International Journal of Engineering Research and Technology.
ISSN 0974-3154 Volume 10, Number 1 (2017), pp. 51-60

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Experimental Investigation of Heat Transfer through Porous Material Heat Exchanger

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Abstract

Latest developments in the manufacturing technology have led to development of advance lightweight materials for thermal applications. Heat transfer through porous materials has gained significance in industrial as well as academic research. In this paper thermal performance including heat transfer and pressure drop through porous material, i.e. metal foam heat exchanger, has been presented. The experimental data has been used to calculate and present graphically various performance parameters such as effectiveness, friction factor, Reynolds number and Nusselt number.The effectiveness of the heat exchangers was compared at $u = 0.5-7$ m/s fluid velocity, it was found that the best performance is exhibited by heat exchanger at effectiveness ($\varepsilon = 30\%$, $u = 0.2$ m/s). Maximum heat transfer occurs at Reynolds number of 900. For further investigation advance methods such as artificial neural networks, fuzzy logic and genetic algorithm can be used.

Keywords: Forced Convection, Heat Exchanger, Heat transfer, Pressure drop.

1. INTRODUCTION:

The use of porous materials as efficient and compact heat exchangers for heat dissipation is under extensive research. The louvered fin has a same level of the surface area density as the uncompressed metal foams. However, the manufacturing process of a louvered fin is very complicated and is costly as compared to the metal foams. There is a demand to make highly efficient compact heat exchanger by using metal foams which have a high heat transfer rate and structural strength as well as a low-cost manufacturing process [1]. In the past investigations have been carried out for heat transfer in metal foams for practical applications, including compact heat exchangers. There is a need to investigate the performance of porous material heat exchanger. Various researchers have conducted research on various aspects of porous media.Leong [1]studied heat transfer in oscillating flow through a channel filled with aluminum foam.

Calmidi et. al.[2] reported an experimental and numerical study of forced convection in highporosity (ε=0.89–0.97) metal foams. Schampheleire et. al.[3] carried out Thermo-hydraulic comparison of 10 ppi metal foam and louvered fins for low velocity applications. Joen et. al. [4]conducted Thermo-hydraulic study of a single row heat exchanger consisting of metal foam covered round tubes. Kashif Nawaz[5] conducted experimental studies to evaluate the use of metal foams in highly compact air-cooling heat exchangers.Experimental air heat transfer and pressure drop through copper foams. Mancin et. al.[6-8]studied heat transfer and pressure drop through copper and aluminum foams.Hooman [9] studiedHeat and fluid flow in a rectangular micro-channel filled with a porous medium.Tamayol[10] theoretically studied thermal assessment of forced convection through metal foamHeat Exchangers.

The aim of the experiment was to measure the thermal and hydraulic performance of the porous material heat exchangers in a cross flow arrangement. The concept was to supply the cold air at ambient temperature to flow through a square duct in which the porous material heat exchanger is placed, occupying the entire cross-section of the duct. The experiment of the porous material heat exchangers included measuring the coolant temperature and the pressure drop across the heat exchangers for various coolant flow rates.

