A PROJECT REPORT

ON

"Heat transfer analysis of square and circular perforated fins in staggered arrangement."

Submitted by

HAJWANE YASER

MALIK SAQUIB

SHAH UVAIS

SHAIKH NAVED

In partial fulfillment for the award of the Degree

Of

BACHELOR OF ENGINEERING

IN

MECHANICAL ENGINEERING

UNDER THE GUIDANCE

Of

Prof. RIZWAN SHAIKH

DEPARTMENT OF MECHANICAL ENGINEERING

ANJUMAN-I-ISLAM

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NAVI MUMBAI – 410206

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(Approved by AICTE, recg. By Maharashtra Govt. DTE,

Affiliated to Mumbai University)

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Tel.: +91 22 27481247/48 * Website: www.aiktc.org

CERTIFICATE

This is to certify that the project entitled

"Heat transfer analysis of square and circular perforated fins in staggered arrangement."

Submitted by,

HAJWANE YASER

MALIK SAQUIB

SHAH UVAIS

SHAIKH NAVED

To the Kalsekar Technical Campus, New Panvel is a record of bonafide work carried out by him under our supervision and guidance, for partial fulfillment of the requirements for the award of the Degree of Bachelor of Engineering in Mechanical Engineering as prescribed by **University Of Mumbai**, is approved.

Project co-guide	**Internal Examiner**	**External Examiner**
(Prof .RIZWAN SHAIKH)	(Prof.)	(Prof.)

Head of Department	**Director**
(Prof. ZAKIR ANSARI)	(Dr.ABDUL RAZZAK HONNUTAGI)

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APPROVAL OF DISSERTATION

This is to certify that the thesis entitled

"Heat transfer analysis of square and circular perforated fins in staggered arrangement.**"**
Submitted by,

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In partial fulfillment of the requirements for the award of the Degree of Bachelor of Engineering in Mechanical Engineering, as prescribed by University of Mumbai approved.

(Internal Examiner) **(External Examiner)**

___________________ ___________________

Date: ___________

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ABSTRACT

The experiment on heat transfer enhancement over a surface equipped with square and cylindrical pin fins with cylindrical perforations in a rectangular channel in a staggered manner. The channel has a cross-section of 100x250 mm^2. The experiment covered the following range Re=13,500-42000, Clearance ratio=0, Inter –fin spacing Sy/D=1.208, 1.524.

Co-relation equations were developed for the heat transfer, Friction factor, Effectiveness. The experimental result showed that the use of cylindrical fin gives the max heat transfer enhancement. Effectiveness varies from 1.1-1.5 depending upon the inter-fin spacing. The friction factor also decreases. Thus the maximum effectiveness is obtained at minimum velocity and more inter-fin spacing. The average Nusselt number calculated on the basis of projected area increased with decreasing inter-fin spacing ratio. The friction factor increased with decreasing inter-fin distance ratio. Enhancement efficiencies increased with decreasing Reynolds number. Therefore, relatively lower Reynolds number led to an improvement in the heat transfer performance.

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NOMENCLATURE:

A	Heat transfer area
D_h	hydraulic diameter of duct
D	Diameter of pins
d	Diameter of perforation
f	Friction factor
h	Heat transfer coefficient
H	Height of the fins
Sy/D	Interfins spacing
C/H	Clearance ratio
L	Length of base plate
W	Width of base plate
I	Current
V	Voltage
U	Mean velocity of the air
ν	Kinematic viscosity
ε	Effectiveness

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION TO HEAT TRANSFER

Heat transfer is a science that studies the energy transfer between two bodies due to temperature difference. This temperature difference is thought of as a driving force that causes heat to flow.Heat transfer occurs by three basic mechanisms or modes:

1) Conduction.
2) Convection.
3) Radiation.

Following are the parameters which affect the heat transfer,

- Velocity of air
- Density of air
- **Viscosity**
- Dynamic (absolute) Viscosity
- Kinematic viscosity
- Material of fins.
- Nusselt Number.
- Reynolds Number.
- Prandtl Number. [12]

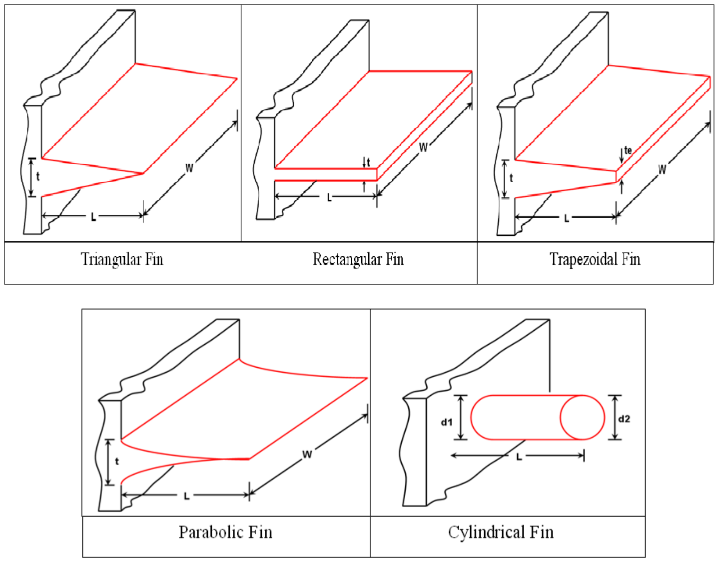
1.2INTRODUCTION OF FINS

Extended surfaces (fins) are frequently used in heat exchanging devices for the purpose of increasing the heat transfer between a primary surface and the surrounding fluid. Various types of heat exchanger fins, ranging from relatively simple shapes, such as rectangular, square, cylindrical, annular, tapered or pin fins, to a combination of different geometries, have been used. These fins may protrude from either a rectangular or cylindrical base. One of the commonly used heat exchanger fins is the pin fin. A pin fin is a cylinder or other shaped element attached perpendicular to a wall with the transfer fluid passing in crossflow over the element. Pin fins having a height-to-diameter ratio, H/d, between 0.5 and 4 are accepted as short fins, whereas long pin fins have a pin height-to-diameter ratio, H/d, exceeding 4. Short pin fins are widely used in the trailing edges of gas-turbine blades, in electronic cooling and in the aerospace industry. The large height-to-diameter ratio is of particular interest in heat-exchanger applications in which the attainment of a very high heat-transfer coefficient is of major concern.

Fins are used to enhance convective heat transfer in a wide range of engineering applications, and offer a practical means for achieving a large total heat transfer surface area without the use of an excessive amount of primary surface area. Fins are commonly applied for heat management in electrical appliances such as computer power supplies or substation transformers. Other applications include IC engine cooling, such as fins in a car radiator. It is important to predict the temperature distribution within the fin in order to choose the configuration that offers maximum effectiveness.

Enhancement of heat transfer is of vital importance in many industrial applications. One of the methods of enhancing heat transfer is the use of extended surfaces or fins. Extended surfaces are used to enhance heat transfer in a wide range of engineering applications and offer a practical means for achieving a large total heat transfer surface area. Fins are commonly applied for heat management in electrical appliances such as computer power supplied, or other applications include IC engine cooling such as fins in a car radiator. [11].

Various geometry of fins



FIN USES

1. Fins are most commonly used in heat exchanging devices such as radiators in cars and heat exchangers in power plants .
2. They are also used in newer technology such as hydrogen fuel cells.
3. Nature has also taken advantage of the phenomena of fins.
4. The ears of jackrabbits and Fennec Foxes act as fins to release heat from the blood that flows through them
5. Now a day’s dissipation of heat transfer is more important in field of mechanical devices such as IC engine, radiators,
6. It is also used in computer CPU for cooling purpose.

CHAPTER 2
PROBLEM DEFINITION

PROBLEM DEFINITION

- The subject of enhanced heat transfer has become much more important to industry with time elapse, use of intricate geometries were initially limited due to manufacturing process. However, new manufacturing methods allow manufacture of intricate surface geometries.
- Extended surfaces have fins attached to the primary surface. Pin fins are primarily used to increase the total rate of heat transfer. Enhanced fin geometries also increase the total rate of heat transfer.
- The effect of Reynold’s number, Friction factor and Air flow velocity are of great importance therefore they need to be evaluated.
- Therefore, development of newer fin geometries and materials to increase the heat transfer is necessary.

CHAPTER 3

OBJECTIVE

OBJECTIVE

The main objective of our experimental work can be listed as below:

- To find the effect of providing perforation on fins as compared to fins without perforation.
- To find the optimum inter-fin spacing which will give higher rate of heat transfer.
- To check the effect of staggered arrangement on friction factor, effectiveness, heat transfer rate.
- To check the effect of air velocity on the effectiveness of fins.

CHAPTER 4

LITERATURE SURVEY

LITERATURE SURVEY.

4.1) **"**BayramSahin" et al **-** Performance analysis of heat exchanger having perforated square fins. Studied the heat transfer enhancement and the corresponding pressure drop over a flat surface with square c/s perforated pin fins having inline arrangement in a rectangular section.Developed correlations equation for heat transfer,frictionfactorand enhancement efficiency.(ELSEVEIR, 2007)

4.2) **"**A.V. Zoman" et al - Heat transfer enhancement using fins with perforation.The paper deals with heat transfer enhancement by using perforated fins by varying different parameter like shape,diameter and number of perforation.

