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# Experimental Studies to Evaluate the Use of Metal Foams in Highly Compact Air-Cooling Heat Exchangers

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## *Abstract:*

open-cell aluminum foam is considered as a highly compact replacement for conventional fins in brazed aluminum heat exchangers. The experimental data needed to for evaluation are obtained through wind-tunnel experiments. Using a closed-loop wind tunnel, heat transfer and pressure drop measurements are undertaken, and the results are characterized as Colburn *j*-factor and friction factor for geometric configurations of foam and flat tubes that mimic current multi-louver, microchannel geometries. The data obtained in the wind-tunnel are used to evaluate the potential for using metal foam (of varying porosity) as a replacement to conventional fins for a range of configurations and operating conditions typical for air-cooling applications. Finally, we comment on some of the challenges that must be met for the adoption of this technology in air-cooling systems.

## *Introduction:*

Metal foams are porous media with low density and novel thermal, mechanical, electrical, and acoustic properties [1]. They can be categorized as open-cell or closed-cell foams, but only open-cell metal foams appear to have promise for constructing heat exchangers. Open-cell metal foams have high specific surface area, relatively high thermal conductivity, and a tortuous flow path to promote mixing. Metal foam have been studies by a number of researchers for thermal applications; some were focused on metal-foam heat exchangers (and heat sinks), and many others investigated the basic thermal transport properties of metal foams. The basic properties of the metal foams include the effective thermal conductivity, permeability, and inertial coefficient. Calmidi and Mahajan [2] investigated the effective thermal conductivity of high-porosity fibrous metal foams experimentally. An empirical correlation was developed and a theoretical model was derived. The model predictions agreed closely with the experimental data and were used for the evaluation of metal foams as possible candidates for heat sinks in electronics cooling applications. Boomsma and Poulikakos [3] see also [4] developed a one-dimensional heat conduction model for use with open-cell metal foams, based on idealized three-dimensional cell geometry of the foam. Their model showed that the fluid-phase conductivity has a relatively small effect on the effective thermal conductivity, and the overall effective thermal conductivity of the metal foam is controlled by the solid-phase conductivity to a large extent. Bhattacharya *et al.* [5] conducted research on the determination of the effective thermal conductivity, permeability, and inertial coefficient of highly porous metal foams. A theoretical model was formulated and the analysis showed that the effective thermal conductivity depends strongly on the porosity and the ratio of the cross-sections of the fiber and the intersection, but no systematic dependence on pore density was found. Fluid flow experiments were conducted and the results showed that permeability increases with pore diameter and porosity of the medium, and the inertial coefficient depends only on porosity. They proposed a theoretical model for predicting inertial coefficient and a modified permeability model; the models were shown to agree with experimental results. Tadrist *et al.* [6] discussed the characteristics of randomly stacked fibers and metallic foams and analyzed the transport properties for both materials.

Convection in porous media has been widely investigated, but most studies focused on packed beds and granular materials with low porosities in the range 0.3-0.6. The porosity of open-cell metal foams is much higher ( $\varepsilon$ >0.90), and only during the past decade has convection in high-porosity metal foams started to receive attention. Calmidi and Mahajan [7] investigated forced convection in high-porosity metal foams experimentally and numerically. Experimental results showed that the transport enhancing effect of thermal dispersion is extremely low with foam-air combinations, but for foam-water combinations it can be very high. In the numerical study, a thermal non-equilibrium model was used and a Nusselt number correlation was determined. Zhao *et al.* [8] studied natural convection and its effect on overall heat transfer in highly porous open-cell FeCrAlY foams experimentally and numerically. Experimental results showed that natural convection is significant in metal foams due to the high porosity and inter-connected open cells. Numerical calculations showed that the so-called non-equilibrium effect (the metal and fluid being at different temperatures) cannot be neglected and hence a two-equation energy model should be used instead of one-equation model for convection in metal foams. Hetsroni *et al.* [9] studied natural convection heat transfer in metal foam strips with internal heat generation by experiments. Infrared images on both the surface and the inner region of the metal foam were analyzed, and the non-equilibrium temperature distribution was estimated. The result indicated that the non-equilibrium effect is significant.

