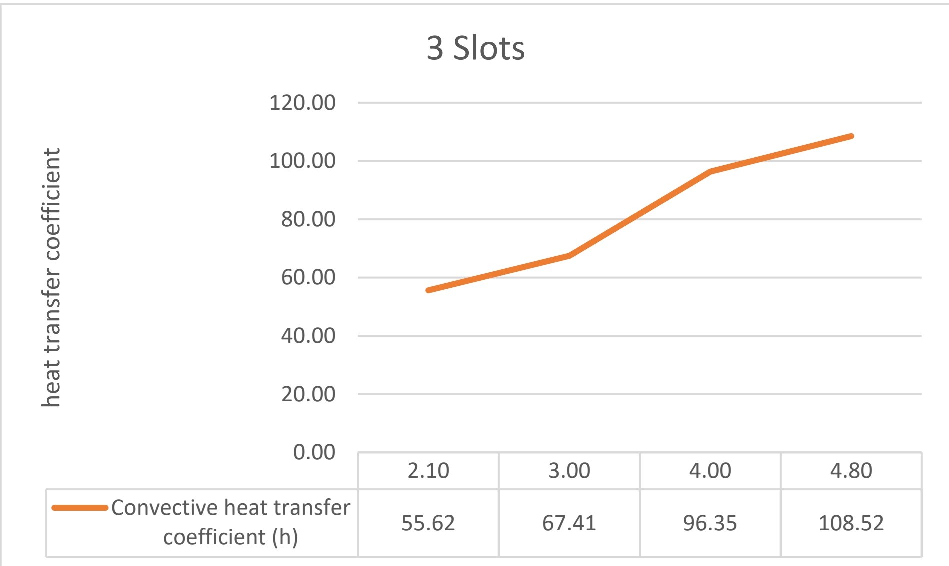
$h = 45.9526$ W/m^{2}K

CHAPTER 7
RESULTS

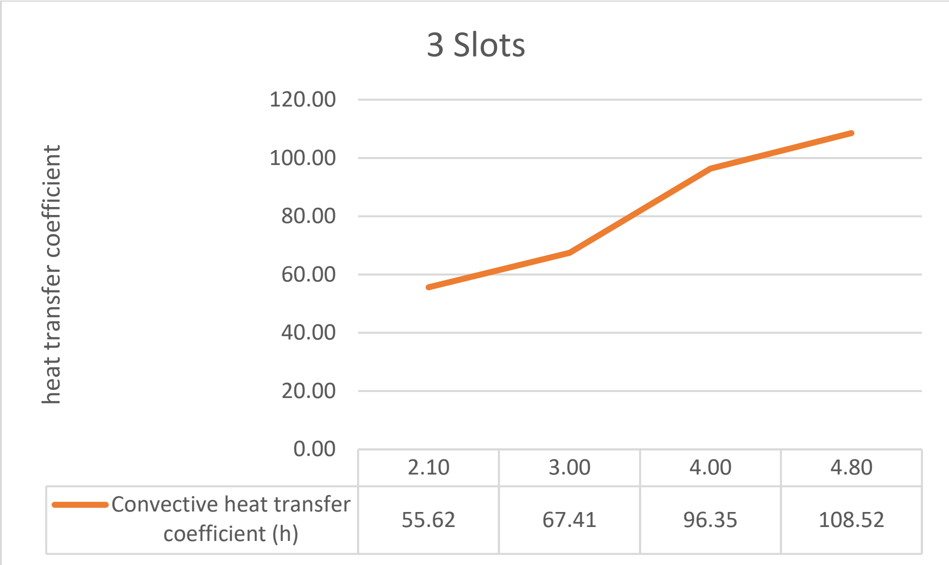
7. 1 Graphs of Heat transfer Coeffient vs Air Velovity for different ribs turbulators