1.Higher heat transfer coefficient than solid fin.

2.Light in weight,saves material.

3. Heat transfer depends on size,shape, no. of perforations and thickness of fins.

(IJESRT, 2016)

4.3) **"**Leonardo micheli" et al - General correlations among geometry, orientation and thermal performance of natural convective micro-finned heat sinks.Correspondence between fin geometries and heat transfer coefficient as well as the effect of orientation are experimentally investigated. The heat transfer coefficient have been found to increase when the spacing is increased and fin height is decreased. (Heat and Mass transfer, 2015)

4.4) **"**RashmitaBehera" et al **-** CFD Optimization of free convective cooling of finned heat sinks: Effect of fin spacing.The flow characteristics and heat dissipation under different spacing are theoretically interpreted so the overall cooling effectiveness decreases due to lesser heat transfer rate available per unit length, hence an optimal spacing is

required.Increase in heat dissipation and fin effectiveness with increase in spacing due to enhance flow circulation. (Procedia Engineering, 2015)

4.5) **"**Amol E Dhumne" et al **-** Performance analysis of a heat exchanger having perforated cylindrical fins.Studied the heat transfer enhancement and the corresponding pressure drop over a flat surface with cylindrical c/s perforated pin fins having inline arrangement in aa rectangular section.Developed correlations equation for heat transfer, friction factor and enhancement efficiency.(IJITEE**,** 2013).

4.6) **"**Md. Farhad Ismail" et al - Numerical investigation of turbulent heat convection from solid and longitudinally perforated.To improve the cooling performance perforations such as square or circular c/s were included.Increased contact surface with the fluid than solid fins. (Procedia Engineering, 2013)

CHAPTER 5

MATERIAL PROCUREMENT

COSTING

Sr.No.	Items	Quantity	Cost Per Unit	Total Cost (Rs)
1	Temperature controller	3 No.	1450	4350
2	RTD Sensor	2 No.	520	1040
3	Circular RTD Sensor	1	620	620
4	Temperature Contractor (16 amp.)	1	1250	1250
5	Ammeter (0 to 15 Ampere)	1	600	600
6	MCB Hegger (16 amp., Two Poles)	1	545	545
7	Toggle Switch	3	106	318
8	Wire, Cable, Flux	-	-	500
9	Heater (250mm x250mm)	1	1150	1150
10	Glass Wool (250mm x250mm)	1	250	250
11	Manometer Tube	2	125	250
12	M.S. Rod	-	-	2350
13	Control Panel	1	1800	1800
14	Wooden Sheet	1	2200	2200
15	Blower	1	1600	1600
16	Anemometer	1	7800	7800
17	Aluminum Plate	-	-	6650
18	Aluminum Bar	-	-	5250
19	Plate Machining			1500
20	Aluminum Bar Cutting			2000
21	Plate Drilling			1250
22	M.S. Net			500
23	Fabrication Cost			3000
24	Data Unit Box	1	1800	1800
Total				46593

Table 1:Material procurement

THERMAL CONDUCTIVITY OF VARIOUS MATERIALS

Material	Thermal Conductivity (k) of Material W/m °C
Silver (pure)	410
Copper (pure)	385
Aluminum (pure)	240
Nickel (pure)	93
Iron (pure)	73
Carbon Steel, 1% C	43
Lead (pure)	35
Chrome-nickel steel	16.3

Table 2: Thermal Conductivity of Various Materials at 0°C

5.1 Material of fins -Aluminium1067

- **Physical properties**

Aluminium is a relatively soft, durable, lightweight, ductile and malleable metal with appearance ranging from silvery to dull gray, depending on the surface roughness. It is nonmagnetic and does not easily ignite. A fresh film of aluminium serves as a good reflector (approximately 92%) of visible light and an excellent reflector (as much as 98%) of medium and far infrared radiation. The yield strength of pure aluminium is 7–11 MPa, while aluminium alloys have yield strengths ranging from 200 MPa to 600 MPaAluminium has about one-third the density and stiffness of steel. It is easily machined, cast,drawn and extruded.Aluminium atoms are arranged in a face-centered cubic (fcc) structure. Aluminium has a stacking-fault energy of approximately 200 mJ/m^2.

Aluminium is a good thermal and electrical conductor, having 59% the conductivity of copper, both thermal and electrical, while having only 30% of copper's density. Aluminium is capable of being a superconductor, with a superconducting critical temperature of 1.2 Kelvin and a critical magnetic field of about 100 gauss (10 milliteslas).

- **Melting Temperature**

The melting point of aluminium is sensitive to purity, e.g. for 99.99% pure aluminium at atmospheric pressure it is 660°C but this reduces to 635°C for 99.5% commercial pure aluminium. The addition of alloying elements reduces this still further down to 500°C for some magnesium alloys under certain conditions. The melting point increases with pressure in a straight line relationship to 980°C at 50 kbar.

- **Thermal Conductivity**

The thermal conductivity, κ, of 99.99% pure aluminium is 244 W/mK for the temperature range 0-100°C which is 61.9% of the IACS, and again because of its low specific gravity its mass thermal conductivity is twice that of copper (see Figure 1501.02.05). Thermal conductivity can be calculated from electrical resistivity measurements using the formula $\kappa = 5.02\lambda T \times 10{-}9{+}0.03$, where κ is the thermal conductivity, λ is the electrical conductivity and T the absolute temperature in degrees Kelvin; this method is usually used to derive the values quoted in reference books. The thermal conductivity is reducedslightly by the addition of alloying elements. Application of the formula has been found to be largely independent of composition with the exception of silicon. The combined properties of high thermal conductivity, low weight and good formability make aluminium an obvious choice for use in heat exchangers, car radiators and cooking utensils while in the cast form it is extensively used for I/C engine cylinder heads.

Thermal Conductivity of Various Materials at 0°C is as follows:

CHAPTER 6

EXPERIMENTAL SETUP

The experimental set-up consisting of the following parts

1) Main Duct (Tunnel)

2) Heater Unit

3) Base Plate

4) Data Unit

1. Main Duct (Tunnel):

Tunnel constructed of wood of 20 mm thickness, had an internal cross-section of 250 mm width and 100 mm the total length of the channel is 1000 mm. It will be operated in force draught mode by the blower of 0.5 H.P., 0 to 13000 rpm, 220W, 1.8Kg, variable speed 1 to 6 and it is fitted at 45cm away from the entry of the tunnel positioned horizontally and flow of air is controlled by the flow control valve mounted just after the blower. It has a convergent and divergent section at both ends having the inclination of 30°. A Matrix anemometer is mounted in a tunnel to measure the mean inlet velocities of the air flow entering to the test section the range of this anemometer is 0 to 40m/sec

The Reynolds number range used in this experiment was 13,500–42,000, which is based on the hydraulic diameter of the channel over the test section (Dh) and the average velocity (U) The inlet and outlet temperature of the air stream and temperature of base plate will be measured by RTD Sensors having a range of 0°c to 450°c which mounted in wind tunnel.

Figure 1:Main Duct

2. Heater Unit:

Heater Unit (test section) has a cross-section of 250 mm x 250 mm square; the heating unit mainly consisted of an electrical heater placed between two M.S. Plate having the same dimension of base plate, a two firebrick of 250x 220 mm. Dimensions of the electrical heater placed on the firebrick are 250 mm x 250 mm. The heater output has a power of 200 W at 220V and a current of 10 amp.

Figure 2: Heater Unit

3. Base Plate:

It consist of square plate at base having the dimension 250mm x 250 mm, thickness is 6mm and The pin fins and base plate made of the same material i.e. Aluminum because of the considerations of conductivity, machinability and cost. The fins have a circular cross section of 15 mm x15 mm and are attached on the upper surface of the base plate as shown in Fig. 3. Circular pin fins with different lengths, corresponding to C/H (Clearance ratio) values of 0, 0.333 and 1, are perforated at the 17 mm from bottom tip of those by an 8 mm diameter drill bit. The pin fins are fixed uniformly on the base plate with a constant spacing between the spanwise directions of 18.125 mm, with different spacing between the pin fins in the streamwise direction. The spacing ratios of the pin fins in the streamwise direction (Sy/D) were 1.208, 1.524, 1.944 and 3.417 mm, giving different numbers of the pin fins on the base plate. It is well-known fact that if the inter-fin spacing in the spanwise direction decreases, the flow blockage will increase and thus, pressure drop along to tested heat exchanger will increase. Because the aim of the study is to determine inter-fin spacing in streamwise direction, the spacings in the spanwise direction will not be considered in this study.). The temperature of the base plate is measured RTD Sensors which can sense the temperature from 0°c to 450°c and it is screwed into groove in the base plate the readings of the RTD Sensors will be shown on data unit.

