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Oronzio Manca
Seconda Universita' degli Studi di Napoli, oronzio.manca@unina2.it

Bernardo Buonomo
Seconda Università degli Studi di Napoli

Luca Cirillo
Seconda Università degli Studi di Napoli

Sergio Nardini
Seconda Università degli Studi di Napoli

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AN EXPERIMENTAL INVESTIGATION ON EFFECT OF PORES PER INCH IN COMPACT HEAT EXCHANGER WITH ALUMINUM FOAM

Bernardo Buonomo, Luca Cirillo, Oronzio Manca and Sergio Nardini
Dipartimento di Ingegneria Industriale e dell'Informazione, Seconda Università degli Studi di Napoli, Via Roma 29,81031, Aversa, Caserta, Italy

ABSTRACT
In this paper an experimental investigation on forced convection in a compact heat exchanger made up with an aluminum foam plate of 212.5mm x 212.5mm with a thickness of 40 mm and a single array with five circular tubes is presented. The foam has a porosity of 0.93 with 10, 20 and 30 pores per inch and the tubes in aluminum have internal and external diameters equal to 9.5 mm and 12.5 mm. The test rig consists of an open air channel and a closed water cycle and the aluminum foam plate is placed inside the channel. The performances of the compact heat exchanger are evaluated for assigned hot water mass flow rate and different hot water inlet temperatures and air mass flow rate.
Results are given in terms of heat transfer rates and pressure drops as a function of air velocity and Reynolds numbers. The evaluation of dimensionless, Colburn factor and Nusselt number is performed for different air mass flow rates and hot water inlet temperatures.

INTRODUCTION
Metal foams are a new class of materials with low densities and novel thermal and mechanical properties. The open porosity, low relative density and high thermal conductivity of the cell edges, the large accessible surface area per unit volume, and the ability to mix the cooling fluid by promoting eddies all make metal foam heat exchangers efficient, compact and light weight. Aluminum foams combine low weight with good rigidity, strength, damping of vibrations and noise, shock resistance and low thermal conductivity [1]. The possible applications are dependent on many parameters such as the type of metal, the porosity (open or closed), the process by which the foam is obtained (temperature, pressure, residence time, melt flow etc.), density and overall costs.
The advantages of metal foams reside in low density and high strength of the structure. The low thermal inertia allows a transfer of heat faster than in the ceramic materials. The material has a high ratio between surface area and pressure drop, and with uniform lower density. The pressure drop is lower than in ceramic structures when considering the unit of volume [2]. Sumithra Raju and Narasimham [3] carried out a numerical analysis on heat transfer and pressure drop of an inline tube bundle (5 rows) with porous fins and air as fluid. The thickness of the porous fin varied from 0 to the half of the transversal tube pitch. The results showed that for a porosity equal to 0.58 and fin thickness equal to the tube radius an increase of Nusselt number and in pressure drop with respect the system without porous fin of about 30% and 25%, respectively. However, for compactness of the heat exchanger, the porous fins with high pore density and low porosity are preferable. Aluminum foam as a small compact heat exchangers was investigated by Boomsma et al. [4]. It was employed as a heat sink in electronic cooling with the foam brazed onto a heat spreader plate and water was used as working fluid through the metal foam. The impact of the compression on the resulting thermo-hydraulics was studied and it was found that increasing the compression rate resulted in both a larger pressure drop and heat transfer. The increase in metal foams compression determines an heat transfer rate increase but compressing the foams too much an overall performance worsen is detected due to the very sharp increase in pressure drop. Kim et al. [5] carried out an experimental analysis on heat transfer and pressure drop of aluminum foam brazed between two flat tubes (porous fin) of a plate-fin water–air heat exchanger. Comparison of performance between the porous fins and the conventional louvered fins was provided. In the investigation, six types of foam were considered with varying porosity (0.89–0.96) and varying the pores per inch, PPI, (10–40). The porous fins showed similar thermal performance to the conventional louvered fin although the louvered fin presented a little better performance in terms of pressure drop. An experimental investigation on a single row of aluminum tubes, covered with layers of aluminum foams, was carried out by T'joen et al. [6]. A range of foam layer thickness, Reynolds number tube spacing and different type of foam were considered and compared with compact helically finned tube heat exchangers. Results indicated that thermal contact resistances have a strong effect on the performance of the metal foam heat exchangers. It was observed that for stream velocity higher than 4 m/s, a good metallic bonding, small tube spacing and thin foam layer thickness potentially improve the