Some studies have focused on metal-foam convective heat transfer devices. Boomsma *et al.* [10] studied an open-cell aluminum foam heat sink for electronics cooling applications. They found that compressed aluminum foams performed well, offering a significant improvement in the efficiency over several commercially available heat exchangers. They also found the metal foam can decrease the thermal resistance to nearly half that of currently used heat exchangers in the same application. Zhao *et al.* [11] and Lu *et al.* [12] analyzed forced convection heat transfer performance in high-porosity, open-cell, metal-foam-filled heat exchanger tubes and metal-foam-filled pipes using the Brinkman-extended Darcy momentum model and the two-equation heat transfer model for porous media. The results showed that, compared to conventional, finned-tube heat exchangers, the heat exchangers with metal-foam-filled tubes have better heat transfer performance, and the metal-foam-filled pipes have much better thermal performance than a plain tube, but at the expense of higher pressure drop. Mahjoob and Vafai [13] have discussed the effects of micro-structural metal foam properties on heat exchanger performance, and they categorized and investigated the extant correlations for flow and thermal transport in metal-foam heat exchangers. Tube and channel metal-foam heat exchangers were used to evaluate thermal-hydraulic performance, and the results showed a considerable improvement in performance by inserting the metal foam. Ejlali *et al.* [14] numerically investigated the fluid flow and heat transfer of an air-cooled metal-foam heat sink under a high speed laminar jet confined by two parallel walls at Reynolds numbers from 600 to 1000. They compared the performance of the metal-foam heat sink to that of conventional finned design and found that the heat removal rate can be greatly improved without additional cost. Dai *et al.* [15] presented the comparison of metal-foam heat exchangers to louver-fin heat exchangers based on the $\varepsilon$ -$N_{TU}$ method; the results showed that with the same thermal-hydraulic performance, the metal-foam heat exchanger can be lighter and smaller, but much more expensive.

As noted above, there are numerous studies of material properties and transport phenomena, and fewer studies of metal-foam heat sinks. However, thermal-hydraulic data which can be used to evaluate the use of metal foams in air-cooling heat exchangers are currently scanty in the literature. Previous evaluations of this technology have been limited by the dearth of experimental support for the evaluation. In this study, a closed-loop wind tunnel is used to get the needed thermal-hydraulic data for a metal-foam heat exchanger under dry conditions. The measurements are compared to the results calculated by some existing correlations from the open literature. Finally, some comments on some of the challenges that must be met for the adoption of this technology in air-cooling systems are made.