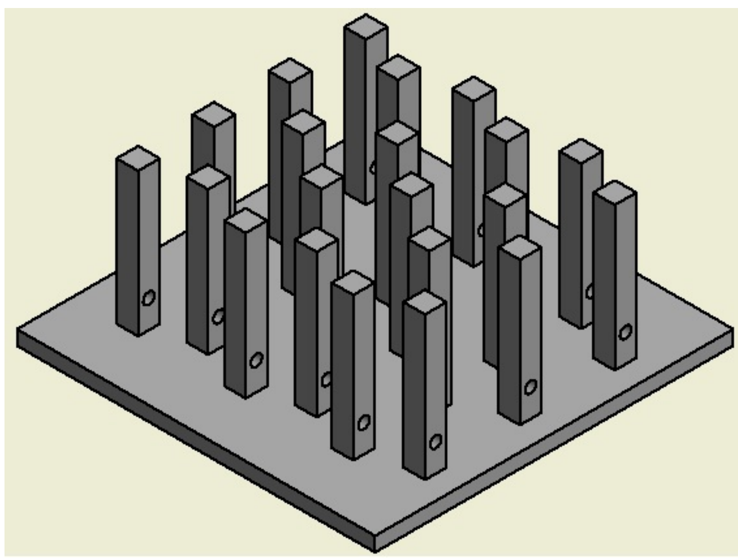


Figure 3: Base Plate having Sy/D=1.524.No. of fins 21

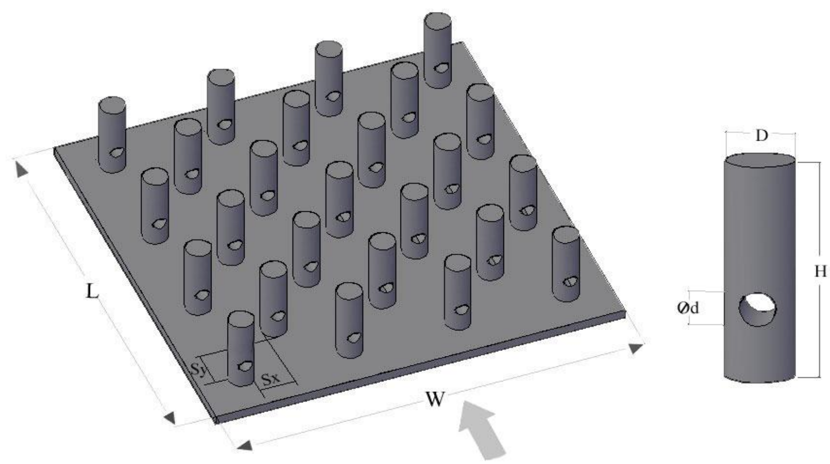


Figure 4: Base plate having Sy/D=1.208. No. of fins 25

Sr No.	Streamwise distance to diameter ratio i.e, Sy/D	No. of fins on Base plate
1	1.208	25
2	1.524	21

Table 3:Streamwise distance ratio and no. of fins on base plate

Sr No.	Clearance ratio i.e, C/H	Height of fins (mm)
1	0	100

Table 4: Clearance ratio and height of fins

4. Data Unit:

It consist of various indicating devices which indicates the reading taken by the various component like RTD sensors, Anemometer There are three Temperature indicator which shows reading taken by the RTD Sensors in the range 0°c to 450 °c among this two gives inlet and out temperature of air and one gives temperature of base plate There is one temperature contractor which can maintain the temperature of base plate it will not allow to exceed the temperature of base plate above desired values. Inlet flow rate of air is indicated by velocity indicator using Anemometer. MCB Hegger switches are mounted to cut off the power supply in case any short circuit.

Figure 5: Data Unit

(1)Inlet temp. RTD Sensor(2) Base plate temp. RTD sensor(3)Outlet temp. RTD sensor(4)Ammeter.

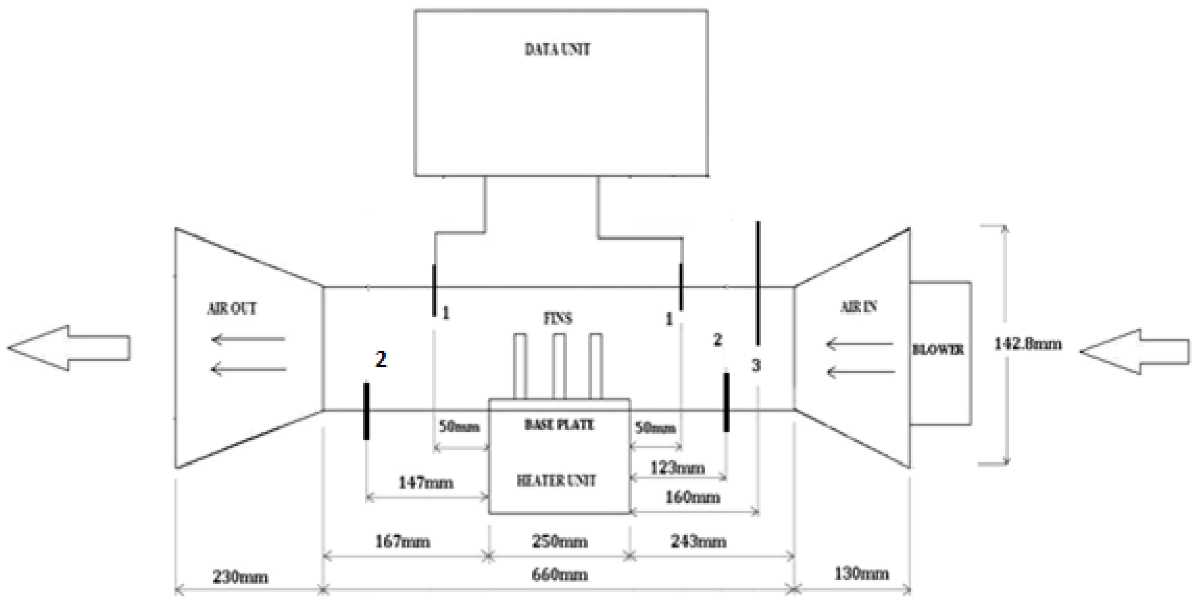


Figure 6: Experimental Set-up.

(1)RTD Sensor (2) Manometer (3) Anemometer

Figure 7: Actual Experimental Set up

CHAPTER 7

METHODOLOGY

EXPERIMENTAL PROCEDURE

- First of all make all the necessary attachment to wooden tunnel required for experimental set-up i.e, attachment of data unit, attachment of blower unit, attachment of heater unit.
- Keep the aluminum base plate of required dimensions on heater unit.
- Move the heater unit upward by rotating the screw jack.
- Switch on the RTD sensors of inlet, outlet and base plate temperature indicatorand check whether it is properly functioning or not.
- Measure the room temperature by anemometer by changing the function of it to temperature mode.
- Switch on the heater, as soon as base plate temperature reached upto 100˚C, the temperature controller of RTD sensors comes in operation and it will cut off the power supply of heater. Once the temperature of base plate reduced upto 99˚C again temperature controller of RTD sensors comes in operation and start the power supply of heater and cycle gets repeat. .
- Now switch on the blower and measure the velocity of inlet air by using digital anemometer.
- Make the inlet air velocity constant at required velocity by regulating the speed of blower i.e. 2m/s, 3m/s, 4m/s, and 5m/s.
- Now air will pass over heated base plate through tunnel
- Measure the outlet temperature of outgoing warm air.
- Now due to forced convection the temperature of base plate falls below 100˚C. As soon as the temperature of base plate falls below the 100˚C heater unit will start heating the base plate to achieve the constant 100˚C temperature by supplying constant electrical input as the air is continuously flowing over the base plate heat get transferred from it to the flowing air. Thus the temperature of base plate falls continuously and take the temperature readings of base plate after 90 seconds of air flow. The duration of air flow is constant for all types of base plate mentioned above.
- Similarly, repeat the same procedure for velocity 3m/s, 4m/s, and 5m/s and take the similar readings.

CHAPTER 8
EXPERIMENTATION

1. EXPERIMENTAL DATA COLLECTION

8.1) 21 STAGGERED CYLINDRICAL FINS

SR.NO.	Inlet Temperature °C	Outlet Temperature °C	Base Plate Temperature °C	Air Velocity m/s	Current Amp
1	35	56	100	2	8.25
2	35	58	100	3	8.25
3	35	61	100	4	8.25
4	35	64	100	5	8.25

TABLE 5:21 STAGGERED CYLINDRICAL FINS

8.2) 25STAGGERED CYLINDRICAL FINS

SR.NO.	Inlet Temperature °C	Outlet Temperature °C	Base Plate Temperature °C	Air Velocity m/s	Current Amp
1	35	64	100	2	8.25

2	35	65	100	3	8.25
3	35	67	100	4	8.25
4	35	70	100	5	8.25

TABLE 6: 25STAGGERED CYLINDRICAL FINS

8.3) 21 STAGGERED SQUARE FINS

SR.NO.	Inlet Temperature °C	Outlet Temperature °C	Base Plate Temperature °C	Air Velocity m/s	Current Amp
1	35	53	100	2	8.25
2	35	54	100	3	8.25
3	35	55	100	4	8.25
4	35	57	100	5	8.25

TABLE 7:21 STAGGERED SQUARE FINS

8.4) 25 STAGGERED SQUARE FINS

SR.NO.	Inlet Temperature °C	Outlet Temperature °C	Base Plate Temperature °C	Air Velocity m/s	Current Amp
1	35	60	100	2	8.25
2	35	61	100	3	8.25
3	35	63	100	4	8.25

4	35	69	100	5	8.25

TABLE 8: 25 STAGGERED SQUARE FINS

CHAPTER 9

ANALYSING& CALCULATION

ANALYSING AND INVESTIGATION [8]

a) Heat transfer and friction factor.