performance of metal foam heat exchangers compared to a conventional finned-tube heat exchanger. An experimental investigation was carried out by Sertkaya et al. [7] to compare three metal foam heat exchangers (10, 20 and 30 PPI) to three finned heat exchangers with the same tube layout and overall dimensions. Results showed that the finned heat exchangers furnished a higher heat transfer and a lower pressure drop than the foamed heat exchangers. Chumpia and Hooman [8] compared in an experimental investigation five foam wrapped tubes to a finned tube as benchmark. The results show that the foam heat exchangers with thicker foam layer perform better than those with thinner foam layer. Moreover, the foam wrapped heat exchanger with suitable foam thickness gives heat transfer benefit while keeping the pressure drop at the same level of the finned tube. A strong dependence on the foam's morphology and foam material of thermal hydraulic performance of metal foam heat exchangers was provided by of Huisseune et al. [9]. Dixit and Ghosh [10] have been performed an experimental study on open cell metal foam as extended heat transfer surface. Experiments have carried out with high porosity, open cell copper foam blocks sandwiched between plates at constant temperature. They have observed that the convective fluid temperature, at any location perpendicular to the direction of the base plate, decreases with increase in velocity. They have also observed that: metal foam attached to a plate can be treated as extended heat transfer surface attached to a primary heat transfer area; the heat transfer analysis for open porous metal foam based on simple cubic model has been able to explain the experimentally obtained temperature data with practically good accuracy. A numerical analysis to evaluate the performance of metal foam heat exchangers and compare it to the performance of a bare tube bundle and of an existing conventional louvered fin heat exchanger was presented by Huisseune et al. [11]. It was found that, at the same fan power, the foamed heat exchangers show up to 6 times higher heat transfer rate than the bare tube bundle. Whereas, for the same overall dimensions, the finned heat exchanger showed the best performance in comparison with the metal foam heat exchanger. However, a metal foam heat exchanger can outperform the finned heat exchanger if the frontal area is changed. From the previous short review it is clear that the thermal and fluid dynamic performance parameters of metal foam heat exchangers present several uncertainties and more numerical and experimental investigations are necessary. The aim of this experimental investigation on a air-water aluminum foam heat exchanger is to evaluate its thermal and fluid dynamic characteristics. Results are given for different pores per inch (10,20 and 30 PPI) air mass flow rate in a range of laminar flow in terms of heat transfer rate, performance evaluation coefficient, thermal resistance and air difference temperatures.

NOMENCLATURE

A = area of the duct (m^2)
C = heat capacity (W/K)
c_p = specific heat at constant pressure (J/kg K)
D = pipe diameter (m)
D_{hyd} = hydraulic diameter (m)
f = friction factor
h = convection heat transfer coefficient (W/m^2)
j = Colburn factor
k_c = thermal conductivity (W/m K)
L = length of the channel (m)
m = mass flow rate (kg/s)
N = number of cylinders in the bank
Nu = Nusselt number
p = wetted perimeter (m)
PPI = number of pores per inch
Pr = Prandtl number
Re = Reynolds number
T = Temperature (K)
u = velocity (m/s)
Z = depth (m)