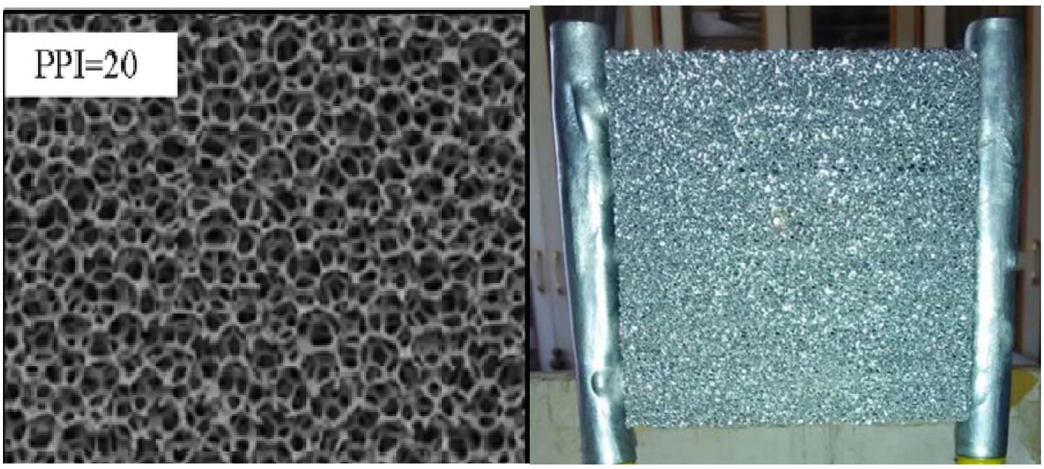

Fig. 1 Sructure of an 20 PPI metal foam and a heat exchanger fabricated out of it.

Porous Material is a solid containing void spaces (pores), connected or unconnected, dispersed within it in either a regular or random manner. These so called pores may contain a variety of fluids such as air, water, oil etc. flowing through it, mixed or unmixed.

Nomenclature:

A	surface area	H	Heat
K	permeability	k_s	thermal conductivity of solid
Cp	specific heat capacity	T	time
Nu	Nusselt number	u	velocity
p	pressure	X, Y, Z	Cartesian coordinates
g	gravitational acceleration		Greek symbols
Pr	Prandtl number	ε	porosity
k	thermal conductivity	μ_f	viscosity of fluid
k_f	thermal conductivity of fluid	ρ	density of fluid
K	Permeability	μ	Dynamic Viscosity
Re	Reynolds Number	θ	Temperature
Th_{mean}	Bulk mean temperature of hot fluid		
Tc_{mean}	Bulk mean temperature of cold fluid		

2.DETAILS OF EXPERIMENTAL SETUP:

The experimental setup consisted of a square duct of (0.101m X 0.101 m) cross section. Air was induced to flow through the duct. The air velocity was varied with a fan and a speed regulator. In order to prevent turbulence from increasing; a flow straightner was provided in the inlet section. Water was used as a hot fluid; the water was heated with a heater of 1 kW installed at the bottom of the tank.

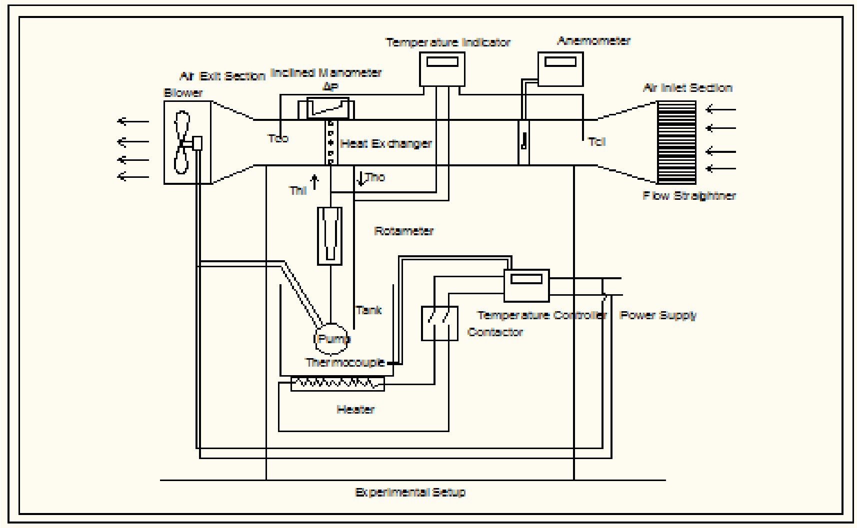

Fig. 2. Experimental Setup used in the study.[12]

The temperature was varied with the thermostat. A submersible pump was used to maintain water circulation in the tank. The water temperature at the set temperature was supplied to the heat exchanger, at constant flow rate. The air velocity was measured with a digital anemometer. The temperatures of hot and cold fluid were measured with K type thermocouples. The observations were recorded after the water temperature reached the steady state. The air speed, pressure drop, the temperatures were manually recorded in to a spreadsheet. Properties of the fluid**s** used in the experiment were taken from the heat transfer data book, corresponding to the data collected, and some of the properties were taken directly from the measured experimental data.