The convective heat transfer rate Q convection from electrically heated test surface is calculated by using

$$Q_{conv.} = Q_{elect.} - Q_{cond.} - Q_{rad.} \quad (1)$$

Where: Q indicates the heat transfer rate in which subscripts conv, elect, cond and rad denotes convection, electrical, conduction and radiation, respectively. The electrical heat input is calculated from the electrical potential and current supplied to the surface.

$$Q_{elect.} = I^2 \times R$$

Where:-*'I'* is current flowing through the heater and *'R'* is the resistance

In similar studies, investigators reported that total radioactive heat loss from a similar test surface would be about 0.5% of the total electrical heat input. The conductive heat losses through the sidewalls can be neglected in comparison to those through the bottom surface of the test section. Using these findings, together with the fact that the two sides walls and the top wall of the test section were well insulated and readings of the thermocouple placed at the outer surface temperature of the heating section were nearly equal to ambient temperature, one could assume with some confidence that the last two terms of Eq. (1) may be ignored.

The heat transfer from the test section by convection can be expressed as

$$Q_{conv.} = h_{av} A_s \left[T_s - \left(\frac{T_{out} + T_{in}}{2} \right) \right] \quad (2)$$

Hence, the average convective heat transfer coefficient have could be deduced via

$$h_{av} = \frac{Q_{conv.}}{A_s\left[T_s - \left(\frac{T_{out}+T_{in}}{2}\right)\right]} \quad (3)$$

Either the projected or the total area of the test surface can be taken as the heat transfer surface area in the calculations. The total area is equal to the sum of the projected area and surface area contribution from the pin fins. These two areas can be related to each other by

Total area = Projected area + Total surface area contribution from the blocks

$$A_s = WL + [\pi DH - 2\pi ab]N_p + [(2\pi r^2 + 2\pi rD) - 2\pi ab]N_p \quad (4)$$

Where:- *W* is the width of the base plate, *L* its length, *Np* is the number of fins, *H* the height of fin and *D* is the diameter of the fins, *a* is semi major axis of ellipse, *b* is semi minor axis of ellipse. The dimensionless groups are calculated as follows:

$$N_u = \frac{h_{av}D_h}{k} \quad (5)$$

Nu based on the projected area will reflect the effect of the variation in the surface area as well as that of the disturbances in the flow due to fins on the heat transfer. But Nu based on the total area will reflect the effect of the flow disturbances only. In this study, heat transfer enhancement characteristics were determined by using Nu-based projected area, while optimization was made by using Nu based total area.

$$F = \frac{\Delta P}{\left[\left(\frac{L_t}{D_h}\right)\left(\rho\frac{U^2}{2}\right)\right]} \quad (6)$$

$$R_e = \frac{D_h U}{\nu} \quad (7)$$

In all calculations, the values of thermo physical properties of air were obtained at the bulk mean temperature, which is

$$T_m = (T_{in} + T_{out})/2$$

b) **Heat transfer.**

In order to have a basis for the evaluation of the effects of the fins, some experiments were carried out without any fins attached to the plate. Using the data obtained from these test, the average Nusselt number (Nus) for the smooth surface (without pin fins) was correlated as function of Re and Pr as follows:

$$\boldsymbol{Nu_s = 0.077Re^{0.716}Pr^{1/3}} \quad \textbf{(8)}$$

The Nusselt number based on both the projected area and total area was related to the Reynolds number, clearance ratio (C/H), inter-fin distance ratio (Sy/D) and Prandtl number. Thus, the following correlation equations were obtained:

$$\boldsymbol{Nu_p = 45.99Re^{0.396}\left(1 + {C}/{H}\right)^{-0.608}\left({S_y}/{D}\right)^{-0.522} Pr^{1/3}} \quad \textbf{(9)}$$

This equations are valid for the experimental conditions of 13,500 ≤ Re ≥ 6 42,000, 1.208 < Sy/D < 3.417, 0 ≤ C/H ≤ 1 and Pr = 0.7, respectively. If the Nusselt number for the surface with fins calculated on the basis of the projected area of the test surface is normalized by Nus, it will reflect the total effect of the pin fins (surface enlargement as well as turbulence) on the heat transfer enhancement. Nu/Nus based on the projected area, as a function of the duct Reynolds number for the three different pin heights, namely C/H = 1, 0.333 and 0 at Sy/D = 1.208. The Nusselt number that is based on the projected area will reflect the effect of the variation in the surface area as well as that of the disturbance in the flow due to pin fins on the heat transfer. Longer fins can also increase the turbulence of the flow in the channel, resulting in an increase in the heat transfer.

c) Friction factor.

The pressure drops in the smooth channel was found to be so small that they could not be measured by the pressure transducer. This resulted from smaller length of the test section and smaller roughness of the duct. The experimental pressure drops over the test section in the finned duct were measured under the heated flow conditions. These measurements were converted to the friction factor, f, using Eq. (6). Using the experimental results, f was correlated as a function of the duct Reynolds number, Re, and geometrical parameters. The resulting equation is

$$\boldsymbol{F = 2.4Re^{-0.0836}(1 + C/H)^{-0.0836}\left(S_y/D\right)^{-0.0814}} \quad (\mathbf{10})$$

This equation is valid for 13,500 < Re < 42,000, 1.208 <Sy/ D < 3.417, 0 < C/H < 1 with a correlation coefficient of r = 0.980.On the other hand, as the resistance to the flow will be smaller due to the perforations, friction factor is lower for the perforated fins than the solid fins.

d) Enhancement efficiency.

For a constant pumping power, it is useful to determine the effectiveness of the heat transfer enhancement of a heat transfer promoter in comparison with a smooth surface.
The enhancement efficiency of the heat transfer technique for a constant pumping power can be expressed as

$$\boldsymbol{\eta = \frac{ha}{hs}} \quad \mathbf{(11)}$$

where ha and hs are the convective heat transfer coefficient with and without pin fins, respectively, and the index p denotes the pumping power. Employing (08), (11), and (12), the following equation can be written for the heat transfer efficiency for the pin fins according to total heat transfer surface area:

$$\varepsilon = \frac{ha}{hs} = 51.09\, Re_a^{-0.358} \left(1 + {}^{C}/_{H}\right)^{0.1028} \left({}^{S_y}/_{D}\right)^{0.0812} \quad (12)$$

e) **Hydraulic Diameter.**

For shapes such as squares, rectangular or annular ducts where the height and width are comparable, the characteristical dimension for internal flow situations is taken to be the hydraulic diameter, ***Dh***, defined as:

$$Dh = \frac{4\,A}{P}$$

$$= \frac{4(250 \times 100)}{2(250 + 100)}$$

$$Dh = 142.86mm$$

Temperature (°C)	**Kinematic Viscosity. (10^{-4} ft^2/s)**
40	1.46
42	1.47
44	1.48
46	1.49
48	1.50
50	1.51
52	1.53
54	1.54
56	1.55
58	1.56
60	1.58
62	1.59
64	1.60
66	1.61
68	1.62
70	1.63

TABLE 9:VALUES OF KINEMATIC VISCOSITY AT VARIOUS TEMPERATURES [10]

CALCULATIONS

10.1 Cylindrical Fins:

For Perforated Fins Sy/D=1.524 i.e. No. of Fins on one base plate=21 No.

For observation Table [A]

For Perforated Fins, C/H=0, i.e. Fin height=100mm

The convective heat transfer rate Q convection from electrically heated test surface is calculated by using

$$\boldsymbol{Q_{conv.}} = \boldsymbol{Q_{elect.}} - \boldsymbol{Q_{cond.}} - \boldsymbol{Q_{rad.}}$$

Where: Q indicates the heat transfer rate in which subscripts conv, elect, cond and rad denotes convection, electrical, conduction and radiation, respectively. The electrical heat input is calculated from the electrical potential and current supplied to the surface.