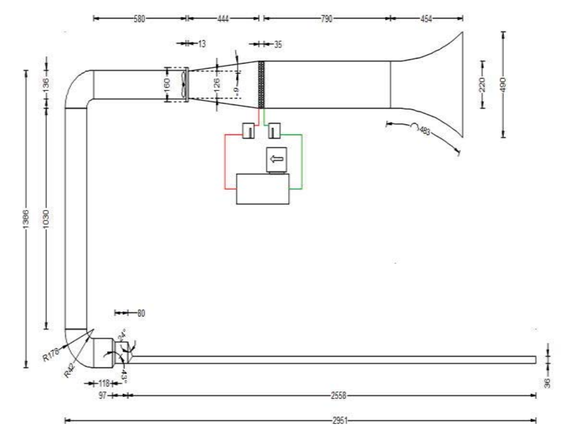
Greek symbols
ΔP = pressure loss (Pa)
ε = effectiveness
ρ = density of the air (kg/m^3)

Subscripts
c = cold fluid
in,c = inlet cold fluid
out,c = outlet cold fluid
h = hot fluid
in,h = inlet hot fluid
out,h = outlet hot fluid

1 Experimental apparatus

The evaluation of heat transfer behavior related to the aluminum foam heat exchanger, in terms of convective average coefficient, Nusselt number and effectiveness was carried out by means of the apparatus sketched in Fig. 1a. The hot fluid is water and it flows in the internal tubes placed inside the metal foam whereas the cold fluid is air. The water enters in the heat exchanger at an assigned temperature and its value is monitored through a thermocouple. The maximum volumetric flow rate of water in the circuit was about 106.2 l/h. The air enters inside a duct where is placed the heat exchanger, through a convergent channel, as reported in Fig. 1 on the right of the upper part. The duct has a square transversal section of 220 mm x 220 mm and 760 mm long, whereas the convergent duct is 454 mm long and it has the squared transversal inlet and outlet sections equal to 490 mm x 490 mm and 220 mm x 220 mm, respectively. Air motion is obtained by means of a fan, which is modulated by an AGILENT E3633A power supply system, in order to obtain different air flow rates. The air volumetric flow rate ranges between 1.0 m^3/h and 6.0 m^3/h. At the end of this duct there is the heat exchanger, as shown in Fig. 1. Beyond the test section, the air flow enters a second duct of trapezoidal shape, having a larger section and smaller section of 220 mm x 220 mm and 126 mm x 126 mm, respectively. A series of pipes allows to reduce the section from 160 mm, downstream the fan, to 36 mm at the exit section of apparatus. in order to obtain a more accurate measurement of the velocity by means of a Pitot tube. The exit tube has a diameter equal to 36 mm and 2.550 m long. The tubes downstream the fan were in PVC and the junctions between two consecutive tubes were made using a silicone sealant.

The heat exchanger is made of two plates of aluminum foam produced by M-PORE [12] with a porosity of 0.95 and 10, 20 and 30 pores per inch (PPI). The plates have dimensions equal to 212.5 mm x 212.5 mm x 20 mm, as shown in Fig. 1b. Grooves were made in such a way as to allow the location of five aluminum pipes 12.5 mm diameter and length equal to 270 mm. Flexible steel tubes were used in order to connect the aluminum tubes inside the metal foam. The side surfaces were insulated with mineral wool to avoid heat losses between the heat exchanger and the external ambient. An AGILENT 34980A multifunction measurement unit and a computer were used for the data acquisition. The air and water temperatures were measured by means of 0.50 mm OD ungrounded iron-constantan (J-type) thermocouples. In the air, one thermocouple was placed at the inlet section, close to the surface of the convergent channel of the duct, and nine thermocouples were allocated in the center line of the section at 30 mm behind the heat exchanger. They were at different positions from 40 mm to 200 mm with a step of 20 mm with respect to the lower horizontal surface of the duct. To evaluate the water temperature at the inlet and outlet sections of the heat exchanger one thermocouple was placed inside an adiabatic mixer at the inlet section and two thermocouples were inside an adiabatic mixer at the outlet section. Moreover, surface temperature of aluminum tubes were measured at inlet and outlet of each five tubes. Average surface temperature, T_{sa}, was evaluated by the average among these ten temperature values. An Isotech instrument mod. 938 ice point, with an accuracy of ±0.04 °C and 50 channels, was used as thermocouple reference junctions. The thermocouples were calibrated in a range from 19 °C to 75 °C and calibration of the temperature measuring system showed an estimated precision of ±0.1 °C. Mass flow rate was evaluated by measuring the velocity at the exit tube, with a diameter of 36 mm, by a Pitot tube Fluke mod. 922 Airflow meter. The Pitot tube was located at the center of the section at 0.100 m from the outlet section of the apparatus. In this section a fully developed flow was attained and in the air mass flow rate, considered in the present work, a laminar flow was achieved. Pitot tube was located at 60 tube diameters from the inlet of the exit tube. To evaluate the pressure drop a digital manometer, Fluke mod. 922 Airflow meter with an accuracy ±1%, was used. The measurements were obtained estimating the pressure upstream and downstream the heat exchanger.