3. EXPERIMENTAL PROCEDURE:

After heating of water in each experimental stage, as soon as steady state was attained, Air inlet (Tci) and air outlet temperatures (Tco), hot water inlet (Thi) and hot water outlet (Tho), was measured from the respective inlet and exit points using thermocouples. An inclined manometer was used to read the pressure difference across the inlet and outlet of the test zone (Heat Exchanger). Note that, the hot fluid inlet temperature is heat exchanger's inlet temperature to the tubes whereas; the hot fluid outlet temperature is heat exchanger's outlet temperature.

The thermocouples were inserted into the inlet and outlet pipes in order to protect them from environmental effects**.** Hot water at a temperature range of 303–333 K was made to pass through the pipes at a constant mass flow rate of 0.075 kg/s. The pressure drop between the inlet and outlet of the test sections was around 3.5-8.5 mm-wc. The velocity of the cold fluid was measured with digital anemometer to calculate the mass flow rates of cold fluid. Air velocity (u) was varied from 0.2 m/s to 2.6 m/s at an increasing interval of 0.3 m/s. All measurements were conducted for steady-state conditions. The indicated values were recorded into a Microsoft Excel spreadsheet. The experimental data were used for calculations and plotting graphs which helped in determining the thermal performance of heat exchanger. The heat capacity of open cell aluminum foam heat exchangers were examined experimentally. For the aluminum foam heat-exchanger test unit, AlSi7 Mg aluminum alloy foam material with 20 pores per inch (PPI) was used (Fig. 1).

4.HEAT EXCHANGER DETAILS:

Open-cell aluminum metal foam sample having dimensions of (length X width X height) 0.010 X 0.101 X 0.101 mm sandwiching the aluminum tubes acted as fins for the heat exchangers. Open-cell aluminum metal foam heat exchangers were designed as one pass heat exchangers with 5 tubes having diameters of 0.01m connected in parallel arrangement to each other. The aluminum tubes were made to be in secure contact with the filaments using 6 steel pins. Detailed thermo-physical parameters and dimensions of aluminum foams have been presented in Table 1. The dimensions of the heat exchangers, number of pipes, and pipe diameters were kept the same during the experimentation process. Heat exchangers' surfaces that face outwards were insulated. For fabricating heat exchanger, two foam panels were imported.

Table 1. Thermo-physical parameters and dimensions for aluminum foams panel (ERG Aero.)

Pore density (PPI)	**20**
Material	AlSi7Mg
Porosity, ε	0.90
Heat conductivity, k(W/m-K)	165
Density ρ(kg/m^3)	230
Height, H (mm)	101
Length, L (mm)	101
Width, W (mm)	10

5.CONSTANT PARAMETERS IN THE EXPERIMENTATION:

The hot fluid (water) Mass flow rate (m_h), Porosity (ε_p=0.90), The number of tubes parallel to the flow direction, N_L=1, The number of tubes perpendicular in the flow direction, N_T=5, The distance between the tubes perpendicular to the flow direction, S_T=0.01 m, The number of tubes, N_t=05, Tube diameter, d_i=0.010m, Hydraulic diameter, d_h=0.101m, Perimeter of the duct, P=4×0.101m=0.404m, The duct cross-section area, A_c= 0.101m x0.101 m =0.010 m^2, The length of one side of the square duct=0.101m, Number of passes=1.