$\boldsymbol{Q_{elect.}} = \boldsymbol{I^2} \times \boldsymbol{R}$

Given data, I=8.25 amp and V=230V

$$R=\frac{V}{I}=\frac{230}{8.25}=27.66\Omega$$

Therefore,

$$Q_{elect.}=I^2\times R=8.25^2 \text{ x } 26.66=1815 \text{ W}$$

$\boldsymbol{Q_{conv.}} = \boldsymbol{Q_{elect.}}=1815 \text{ W}$

$$\boldsymbol{h_{av}} = \frac{\boldsymbol{Q_{conv.}}}{\boldsymbol{A_s}\left[\boldsymbol{T_s} - \left(\frac{\boldsymbol{T_{out}} + \boldsymbol{T_{in}}}{\boldsymbol{2}}\right)\right]}$$

Total area (As) = Projected area + Total surface area contribution from the blocks

Total area (As)for perforated fins.

$$\boldsymbol{A_{s(perforated)} = WL + [\pi DH - 2\pi ab]N_p + [(2\pi r^2 + 2\pi rD) - 2\pi ab]N_p}$$

$$A_s=250x250+[3.14x15x100-2x3.14x4x4.8]21+\left[\left(2x3.14x4^2+2x3.14x4x15\right)-2x3.14x4x4.8\right]21$$

As = 0.16692m^2

Total area As for smooth surface

$$\boldsymbol{A_{s(smooth)} = W \times L}$$

$$\mathrm{A_s} = 250 \times 250$$

$$\mathrm{A_s} = 62500mm^2$$

For Velocity V=2 $^{m}/_{s}$

$$\text{Heat Transfer Coefficient } h_{av1}=\frac{1815}{0.18621\left[100-\left(\frac{35+56}{2}\right)\right]}$$

$$h_{av1}= 199.51 \text{ w/m2°c}$$

Reynolds No.

$$\boldsymbol{R_e = \frac{D_h U}{\nu}}$$

$$R_e=\frac{0.14286x2}{1.44x10^{-5}}$$

$$R_e=19731$$

Nu_s For Smooth Duct(Without Fin)

$$\boldsymbol{Nu_{s2} = 0.077 Re^{0.716} Pr^{1/3}}$$

$$\mathrm{Nu}_{s2} = 72.273$$

Nussult no. based on projected area

$$\boldsymbol{Nu_{p2} = 45.99 Re^{0.396} \left(1 + {}^{C}/_{H}\right)^{-0.608} \left({}^{S_y}/_{D}\right)^{-0.522} Pr^{1/3}}$$

$$\mathrm{Nu}_{p2} = 45.99(16735.415)^{0.396}(1+0)^{-0.608}(1.208)^{-0.522}0.7^{1/3}$$

$\mathrm{Nu}_{p2} = 1541.95$

This equations are valid for the experimental conditions of 13,500 ≤ Re ≥ 6 42,000, 1.208 <Sy/D < 3.417, 0 ≤ C/H ≤ 1 and Pr = 0.7 by using this equation the Nu/Nus and Re will be determine for perforated fins for different C/H ratio i.e. C/H=0, C/H=0.333, C/H=1 and for different Sy/D ratios i.e. Sy/D=1.208, Sy/D=1.524, Sy/D=1.944, Sy/D=3.417 The same will be find out for solid fins and the comparative graph between Nu/Nus and Re for perforated fins and solid fins

$$\frac{\boldsymbol{Nu_{p2}}}{\boldsymbol{Nu_{s2}}} = 21.97$$

Friction Factor:

The pressure drops in the tunnel without fins is so small that they could not be measured by the Manometer. This resulted from smaller length of the test section and smaller roughness of the duct. The experimental pressure drops over the test section in the finned duct will be measured under the heated flow conditions. These measurements will be converted to the friction factor 'F' Using the experimental results, f was correlated as a function of the duct Reynolds number, Re, and geometrical parameters. The resulting equation is

$$\boldsymbol{F_2 = 2.4 Re^{-0.0836}(1 + C/H)^{-0.805}\left(S_y/D\right)^{-0.0814}}$$

$$F_2 = 2.4\mathrm{x}16735.415^{-0.0836}(1+0)^{-0.0836}(1.208)^{-0.0814}$$

$$F_2=1.014$$

Enhancement Efficiency:

$$\varepsilon = \frac{ha}{hs} = 51.09\, Re_a^{-0.358} \left(1 + {}^{C}/_{H}\right)^{0.1028} \left({}^{S_y}/_{D}\right)^{0.0812}$$

$$\varepsilon = 1.532$$

Similarly, we have calculated all above parameters for velocities 3, 4, and 5 m/s.

10.2 For Perforated Fins Sy/D=1.208 i.e. No. of Fins on one base plate=25 No.

For observation Table [B]

$$Q_{conv.} = h_{av} A_s \left[T_s - \left(\frac{T_{out} + T_{in}}{2} \right) \right]$$

Hence, the average convective heat transfer coefficient have could be deduced via

$$h_{av} = \frac{Q_{conv.}}{A_s \left[T_s - \left(\frac{T_{out} + T_{in}}{2} \right) \right]}$$

Total area (As) = Projected area + Total surface area contribution from the blocks
Total area (As)for perforated fins.

$$A_{s(perforated)} = WL + [\pi DH - 2\pi ab]N_p + [(2\pi r^2 + 2\pi rD) - 2\pi ab]N_p$$

$$A_s = 250x250 + [3.14x15x100 - 2x3.14x4x4.8]25 + \left[\left(2x3.14x4^2 + 2x3.14x4x15\right) - 2x3.14x4x4.8\right]25$$

$As = 0.18621 m^2$

Total area As for smooth surface

$$A_{s(smooth)} = W \times L$$

$$A_s = 250 \times 250$$

$$A_s = 62500 mm^2$$

For Velocity V=2 $^{m}/_{s}$

$$\text{Heat Transfer Coefficient } h_{av1} = \frac{1815}{0.18621 \left[100 - \left(\frac{35+64}{2}\right)\right]}$$

$h_{av1} = 193.104$ w/m2°c

Reynolds No.

$$R_e = \frac{D_h U}{\nu}$$

$$R_e = \frac{0.14286x2}{1.49x10^{-5}}$$

$$R_e = 19058$$

Nu_s For Smooth Duct(Without Fin)

$$Nu_{s2} = 0.077 Re^{0.716} Pr^{1/3}$$

$$Nu_{s2} = 66.909$$

Nussult no. based on projected area

$$Nu_{p2} = 45.99 Re^{0.396} \left(1 + {C}/{H}\right)^{-0.608} \left({S_y}/{D}\right)^{-0.522} Pr^{1/3}$$

$$Nu_{p2} = 45.99(16836.77)^{0.396}(1+0)^{-0.608}(1.208)^{-0.522} 0.7^{1/3}$$

$Nu_{p2} = 1668.113$

$$\frac{Nu_{p2}}{Nu_{s2}} = 24.30$$

Friction Factor:

$$F_2 = 2.4 Re^{-0.0836} (1 + C/H)^{-0.805} \left(S_y/D\right)^{-0.0814}$$

$$F_2 = 2.4x16967.75^{-0.0836}(1+0)^{-0.0836}(1.208)^{-0.0814}$$

$$F_2 = 1.036$$

Enhancement Efficiency:

$$\varepsilon = \frac{ha}{hs} = 51.09\, Re_a^{-0.358} \left(1 + C/_H\right)^{0.1028} \left(S_y/_D\right)^{0.0812}$$

$\varepsilon = 1.522$

Square Cross section:

10.3 For Perforated Fins Sy/D=1.524 i.e. No. of Fins on one base plate=21 No.

For observation Table [C]

$$h_{av} = \frac{Q_{conv.}}{A_s\left[T_s - \left(\frac{T_{out} + T_{in}}{2}\right)\right]}$$

Total area (As) = Projected area + Total surface area contribution from the blocks

Total area (As)for perforated fins.

$$A_{s(perforated)} = WL + 4NHD + \pi N_p \,[\mathrm{dD} - 0.5d^2]$$

$$A_s = 250\text{x}250 + [4\text{x}21\text{x}0.1\text{x}0.015] + (\pi\text{x}21)\left[(0.008\text{x}0.015) - \left(0.015\text{x}0.008^2\right)\right]$$

$As = 0.194 m^2$

Total area As for smooth surface

$$A_{s(smooth)} = W \times L$$

$$A_s = 250 \times 250$$

$$A_s = 62500 mm^2$$

For Velocity V=2 $^{m}/_{s}$

$$\text{Heat Transfer Coefficient } h_{av1} = \frac{1815}{0.18621\left[100\text{-}\left(\frac{35+53}{2}\right)\right]}$$

h_{av1}= 167.06 w/m2°c

Reynolds No.

$$R_e = \frac{D_h U}{\nu}$$

$$R_e = \frac{0.14286 \times 2}{1.42 \times 10^{-5}}$$

$$R_e = 20022$$

Nu_s For Smooth Duct(Without Fin)

$$Nu_{s2} = 0.077 Re^{0.716} Pr^{1/3}$$

$$Nu_{s2} = 56.226$$

Nussult no. based on projected area

$$Nu_{p2} = 45.99 Re^{0.396} \left(1 + C/_{H}\right)^{-0.608} \left(S_y/_{D}\right)^{-0.522} Pr^{1/3}$$

$$Nu_{p2} = 45.99(16735.415)^{0.396}(1+0)^{-0.608}(1.208)^{-0.522} 0.7^{1/3}$$

$Nu_{p2} = 1342.03$

$$\frac{Nu_{p2}}{Nu_{s2}} = 22.09$$

Friction Factor:

$$F_2 = 2.4 Re^{-0.0836}(1 + C/H)^{-0.805}\left(S_y/D\right)^{-0.0814}$$

$$F_2 = 2.4 \times 16735.415^{-0.0836}(1+0)^{-0.0836}(1.208)^{-0.0814}$$

$$F_2 = 1.013$$

Enhancement Efficiency:

$$\varepsilon = \frac{ha}{hs} = 51.09\, Re_a^{-0.358} \left(1 + {}^{C}/_{H}\right)^{0.1028} \left({}^{S_y}/_{D}\right)^{0.0812}$$

$$\varepsilon = 1.524$$

10.4 For Perforated Fins Sy/D=1.208 i.e. No. of Fins on one base plate=25 No.

For observation Table [D]

$$h_{av} = \frac{Q_{conv.}}{A_s\left[T_s - \left(\frac{T_{out} + T_{in}}{2}\right)\right]}$$