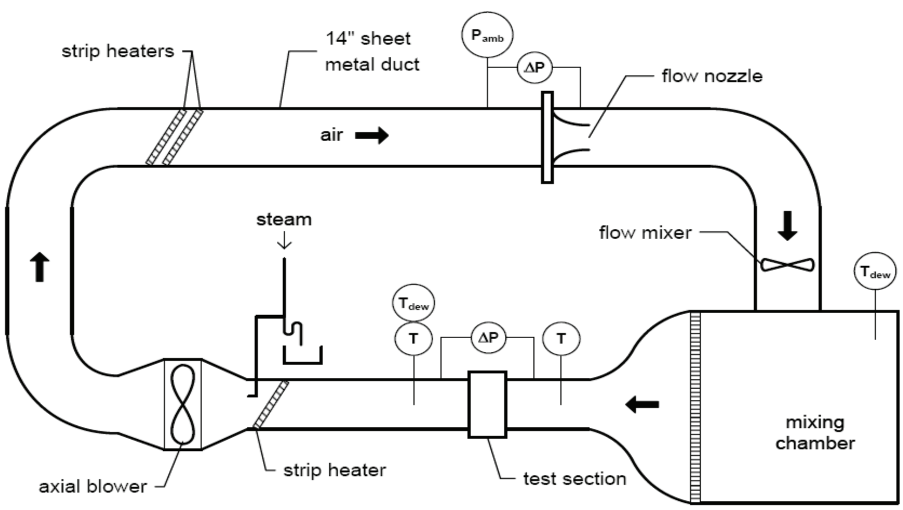
## *Experimental Setup:*

A closed-loop wind tunnel is used to assess the thermal-hydraulic performance of a heat exchanger having metal foam as the fin material. As shown in Figure 1, air downstream of test section passes through a set of electric strip heaters, past a steam injection pipe, through an axial blower and another set of strip heaters, through a flow nozzle, a mixing chamber, a flow conditioning section, a flow contraction, and the test section, completing the loop. A variac controller is used to maintain the desired upstream air temperature and dew point at steady state. Steam is generated by an electric humidifier. The air temperature is measured using thermopile grids, constructed using T-type thermocouples. Chilled-mirror hygrometers are used to measure the upstream and downstream dew points. The cross-sectional flow area in the test section is rectangular—30 cm wide and 20 cm high. An axial blower provides an air flow with face velocities at the test section from 0.5 to 5 m/s. An ASME flow nozzle, with a micro manometer, is used to measure air mass flow rate. Another micro manometer is used to measure air-side pressure drop across the test section. A single-phase liquid, an aqueous solution of ethylene glycol (DOWTHERM 4000), is used as the tube-side heat transfer fluid. A chiller system with a commercial heat pump, two large coolant reservoirs, a PID-controlled electric heater, and a gear pump supplies the flow. The chiller system provides a coolant flow with a steady inlet temperature (within 0.1°C) at a capacity up to 20 kW. Coolant inlet and outlet temperatures are measured using thermocouples imbedded through supply tubes. Coolant flow mixing devices are installed immediately upstream of the thermocouples to provide a well mixed flow and a uniform coolant temperature. A Coriolis-effect flow meter located in the downstream coolant pipe is used to measure mass flow rate. The significant experimental uncertainties involved in the dry and wet wind-tunnel experiments are listed in Table 1.

***Table 1: Uncertainty in measurement for various parameters***

| **Parameter** | **Uncertainty** |
|---|---|
| Air temperature | $\pm 0.1^0\ C$ |
| Coolant temperature | $\pm 0.1^0\ C$ |
| Nozzle discharge coefficient | $\pm 2\%$ |
| Core pressure drop | $\pm 0.17 Pa$ |
| Nozzle pressure | $\pm 0.17 Pa$ |
| Coolant mass flow rate | $\pm 0.1\%$ |
| Dew point Temperature | $\pm 0.1\%$ |

***Fig. 1: Closed loop wind tunnel***



Before beginning wind-tunnel experiments, the heat exchanger specimens are insulated using foam insulation tape. As the specimen has face dimensions different from those of the test section, it is necessary to install within the tunnel an additional flow contraction upstream and a diffuser downstream of the test specimen. The specimen is mounted in the test section, the coolant hoses connected, and the gaps between the specimen and the test section sealed with adhesive tape. The entire wind tunnel, the test specimen, steam pipes, and coolant pipes are insulated to isolate the system as much as possible from the environment. The thermal conductivity of the insulation material is 0.03 W/m-K.

## *Sample Specifications:*

The sample used for the experiments is a flat aluminum tube cross-flow heat exchanger, aluminum foam is sandwiched between tubes as fins. Each tube ends in a plastic manifold which distributes and collects coolant from flat tubes.

***Fig. 2: Metal foam heat exchanger test specimen***

The sample consists of ten metal foam layers (length 200 mm, width 15mm and depth 15 mm) sandwiched between flat aluminum tubes (length 304.8 mm, width 25.4 mm height 3.2 mm and wall thickness 0.5 mm). The face area of the heat exchanger is $200 \times 200m^2$ .Transparent manifolds connect these flat tubes. All the connections are sealed checked to avoid leakage problems. Important characteristic of the metal foam are summarized in the following.

***Table 2 Characteristics of the metal foam sample***

| Porosity (-) | PPI (-) | $d_f$ (mm) | $d_p$ (mm) | $ff$ (-) | $K$ ($\times 10^7 m^2$) | $k_{effective}$ (W/m k) |
|---|---|---|---|---|---|---|
| 0.9272 | 10 | 0.25 | 3.13 | 0.097 | 1.2 | 4.10 |

In order to join the metal foam to the aluminum base plates Arctic Silver 5 High-Density Polysynthetic Silver Thermal Compound was used, having a thermal conductivity of approximately 5 W/m-K.

