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Performance Analysis of Annular Fin Arrays with Forced Convection

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***Abstract* - Fins or extended surfaces are used whenever the available surface is found inadequate to transfer required heat with available temperature drop. These fins are not only chosen for their thermal performance, but also for other parameters such as weight, cost, reliability, etc. which depends upon application. The present paper mainly focuses on comparative analysis of plain annular fin arrays and perforated annular fin arrays to investigate heat transfer characteristics with forced convection. The data used in this performance analysis were obtained experimentally for annular fin arrays with and without perforation for aluminum. It is found that there is considerable change in heat transfer coefficient by 6.3 to 9.6% in case of perforated fin arrays which is a significant amount of increase in heat transfer rate.**
***Keywords*: Perforated Fin Arrays, Convective Heat Transfer Coefficient, Nusselt Number, Forced Convection**

I. INTRODUCTION

In last four decades, the heat transfer enhancement techniques have been implemented tremendously as the size of the heat exchanger is continuously decreasing day by day. Any heat exchanger e.g. condenser, evaporator, etc. is potential equipment for enhanced heat transfer. Therefore, it is necessary to enhance heat transfer characteristics from existing heat exchanger by several methods. These methods can be categorized as Active technique and Passive technique. In the active technique, application of external force (power) is used to enhance the heat transfer characteristics, such as surface vibration, acoustic or electric fields, etc. On the contrary, the passive techniques require the application of specific surface geometries with surface augmentation. To enhance heat transfer between a surface and its adjacent fluid, radial or longitudinal fin are in common use. In order to minimize the size of heat exchangers, the perforated fins are used on gas side to increase the surface area and heat transfer rate between the perforated fins and its surroundings. In case of gas-to-liquid heat exchangers, the heat transfer coefficient on the liquid side is generally higher than the liquid side.

II. THEORY

The convective heat transfer between a solid surface and its surrounding contact fluid is governed by Newton's law of cooling which states that the convection heat transfer is directly proportional to the area of the contact surface and the temperature difference between hot fluid and cold fluid. Therefore, the convective heat transfer can be increased by following ways.

1. Increasing the area of contact between the surface and fluid.
2. Increasing the temperature difference between the surface and fluid.
3. Increasing the convective heat transfer coefficient by fluid flow or change in surface area of fin etc.

In most of the case, it is not feasible to control the temperature difference due to some unavoidable problems and hence to increase heat transfer coefficient, it may require to install a pump or fan or blower which adds extra cost. There is an effective and alternative method to increase the convective heat transfer by increasing surface area or modifying the geometry of existing fins.

Fins are generally classified as
1. Longitudinal fins
2. Radial fins
3. Pin fins

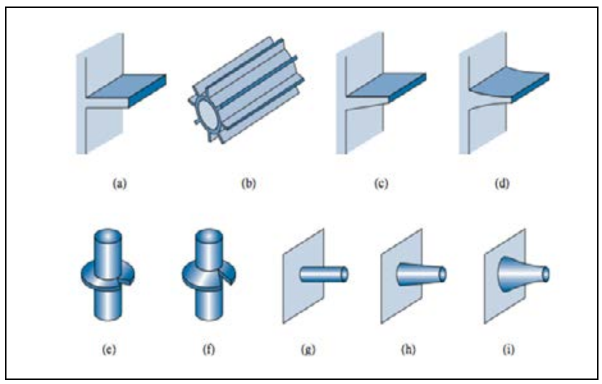

Fig. 1 Layout of Test Section

III. EXPERIMENTAL SETUP

An experimental setup consists of a heater, blower, and three test sections. The blower passes the forced air over the fins in the test section. The fins are mounted on the tube with hot fluid passing through it. This fluid is heated by the heater. By every test, the section has different fin pattern arrays to be tested. Every test section is having the different type of fin that is simple annular fin, serrated fin, perforated fin (annular fins perforation). The selection of appropriate materials to conduct and transfer heat at the better rate and efficiently is very important criteria in heat exchanger design. In this experiment, the Copper material is selected for the inner tube as it is thermally efficient and durable and the same material is used for fins.