Total area (As) = Projected area + Total surface area contribution from the blocks

Total area (As)for perforated fins.

$$A_{s(perforated)} = WL + 4NHD + \pi N_p\ [dD - 0.5d^2]$$

$$A_s = 250x250 + [4x21x0.1x0.015] + (\pi x21)\left[(0.008x0.015) - \left(0.015x0.008^2\right)\right]$$

As = 0.219m^2

Total area As for smooth surface

$$A_{s(smooth)} = W \times L$$

$$A_s = 250 \times 250$$

$$A_s = 62500mm^2$$

For Velocity V=2 $^{m}/_{s}$

$$\text{Heat Transfer Coefficient } h_{av1} = \frac{1815}{0.18621\left[100-\left(\frac{35+60}{2}\right)\right]}$$

$$h_{av1} = 157.86 \text{ w/m2°c}$$

Reynolds No.

$$\boldsymbol{R_e = \frac{D_h U}{\nu}}$$

$$R_e = \frac{0.14286x2}{1.47x10^{-5}}$$

$$R_e = 19344.29$$

Nu_s For Smooth Duct(Without Fin)

$$\boldsymbol{Nu_{s2} = 0.077 Re^{0.716} Pr^{1/3}}$$

$$Nu_{s2} = 66.90$$

Nussult no. based on projected area

$$\boldsymbol{Nu_{p2} = 45.99 Re^{0.396} (1 + {C}/{H})^{-0.608} \left({S_y}/{D}\right)^{-0.522} Pr^{1/3}}$$

$$Nu_{p2} = 45.99(16735.415)^{0.396}(1+0)^{-0.608}(1.208)^{-0.522}0.7^{1/3}$$

$Nu_{p2} = 1668.11$

$$\boldsymbol{\frac{Nu_{p2}}{Nu_{s2}}} = 24.30$$

Friction Factor:

$$\boldsymbol{F_2 = 2.4 Re^{-0.0836} (1 + C/H)^{-0.805} (S_y/D)^{-0.0814}}$$

$$F_2 = 2.4x16735.415^{-0.0836}(1+0)^{-0.0836}(1.208)^{-0.0814}$$

$$F_2 = 1.035$$

Enhancement Efficiency:

$$\varepsilon = \frac{ha}{hs} = 51.09\, Re_a^{-0.358} \left(1 + {}^{C}/_{H}\right)^{0.1028} \left({}^{S_y}/_{D}\right)^{0.0812}$$

$$\varepsilon = 1.515$$

Similarly, we have calculated all above parameters for velocities 3, 4, and 5 m/s.

CHAPTER 11

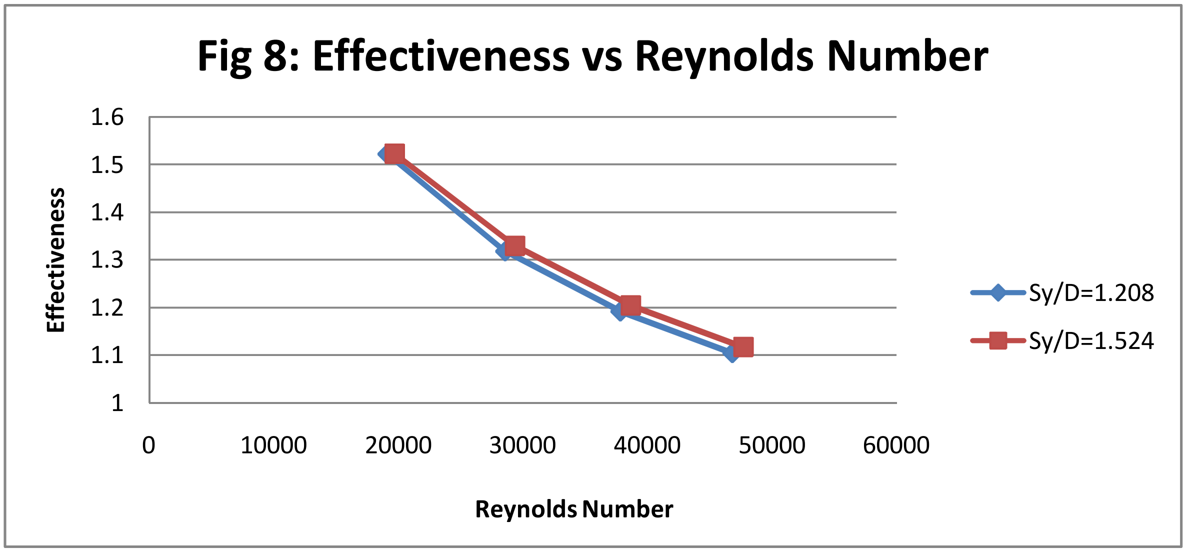
RESULTS

RESULTS AND DISCUSSION

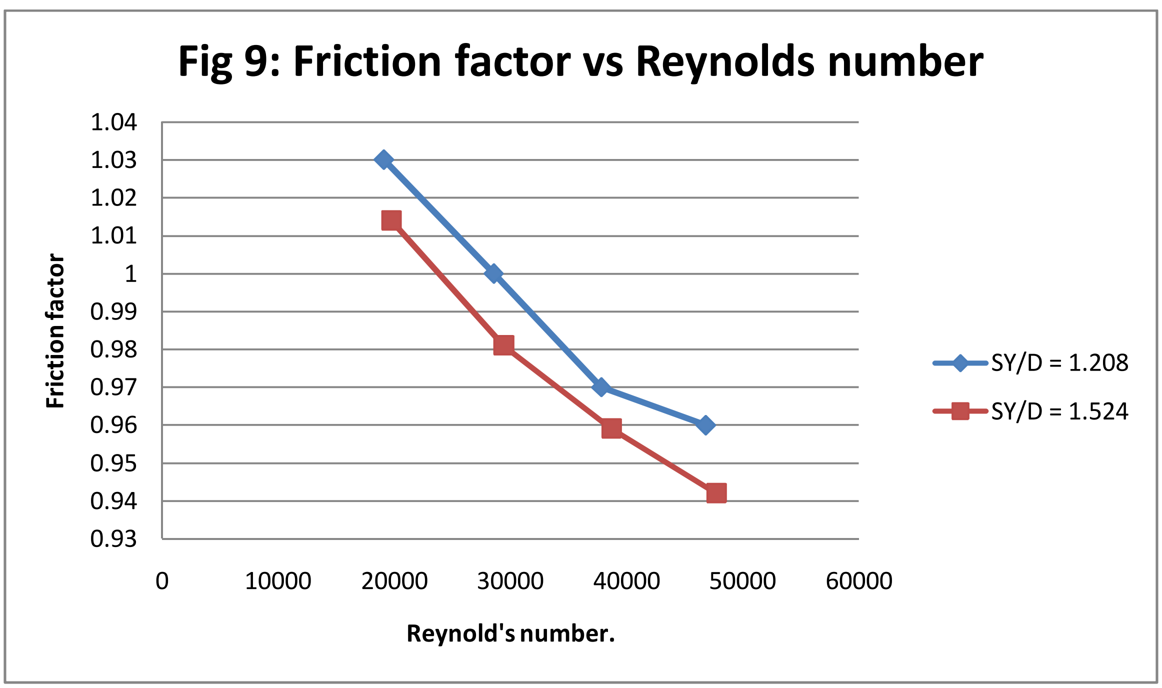
	Re_1	Re_2	Re_3	Re_4	F_1	F_2	F_3	F_4
Square 25	19344	28918	38299	47085	1.035	1.001	0.978	0.961
Square 21	20022	29824	39655	49363	1.013	0.980	0.956	0.939
Cylindrical 25	19058	28533	37792	46777	1.036	1.002	0.979	0.961
Cylindrical 21	19731	29394	38662	47714	1.014	0.981	0.959	0.942

	Nu_p/Nu_s (1)	Nu_p/Nu_s (2)	Nu_p/Nu_s (3)	Nu_p/Nu_s (4)	ε (1)	ε (2)	ε (3)	ε (4)
Square 25	24.30	21.30	19.84	17.98	1.515	1.311	1.186	1.101
Square 21	22.09	19.35	17.82	16.91	1.524	1.322	1.194	1.103
Cylindrical 25	24.30	21.30	19.94	17.98	1.522	1.318	1.192	1.104
Cylindrical 21	21.97	19.24	17.69	16.69	1.532	1.329	1.204	1.117