a)

b)

Figure 1: (a) Sketch of experimental apparatus; (b) Heat Exchanger with metal foam

2 Data reduction

The inlet water temperature was set equal to 50°C, 60°C and 70°C. The metal foams 10, 20 and 30 PPI were investigated. The heat transfer rate from water is:

$$Q = m_h c_p (T_{in,h} - T_{out,h}) \tag{1}$$

where $\dot{m}_h$ is mass flow rate, c_p is specific heat of water,$T_{in,h}$ is the inlet temperature of hot fluid (water) and $T_{out,h}$ is the outlet temperature of hot fluid (water).

$$Nu = \frac{\bar{h} D_{hyd}}{k_c} \tag{2}$$

where D_{hyd} is hydraulic diameter, defined as

$$D_{hyd} = \frac{4A}{p} \tag{3}$$

k_c is the thermal conductivity of cold fluid (air) and $\bar{h}$ is heat transfer coefficient can be evaluated [7]:

$$\bar{h} = \frac{\rho N_t S_t u C_{pc} (T_{out,c} - T_{ic,c})}{\pi NDZ \left(\frac{T_{h,in} + T_{h,out}}{2} - \frac{T_{h,in} + T_{c,out}}{2} \right)} \tag{4}$$

Where N_t (=5) is number of tubes in the transverse direction of a bank, S_t(=2) is transverse pitch in a bank of tubes, N is number of tubes in a bank, D is pipe diameter and Z is depth. Friction factor f is defined as:

$$f = \frac{\Delta P}{4\left(\frac{L}{d_h}\right)\left(\frac{\rho u^2}{2}\right)} \tag{5}$$

The heat exchangers are commonly characterized by Colburn factor j, which provides a further estimate of heat exchanger performance and it is calculated using the equation as follow:

$$j = \frac{Nu}{\mathrm{Re}\,\mathrm{Pr}^{1/3}} \tag{6}$$

Effectiveness is the maximum possible heat transfer rate fraction of the actual heat transfer rate. The effectiveness can be given as below [7]:

$$\varepsilon = \frac{C_c\left(T_{c,out} - T_{c,in}\right)}{C_{\min}\left(T_{h,in} - T_{c,in}\right)} \tag{7}$$

where $T_{c,out}$ is the outlet temperature of cold fluid, $T_{c,in}$ is the inlet temperature of cold fluid and $T_{h,in}$ is the inlet temperature of hot fluid. Here C_c is the heat capacity rate of cold fluid, C_{min} can be espressed as follows:

$$C_c = m_c C_{pc} \tag{8}$$

$$C_{min} = m_h C_{ph} \tag{9}$$

2.1 Uncertainty

The uncertainty of Nusselt number, Reynolds number, efficiency are evaluated with the Kline and McClintock methodology. It is estimated that the uncertainty on the Nusselt number is 3.7%, the Reynolds number is 2.6%, effectiveness is 4.3 % and Colburn factor 3.8%.

3 Results

Tests were carried out with distilled water as working fluid. Results are presented in terms of heat transfer rate, pressure drop, efficiency as a function of air mass flow. Fig. 2 (a-d) shows heat transfer rate as a function of air mass flow rate at different temperatures and PPI of aluminum foam. Heat transfer rate increases with air mass flow as expected. Each value is obtained from the average of ten different measurements. At the same inlet temperature heat transfer rate increases with air mass flow particularly for low values whereas for high values profiles tend to be nearly constant. Heat transfer rate is higher for metal foam of 20 PPI at 70°C of hot fluid, in fact, the higher value is 463 W.