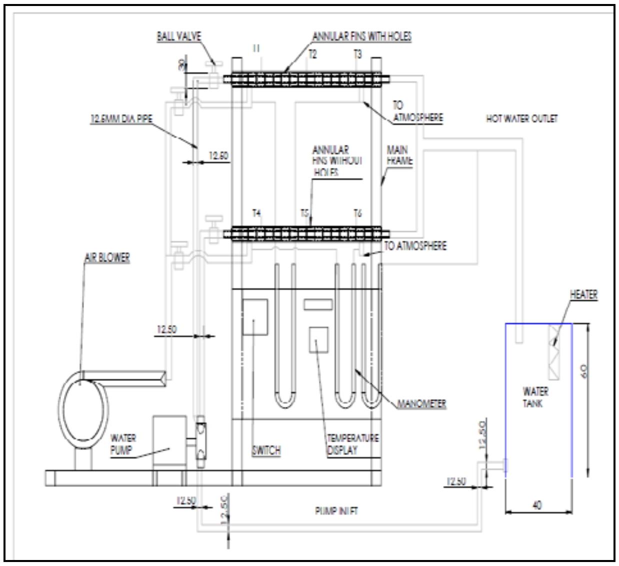

Fig. 2 Block diagram of the experimental setup (Not to scale)

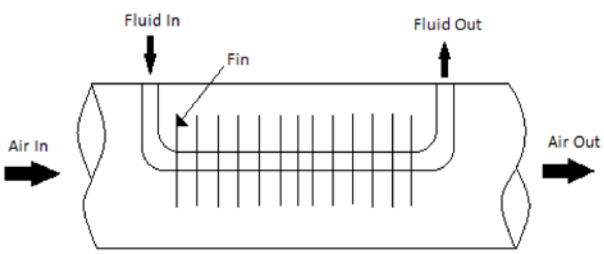

Fig. 3 Layout of Test Section

A. Dimension of Test Fins

1. Plane Annular fin :
   Outer Diameter of fin = 25mm
   Inner Diameter of fin = 16mm
   Number of fins= 20
   Thickness of fin =1mm
2. Perforated Annular fin :
   Outer Diameter of fin = 25mm
   Inner Diameter of fin = 16mm
   Thickness of fin = 1mm
   Number of holes = 4
   Diameter of holes= 2mm
   Number of fins= 20
   Angle between Holes = 90°
3. Distance between two annular fin = 25mm

B. Dimension of Inner Tube

1. Outer Diameter of Inner tube = 16mm
2. Inner Diameter of Inner tube = 13.6mm
3. Thickness of Inner tube = 1.2mm
4. Length of Inner tube =450mm

C. Dimension of Outer Tube

1. Outer Diameter of Outer tube = 30mm
2. Inner Diameter of Outer tube = 24mm
3. Thickness of Outer tube = 3mm
4. Length of Outer tube = 450mm

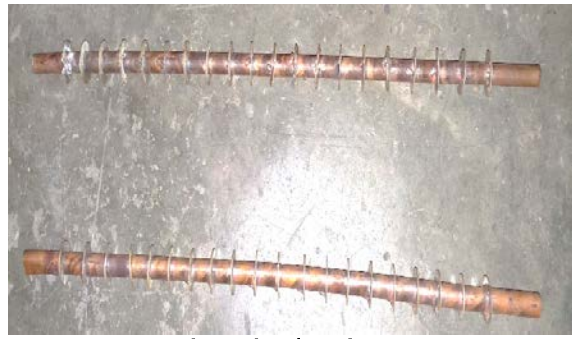

Fig.4 Plain and Perforated Fin Arrays

Fig. 5 Actual Experimental Set-up

Figure 5 shows the actual setup for experimentation. Two test section used for two different types of fins arrangement. The internal pipe having fins on the surface and carries the hot water. The hot water supply is taken from water tank where water is heated with a water heater. The circulation of water takes place with the help of water pump. The mass flow rate of water is controlled with the help of flow control valve.

IV. EXPERIMENTATION ON TEST SECTION

The test is carried out on two different test sections. The detailed procedure to be followed to conduct trail as

1. Start the main switch of the setup.

2. Heat the water to set temperature in a heater.
3. After heating the water to set temperature to adjust the mass flow rate of hot water flowing through heating section. Make sure that water flow to other test section is closed.
4. Start the blower and set the mass flow rate of air flowing through the outer side with the help of flow control valve.
5. Note the temperature at the inlet, outlet of hot water and air; also observe the pressure difference in manometer for presser drop and mass flow measurement of air.

Repeat the process for different mass flow rate of hot water and air.