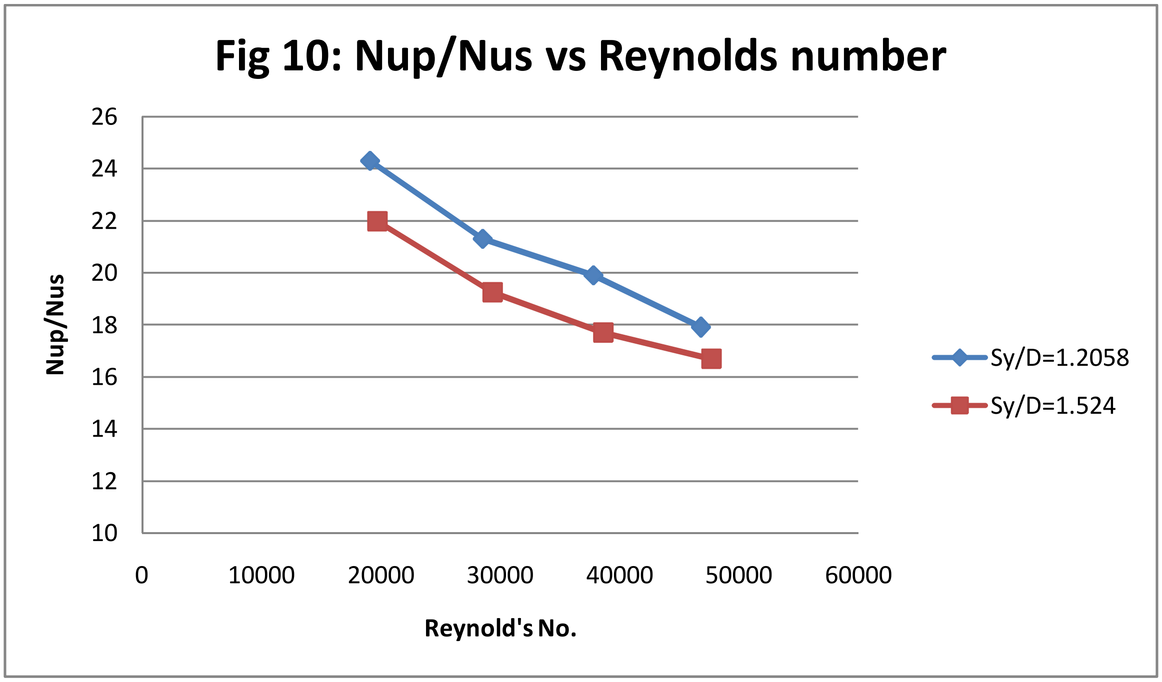
FORCYLINDRICAL CROSS-SECTION



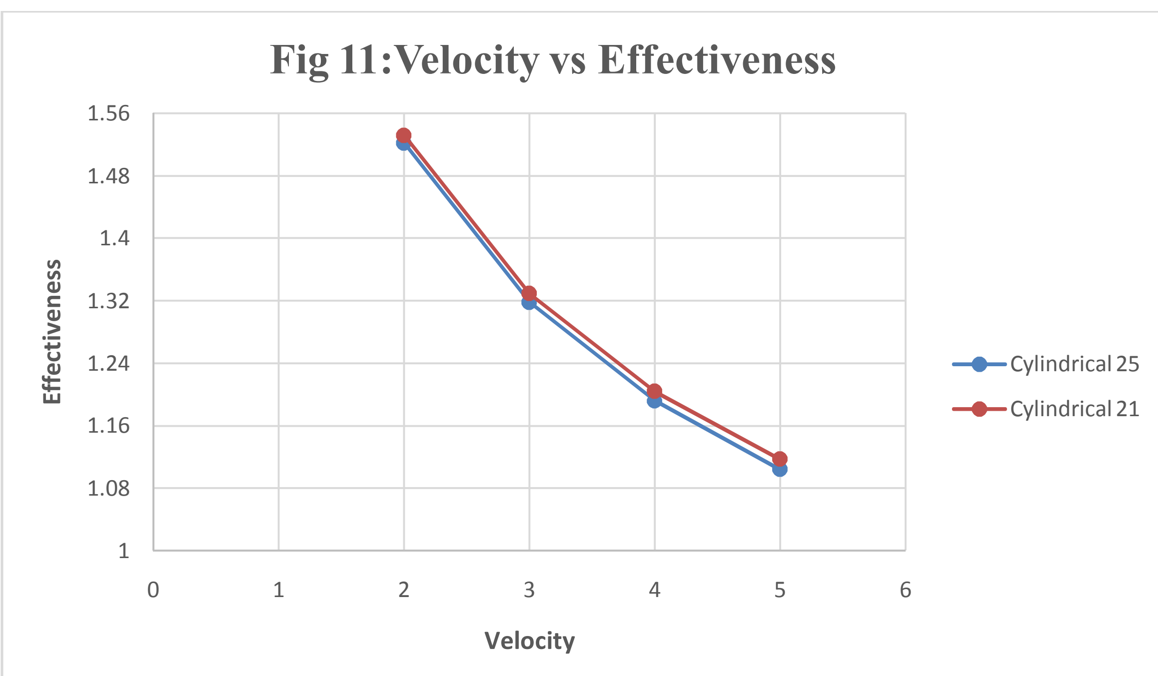
As reynold's no. increases effectiveness of fin decreases, for interspacing Sy/D=1.208 drop in effectiveness is more compared to fin of interspacing Sy/D=1.524



Friction factor reduces with increase in Reynold's number. For Sy/D = 1.524 the reduction in friction factor comparative to Sy/D = 1.208

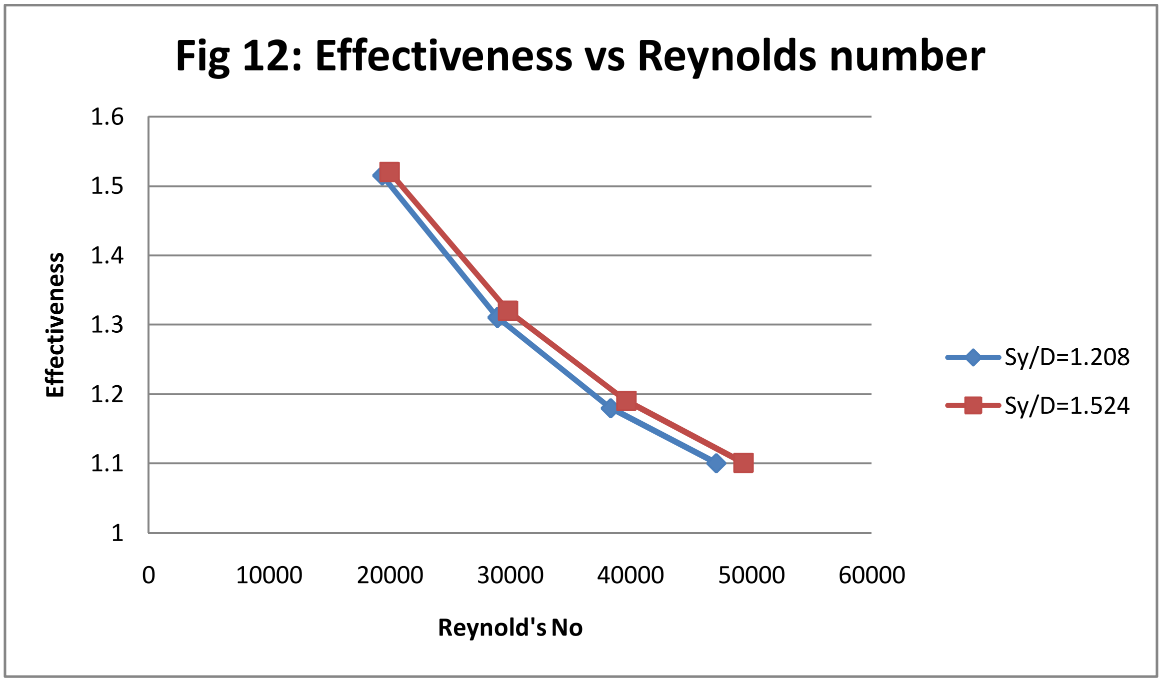


Nup/Nus of Sy/D 1.524 reduces to a greater extent as compared to Sy/D 1.208 corresponding to Reynold's number.