For metal foams the Nusselt number is always increasing and it reaches the maximum value of 8257 for the maximum investigated Reynolds number (fig.3 a-d). Without metal foams the Nusselt number increases until it reaches a costant value. Figure 3 (a-d) shows the efficiency as a function of air mass flow at three different temperatures. It is obtained that the metal foams improve the heat transfer. In particular, the higher efficiency it reaches at 50 °C with metal foam of 20 PPI.The highest Colburn factor was seen on 20 PPI of the aluminum foam while the lowest values was recorded on 10 PPI of the aluminum foam. In this study, for all the heat exchangers, it was found that at low air mass flow, the friction factor is high but as the air mass flow increases friction factor tends to decrease [fig.4 a-b].

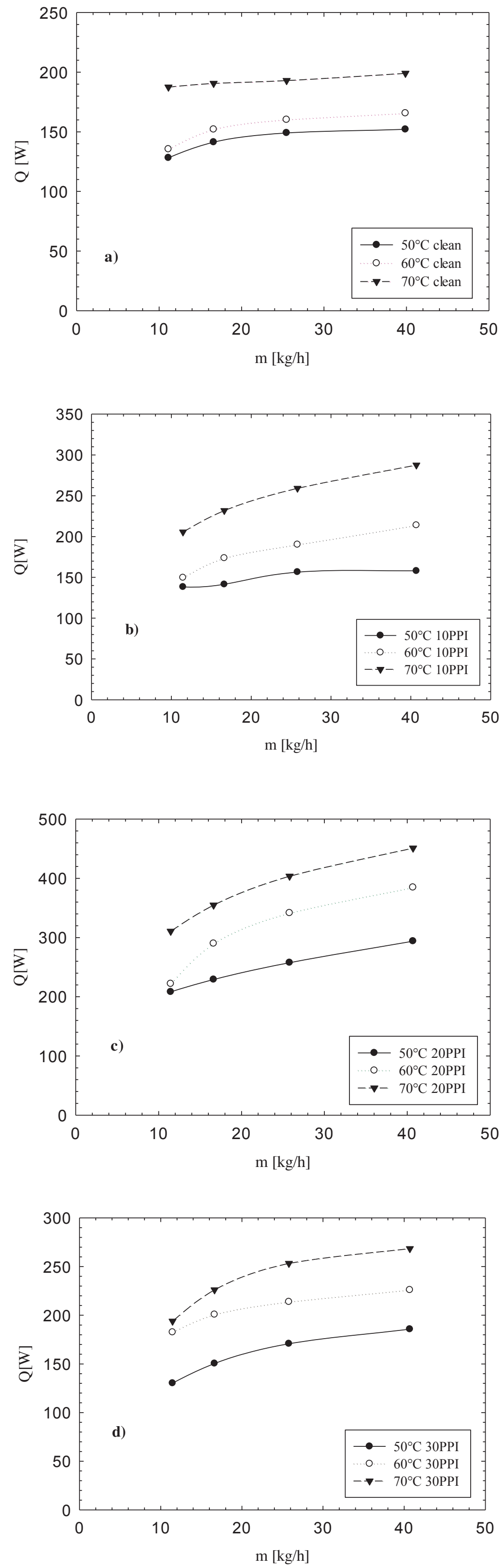


Figure 2: Heat transfer rate as a function of air mass flow at different temperatures: a) clean; b) 10 PPI; c) 20 PPI; d) 30 PPI