V. OBSERVATIONS

1. m_w- Mass flow rate of Hot Water flow (kg/min)
2. m_a- Mass flow rate of air (kg/sec)
3. T_1 – Temperature of Air Inlet (^{0}C)
4. T_2 – Temperature of Air Outlet (^{0}C)
5. T_3 – Temperature of Hot Water Inlet (^{0}C)
6. T_4 – Temperature of Hot Water Outlet (^{0}C)
7. T_5 – Surface Temperature of test section (^{0}C)
8. H_1- Manometer reading for Pressured drop across air flow (m)
9. H_2- Manometer reading for air flow (mm)

TABLE I OBSERVATION TABLE FOR TINS WITH PERFORATION

S. No.	M_w Kg/ min	M_a Kg/sec	Temperature				Surface temp T_5 (^{o}C)	H_1 (m)	H_2 (mm)
			Air		Hot Water				
			Inlet T_1 (^{o}C)	Outlet T_2 (^{o}C)	Inlet T_3 (^{o}C)	Outlet T_4 (^{o}C)			
1	2	0.001000	29.8	39.2	55	54.8	33	0.05	46
2	2	0.001095	29.9	38.9	55	54.8	32.9	0.06	63
3	2	0.001183	29.9	38.7	55	54.8	32.8	0.07	80
4	2	0.001265	29.9	38.5	55	54.8	32.7	0.08	102
5	2	0.001341	29.9	38.3	55	54.8	32.6	0.09	110
6	2	0.001414	29.9	38.1	55	54.8	32.5	0.1	140
7	2	0.001483	29.9	37.7	55	54.8	32.3	0.11	160
8	2	0.001549	29.9	37.5	55	54.8	32.2	0.12	205

VI. CALCULATIONS

A. Calculations for Reynolds Number

1. Input temperature of Air Outlet T1 = 39.2 ^{o}C
2. Output temperature of Air Inlet T2= 29.8^{o}C
3. Bulk mean temperature of air

$$T_b = \frac{T_1+T_2}{2} = \frac{39.2+29.8}{2} = 34.5\ ^0\text{C}$$

4. Properties of water at bulk mean temperature= 34.5 ^{0}C

Density (ρ) = 1.12 kg/m^3
Thermal Conductivity (k) = 0.02625 W/m.K
Dynamic Viscosity (μ) = 0.00001895 Kg/m.s
Specific heat of water (C_p) = 1.007 × 10^3 KJ/Kg. K
5. Discharge of air:
A_o = Area of Cross Section Orifice in m^2

$$A_o = \frac{\pi}{4} \times d^2$$

Q = Discharge of air in m^3 / sec

$$Q = C_d . A_o . \sqrt{2 \times g \times h \times (\rho_w/\rho_a}$$

$$Q = 0.6 \times \frac{\pi}{4} 0.008^2 \sqrt{2 \times 9.81 \times 0.05 \times (\frac{1000}{1.12})}$$

Q= 0.000893 m^3/sec

Where,
C_d = Coefficient of discharge of orifice = 0.6 to 0.58
A_o = area of cross section of orifice in m
ρ_w = Density of water = 1000 Kg/m2
ρ_a = density of air at ambient temp. = 1.12 Kg/m3
h = manometer reading in meter

5. Velocity of air (V_a) = $\frac{Discharge\ (Q)}{Area\ of\ Pipe} = \frac{Q}{\frac{\pi}{4}(Do^2 - Di^2)}$

V_a = 0.043728 m/s

6. Reynolds Number (R_e) = (ρ × V_a × D_i) / μ
= (1.12 × 0.043728 × 0.024) / 0.00001895
R_e = 62.0271
7. Prandlt Number (P_r) = (μ × C_p) / k
P_r = 0.7268

B. Calculation for Heat Transfer Coefficent and Nusselt Number

1. Mass Flow rate of Air
m_a = mass flow rate of air in Kg / sec
m_a = Q x ρ_a
= 0.000893 x 1.12
= 0.001kg/sec
Where,
ρ_a = Density of air at Ambt. Temp. = 1.12 Kg/m^3
2. Heat Transfer Coefficient (h)

A= (π × 0.024 × 0.45) +20 x ((π /4 x 0.024^2) - (π /4 x 0.016^2)) = 0.022861 m^2

$$h = q/[A \times (T_f - T_s))]$$

$$h = 9.46/[0.022861 \times (54.9 - 33)]$$

$h = 18.901\ W/m^2 k$

i) Nusselt Number (Nu) = $(h \times D_i)/k$

Nu = $(18.901 \times 0.024) / 0.02625$

Nu = 17.28

3. Pressure Drop

Area = $0.022861\ m^2$

Velocity, v = $(m/A \times \rho) = 0.043728$ m/s

$\Delta p = (\rho_w - \rho_a) \times g \times \Delta h = (989.1 - 1.12) \times 9.81 \times 0.05 = 445.84 N/m^2$

VII. RESULTS AND DISCUSSION

A. Nusselt's Number

The Fig. 6 shows the variation of Nusselt's number on Reynold's number for test section with fins having perforation and without perforation. As Reynold's number is increased the value for Nusselt's number is also increasing. The value of Nusselt's number is low for plain tube and insert without perforation, as compared to the Nusselt's number of inserts having the perforation. At a given Reynolds number, the use of fins leads a considerable increase of Nusselt's number as compared to without fins. This can be explained that the thermal boundary becomes thicker as Reynolds number decreases, thus the effect of boundary destruction by inserts turns out to be more prominent. The Nusselt's numbers for the fins with perforation are enhanced between 6.2% and 10.7%, over that for the fins without perforations. This is primarily attributed to the effect swirl flow induced by the perforated fins which leads to a stronger turbulent intensity and tangential contact between the fluid flow and tube wall.