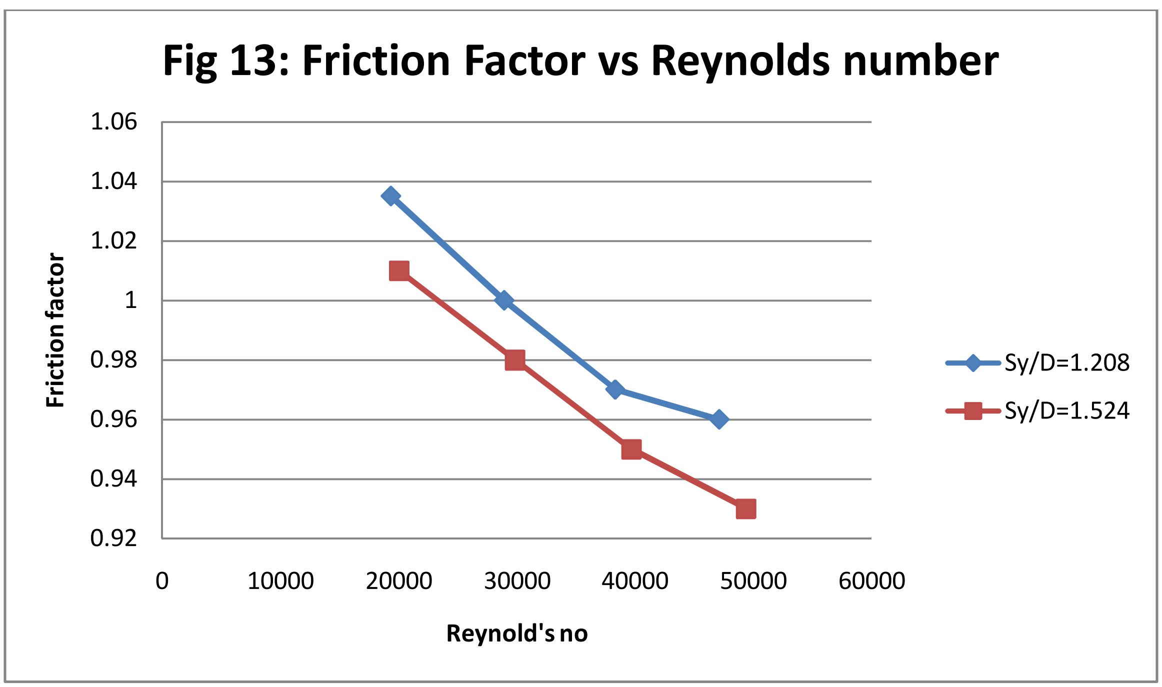


The graph shows that with increase in inlet air velocity the effectiveness goes on decreasing.

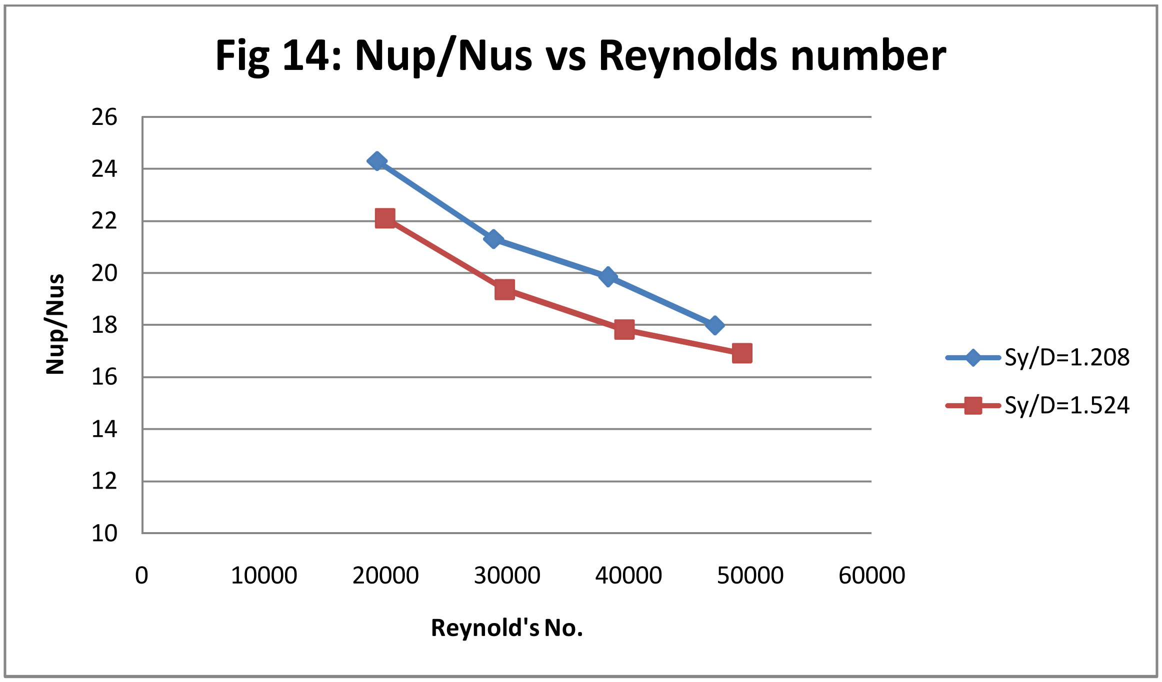
FOR SQUARE CROSS-SECTION



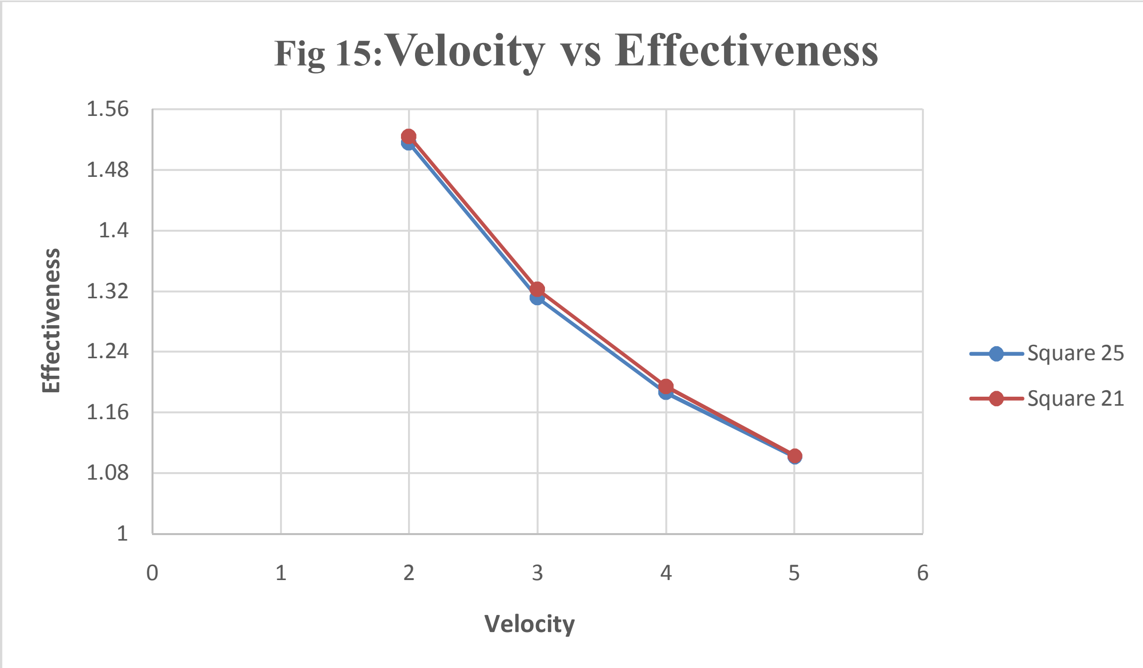
Effectiveness decreases with increase in reynold's number. For Sy/D 1.208 effectiveness is less as compared to Sy/D 1.524.



Friction factor reduces as Reynold's number increases for Sy/D 1.524 friction factor is more as compared to Sy/D 1.208.



With increase in Reynold's number, Nup/Nus decreases, Sy/D = 1.524 shows drop Nup/Nus to a large extent as compared to Sy/D = 1.208.



The graph shows that with increase in inlet air velocity the effectiveness goes on decreasing.

CHAPTER 12

CONCLUSION

CONCLUSION

In this study, the overall heat transfer, friction factor and the effect of the various design parameters on the heat transfer and friction factor for the heat experiment equipped with cylindrical cross-sectional perforated pin fins were investigated experimentally. The effects of the flow and geometrical parameters on the heat transfer and friction characteristics were determined, and the enhancement efficiency correlations have been obtained. The conclusions are summarized as:

(a) The average Nusselt number calculated on the basis of projected area increased with decreasing inter-fin spacing ratio.

(b) The friction factor increased with decreasing inter-fin distance ratio.

(c) Enhancement efficiencies increased with decreasing Reynolds number. Therefore, relatively lower Reynolds number led to an improvement in the heat transfer performance.

(d) The most important parameters affecting the heat transfer are the Reynolds number, fin spaces (pitch) and fin height. Heat transfer can be successfully improved by controlling these parameters.

(e) Lower velocity of air and increase in the inter-fin spacing has led to an increase in the effectiveness of fins

CHAPTER 13

FUTURE SCOPE

FUTURE SCOPE

The project can be carried further to further find out methods to increase the effectiveness.

1) The clearance ratio (C/H) can be varied and their effects can be studied.
2) The shape of the fins other than square and cylindrical shapes can be selected like triangular, elliptical, etc.
3) The shape of the perforated can be varied to increase the surface area of the perforation which will increase the rate of heat transfer.
4) Compound fins can also be used i.e square and cylindrical shapes can be integrated into one fin and their heat transfer can be observed.

CHAPTER 14

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REFERENCES

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