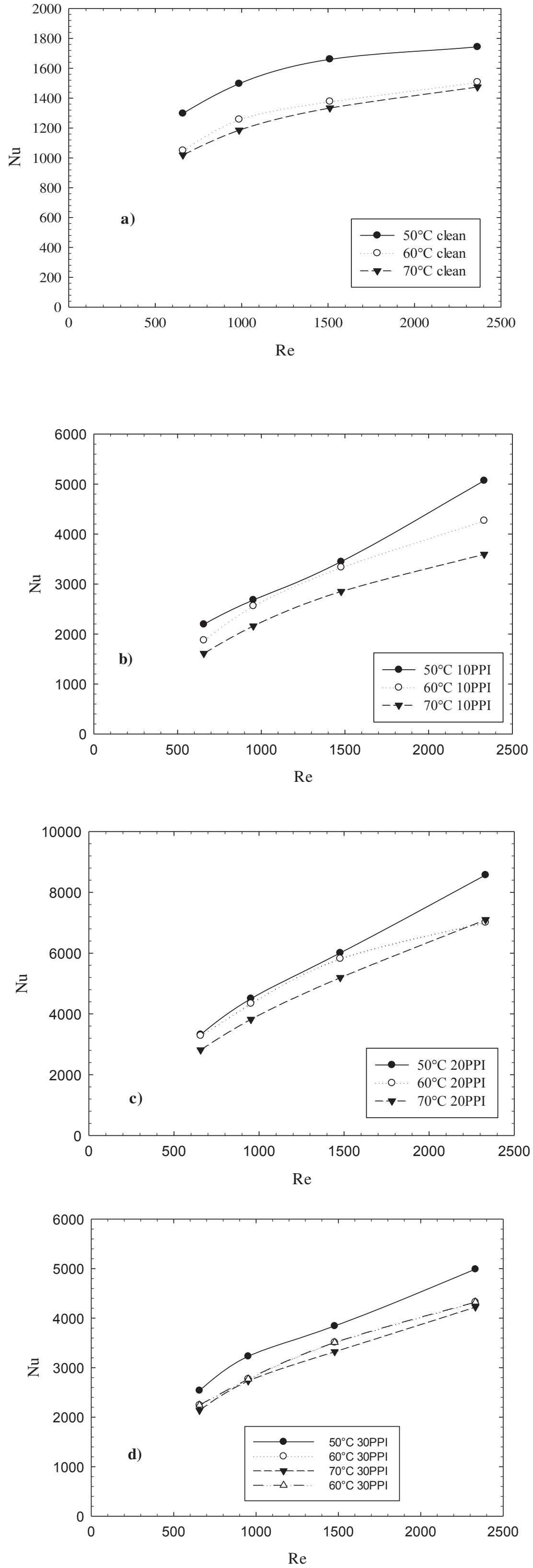


Figure 3: Nusselt number as a function of Reynolds number at different temperatures: a) clean; b) 10 PPI; c) 20 PPI; d) 30 PPI