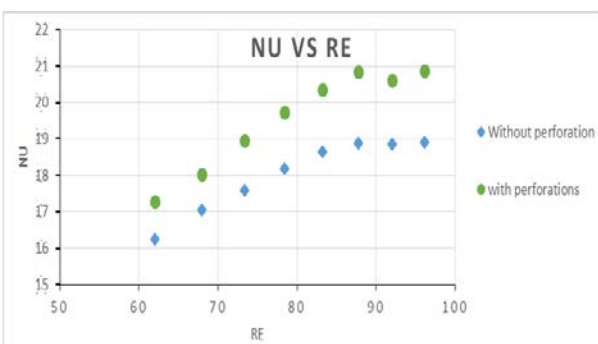

Fig. 6 Nu v/s Re

B. Pressure Drop

The effect of pressure drop for a different arrangement of fins is shown in Fig.7. As we use fins into the tube, it offered the resistance to the fluid motion. Hence the pressure into the system is decreased. For efficient system pressure drop required is minimum but because of resistance by inserts in this study, the pressure will get decreased.

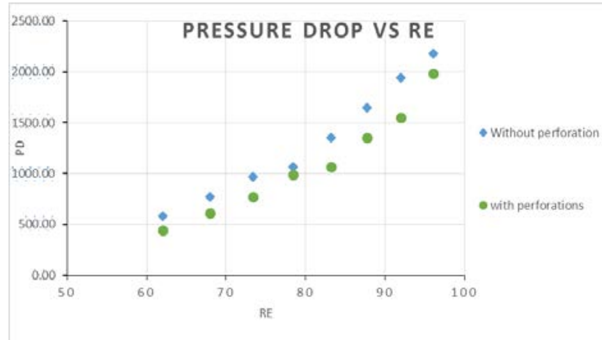

Fig.7 Pressure drop v/s Re

Above graph shows the effect of pressure drop with respect to the Reynolds number. As Reynolds number increases the pressure drop is increased because more turbulence is created and hence pressure drop has occurred. The fins without perforation offer maximum resistance hence pressure drop is more in this study. The minimum pressure drop is obtained into the fins with perforations. The more amount of fluid can pass through the perforation and hence less resistance can be offered by this type of arrangement.

C. Reynolds Number Vs Heat Transfer Coefficient

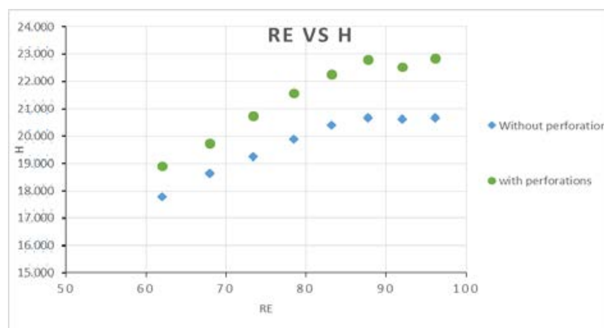

Fig. 8 H v/s Re

Above graph shows the effect of Reynolds number over the heat transfer coefficient. The heat transfer coefficient is directly proportional to Reynolds number. As the value of Reynolds number increases the value of heat transfer is also increases. The heat transfer coefficient value is more I case of fins with perforation as compared to the fins without perforations. The increase in heat transfer coefficient is from 6.3 % to 9.6% in fins with perforation over fins without perforation for the same range of Reynolds number.

VIII. CONCLUSION AND PERSPECTIVES

The effects of the Perforated in fins on the heat transfer enhancement and pressure drop behaviors in a turbulent flow are described. The Perforated fins, along with without perforation fins are tested using the air as the working fluid. The conclusions are drawn as follows

1. With the use of fins, Nusselt's Number increases but at the same time pressure drop also increases.
2. For same range of Reynolds Number, perforated fins

show greater Nusselt's Number, heat transfer coefficient than the value we get for fins without perforation because of an increased degree of turbulence created.

3. Use of Perforated fins increases the heat transfer coefficient by 6.3% to 9.6 % which will be a significant amount of increment in heat transfer rate.

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