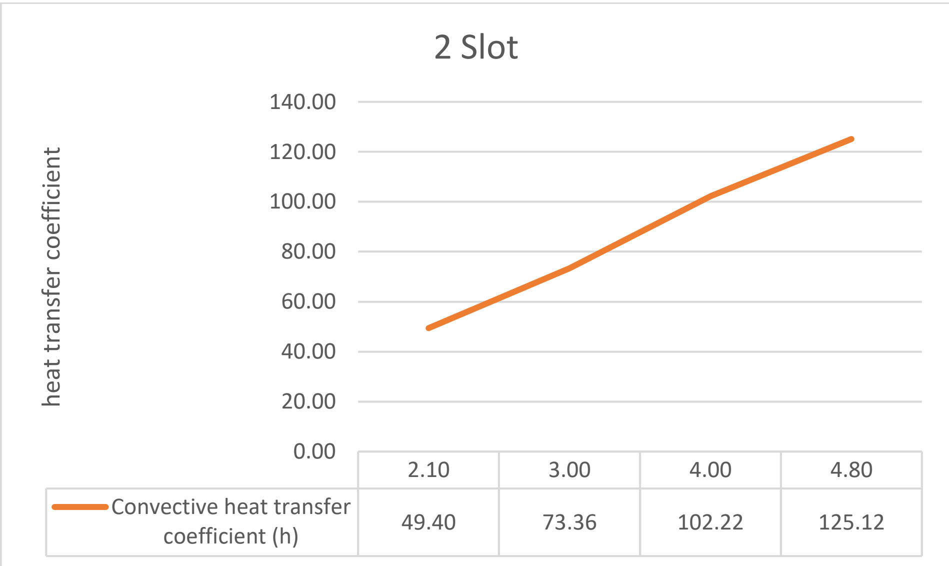
GRAPH: 7.1: Heat Transfer Coefficient Vs Air Velocity For Hybrid Ribs



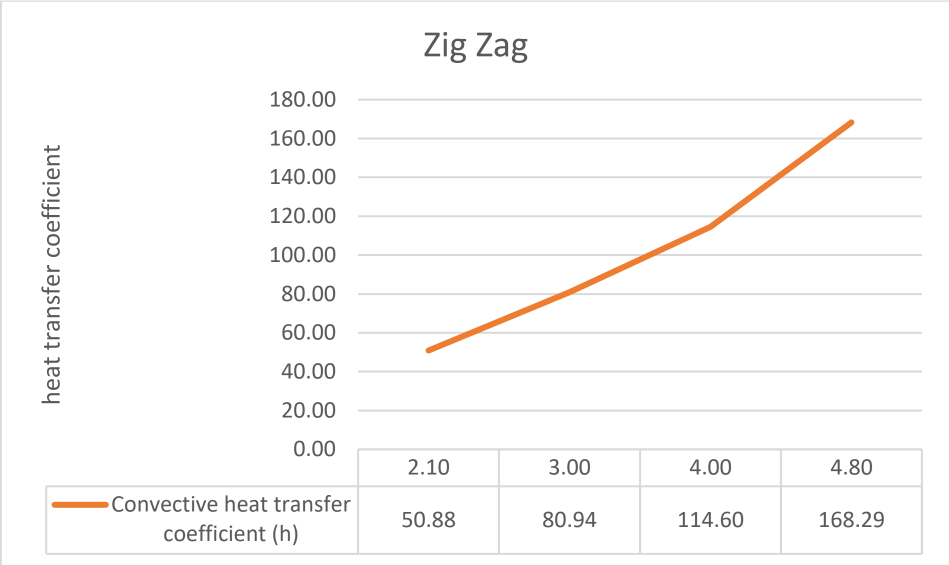
GRAPH: 7.2: Heat Transfer Coefficient Vs Air Velocity For 3 Slots Rib