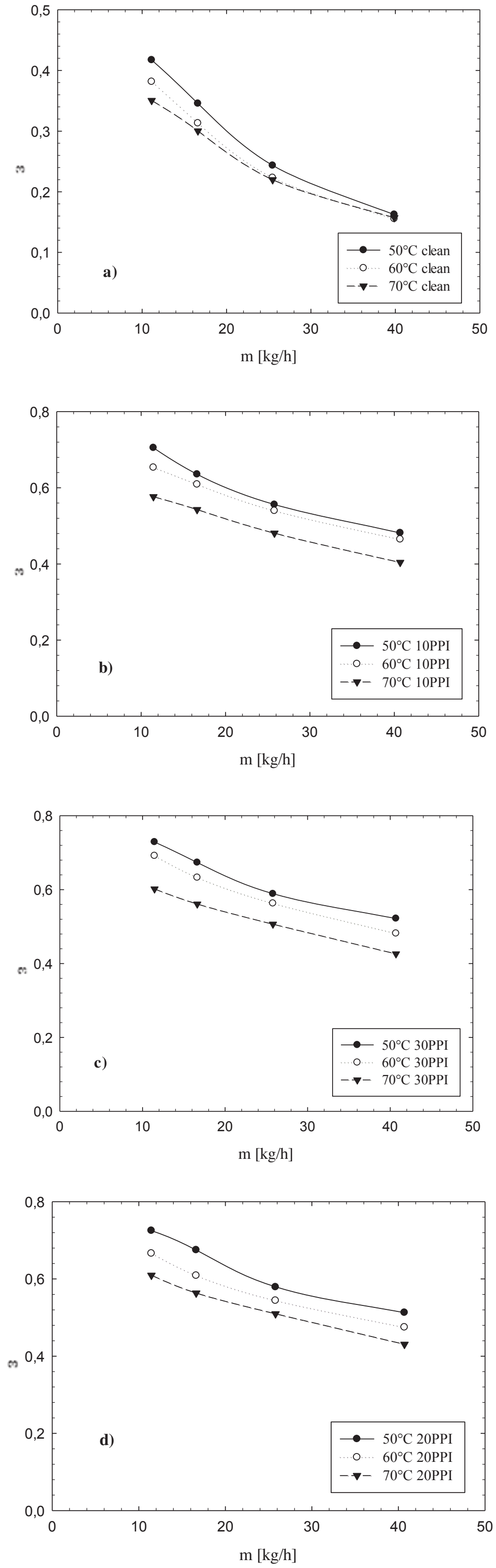


Figure 4: Effectiveness as a function of air mass flow at different temperatures: a) clean; b) 10 PPI; c) 20 PPI; d) 30 PPI

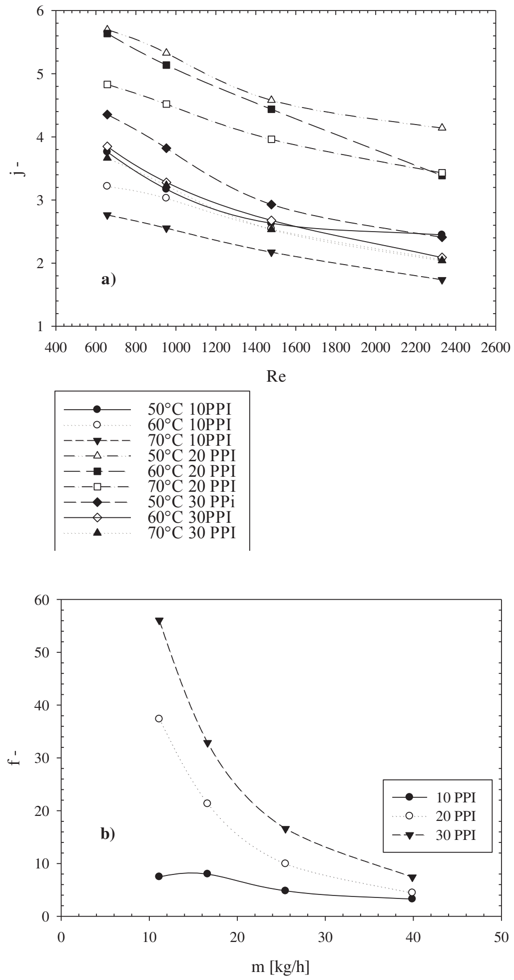


Figure 5: a) Colburn factor as a function of Reynolds number; b) friction factor as a function of air mass flow rate.

CONCLUSIONS

In this experimental investigation the thermodynamic of a metal foam 10, 20 and 30 PPI was considered. The investigation was carried out on a metal foam with 10, 20 and 30 PPI and porosity 0.95 metal foam. Steady state for a heat exchanger is reached later for hot fluid at 50°C than for 60°C and 70°C. In this study, it was found that, for all heat exchangers, effectiveness is high at low air mass flow rate values and vice versa. It is found that the best performance was exhibited by aluminum foams whereas the least effective heat exchanger was that made at clean configuration.Heat transfer rate in 20 PPI aluminum foam heat exchangers is higher than in the clean configurations. Maximum heat transfer occurs with 20 PPI corresponding value of 463 W at 70°C.The effectiveness of 20 PPI aluminum foams is about twice as higher as that of clean configurations. It is hoped that more interesting results will follow in further studies. Heat transfer performance of aluminum foam heat exchangers can be compared with those of other metal foam heat exchangers such as nickel, lead, zinc, magnesium and copper carbon.

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