6.CALCULATION OF FLUID PROPERTIES:

Hot and cold fluid properties were taken at mean film temperature. Accordingly,

i. The specific heat of the hot fluid $(Cp_h) = \frac{(T_{hi} + T_{ho})}{2}$,

ii. The specific heat of the cold fluid $(Cp_c) = \frac{(T_{ci} + T_{co})}{2}$,

iii. Cold fluid density $(\rho_c) = \frac{(T_{co} + T_{ci})}{2}$,

iv. Kinematic Viscosity $(\upsilon) = \frac{(T_{ci} + T_{co})}{2}$,

v. Dynamic viscosity $(\mu) = \frac{(T_{ci} + T_{co})}{2}$,

vi. Prandtl number (Pr), $Pr_{air} = \frac{(T_{hi} + T_{ho})}{2}$

7.CALCULATION OF THERMAL PARAMETERS:

The heat supplied by the hot fluid, $q = m_h c_{ph}(T_{hi}-T_{ho})$ ………..(5.1)

The heat received by the cold fluid, $q = m_c\, c_{pc}(T_{co}-T_{ci})$………..(5.2)

If $C_h > C_c$, then $C_{min} = C_c$ ……………………………………… (5.3)

The following relation can be used for the effectiveness of the heat exchanger,

$$\varepsilon = \frac{q}{q_{max}} = \frac{C_c(T_{co}+T_{ci})}{C_{min}(T_{hi}+T_{ci})} \quad \text{…………………………………………… (5.4)}$$

The heat transfer due to convection is given by,

$$q = h.A_s\left(\frac{(T_{hi}+T_{ho})}{2} - \frac{(T_{ci}+T_{co})}{2}\right) \quad \text{……………………………….. (5.5)}$$

$$q = h(\pi.D.L)N_t\left(\frac{(T_{hi}+T_{ho})}{2} - \frac{(T_{ci}+T_{co})}{2}\right) \quad \text{……………………………(5.6)}$$

At the same time, the heat received by the cold fluid can also be given as follows:

$$q = m_c C_{pc} \left(T_{co} - T_{ci}\right) = \rho_c N_t S_t u\, Cp_c \left(T_{co} - T_{ci}\right) \quad ....(5.7)$$

Equating equation and (5.6) and (5.7), the heat transfer coefficient, Can be written as,

$$h = \frac{\rho_c N_t S_t u_c Cp_c \left(Tco - Tci\right)}{\left(\pi DL\right) N_t \left(Th_{mean} - Tc_{mean}\right)} \quad ....(5.8)$$

Where,

$$Th_{mean} = \frac{Thi + Tho}{2}$$

$$Tc_{mean} = \frac{Tci + Tco}{2}$$

Nusselt number, $Nu = \frac{hd_h}{k}$ ....(5.9)

Duct hydraulic diameter, $d_h = \frac{4A_c}{P}$ ....(5.10)

Reynolds number, $Re_{max} = \frac{\rho u d_h}{\mu}$ ....(5.11)

The friction factor, $f = \frac{\Delta p}{4\left(\frac{L}{d_h}\right)\left(\frac{\rho u^2}{2}\right)}$ ....(5.12)

8.RESULT AND DISCUSSION:

In this study performance of aluminum foam heatexchanger was investigatedexperimentally. By using the experimental data,effectiveness, friction factor, Reynoldsnumber and Nusselt number were calculated and presented inFigs. 3−6.Fig.3shows variation of effectiveness with velocityfor aluminum foam heat exchangersis given. The effectiveness of aluminum foam heat exchangersvaries between 10% and 33%, the filamentsof aluminum foam heat exchangers are in point contactswith the tubes through which the hot fluid passes: and thisdecreases the contact area hence the effectiveness.Fig. 4 shows variation of pressure loss per unit length with velocity for open cell aluminum foam heat exchangers the result is similar to [1-2]. The highest pressure loss was recorded at ΔP/L = 5.2N/m, at u =2.7 m/s). Variation of -frictional factor with Reynolds number for open cell aluminum foam is presented in Fig. 5. The highest friction factor was seen at Reynolds number of 200.It was observedthat the friction factor is high at low Reynolds numbers, but asthe Reynolds number increases friction factor tends to decrease In Fig. 6; the variation of Nusselt number with Reynoldsnumber for open cell aluminum foam is presented. The tests in this study were conducted atReynolds number range of about 200-1700. The values of Nusselt number for aluminumfoam heat exchangers were higher at higher Reynolds number.