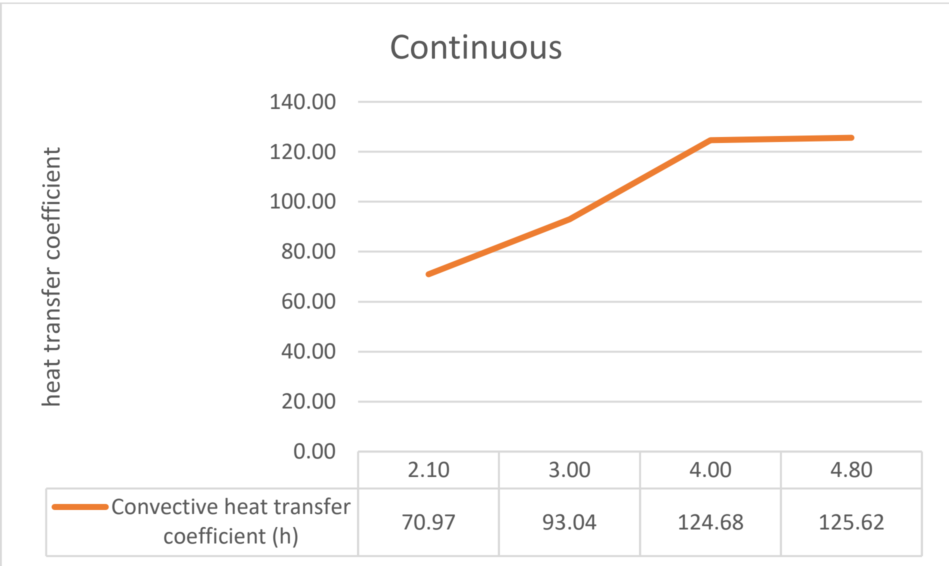
GRAPH: 7.3: Heat Transfer Coefficient Vs Air Velocity For 3 Slots Rib



GRAPH: 7.4: Heat Transfer Coefficient Vs Air Velocity For 2 Slot Ribs



GRAPH: 7.5: Heat Transfer Coefficient Vs Air Velocity For Zig Zag Ribs



GRAPH: 7.6: Heat Transfer Coefficient Vs Air Velocity For Continuous Rib

CHAPTER 8
CONCLUSIONS

The experimental investigation of forced convection heat transfer in a square duct provided with different configurations of Semi-Circular rib turbulators has been carried out.

The effect of different types of Semi-Circular rib turbulators on the heat transfer coefficient and friction factor is studied

Table. 8.1

Sr no	Type of rib turbulator	Heat transfer coefficient Min -Max
1	HYBRID RIBS	45.95 - 107.17
2	3 Slots	55.61 - 108.52
3	Center slot	42.28 - 127.91
4	2 Slot	49.40 - 125.11
5	Zig Zag	50.88 - 168.29
6	Continuous	70.97 - 125.62

CHAPTER 9
FUTURE SCOPE OF WORK

Further studies can be done using this study as base. Some of the possibilities are mentioned below:

1. **Material of the rib** can be varied like copper, mild steel etc.
2. Experimental work can be performed by using **compound technique** of heat transfer enhancement i.e. using any other types of inserts like twisted tape, spring along with these inserts.

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RESEARCH PAPER

Review of Forced Convection Heat Transfer through Rectangular or Square Duct Provided with Different Configurations of Rib Turbulators

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Abstract- Improving the convection heat exchange coefficient is the key to progress the execution of a heat exchange. In common, heat exchangers are expecting to be littler and more reasonable. There are two common sorts of methods that can be utilized to upgrade heat exchange. There's a detached strategy, such as bent tapes, helical screw tape embeds, unpleasant surfaces, amplified surfaces, and fluid and gas added substances. Dynamic strategies, on the other hand, require extra control, such as mechanical helps, liquid vibrations, or electrostatic areas. Comparatively, inactive strategies are found to be more reasonable than dynamic strategies.

Ribs are common warm exchange improvement gadgets that can be utilized in a assortment of heat-exchanging channels. As a result of decreased liquid stream region caused by stream blockages such as ribs, weight drops increment and gooey impacts increment. Distribution, reattachment, and auxiliary stream are all included within the flow around ribs. Also, auxiliary stream gives distant better;a much better;a higher;a stronger;an improved">an improved warm contact between surface and liquid because it makes twirl between surface and liquid. It comes about in a blending of liquid that upgrades the warm angle, which eventually leads to an increment in warm exchange coefficient. An exploratory consider of warm exchange and grinding figure of a square channel with embeds beneath turbulent stream conditions is displayed in this paper. In plain square conduits, with or without embeds, discuss is considered the working liquid. An exploratory set up is created in arrange to assess the warm exchange coefficient and grinding figure. To start with, tests are conducted in plain straight square conduits with and without embeds and the comes about are compared to those within the writing**.**