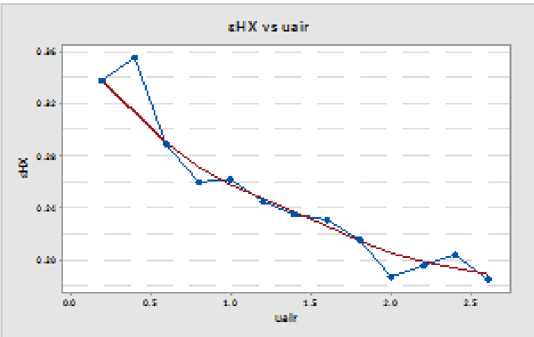

Fig. 3 Effectiveness versus velocity

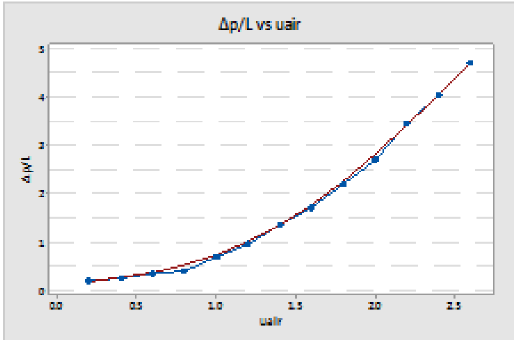

Fig. 4 Pressure drop per unit length versus velocity

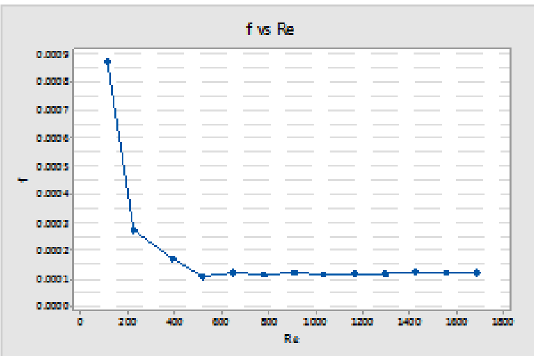

Fig. 5 Friction factor versus Reynolds number

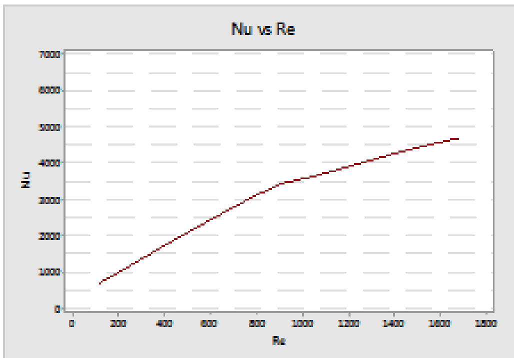

Fig.6 Nusselt number versus Reynolds number

9. CONCLUSION:

The performance of open-cell aluminum foam heatexchanger is summarized as follows:

In this study, it was found that, for heat exchangers, effectivenessis high at low (fluid) velocities and vice versa. Whenthe effectiveness of the heat exchangers are compared atu = 0.5−7 m/s fluid velocity, it is found that the best performancewas exhibited by aluminum foam havingfeature (ε = 30%, u = 0.2 m/s). Maximum heattransfer occurs at interval with corresponding valuesof Nusselt and Reynolds number of 900 and

1800 respectively.Heat transfer performance of aluminum foam heatexchangers can be compared with those of other metal foam heatexchangers such as (nickel, magnesium, lead, zinc, and copper carbon).An optimum PPI value can be investigated by using differentPPI values for open cell aluminum foam heat exchangers. The performance can also be investigated using (artificial neural networksand genetic algorithm methods).

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