## *Results and discussion:*

Figure (3) and (4) show the results obtained during the experimentation. The heat transfer coefficient based on the total surface area of the metal-foam fin and the base plate is plotted against the Reynold's number based on the ligament diameter.

$$\mathrm{Re}_{df} = \rho_m u d_f / \mu_m \qquad (1)$$

$$\frac{1}{UA} = \frac{1}{\eta_o h_a A_a} + R_{wall} + \frac{1}{h_c A_c} \qquad (2)$$

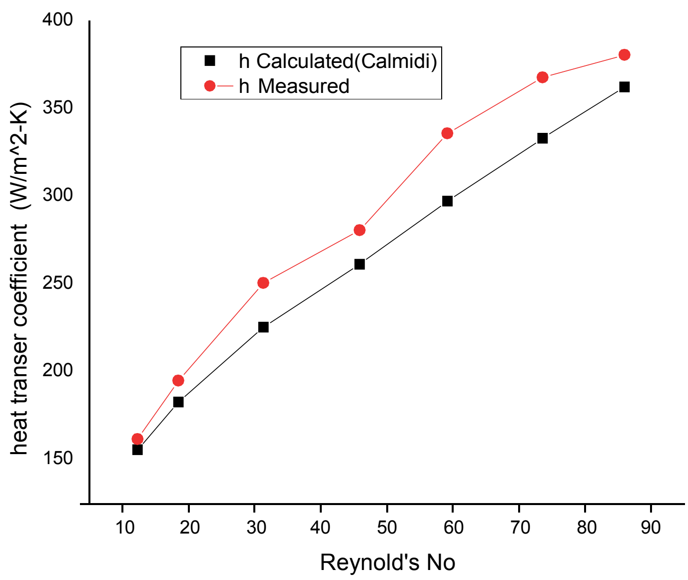
$$Q_{avg} = F \times UA\Delta T_{LMTD} \qquad (3)$$

$$Q_{avg} = \frac{Q_c + Q_a}{2} \qquad (4)$$

$$\Delta T_{LMTD} = \frac{(T_{h,i} - T_{c,i}) - (T_{h,o} - T_{c,o})}{\ln\left(\frac{(T_{h,i} - T_{c,i})}{(T_{h,o} - T_{c,o})}\right)} \qquad (5)$$

F is the correction factor for a single pass, cross flow heat exchangers[19].Fin efficiency is calculated based on the adiabatic tip condition.

***Fig. 3: Heat transfer coefficient (measured and calculated Calmidi and Mahajan, 2000)***



To compare the results, equation (2) provided by Calmidi and Mahajan[7] is used. The calculation performed using the relationship

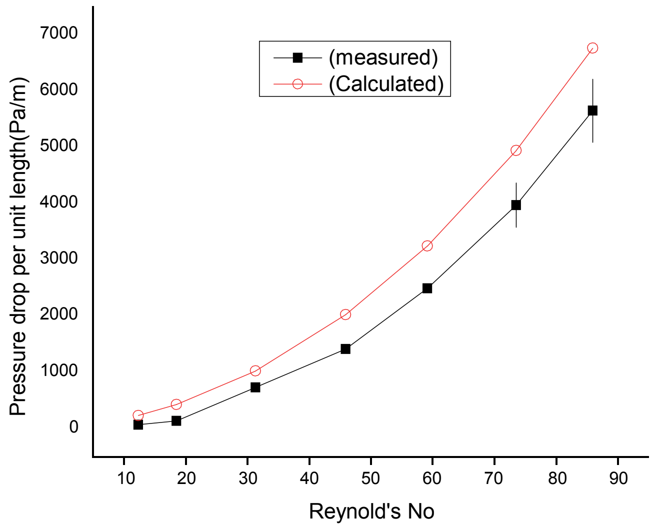
$$h = 0.52\,\mathrm{Re}_{d_f}^{0.5}\,\mathrm{Pr}^{0.37}\,\frac{k_a}{d_f} \qquad (6)$$

The experimental data are compared to the model of Calmidi and Mahajan, 2000 [7] in figure (3). The results show surprisingly good agreement and buttress both the data and the model. The relationship(6) showed that the values differ less than 5% at low Reynolds numbers(0.6 m/s) but the difference approaches 10 % at the highest velocity (4.3 m/s)

Shown in the figure (4) is the pressure drop The pressure drop for metal foam is assumed to be higher as compared to some other compact heat exchangers and similar behavior was achieved during the experiments. The fanning friction factor for metal foam with a face velocity of 3m/s is 0.15 which is 50% more than for multi- louver fin heat exchanger under the same flow conditions. The results are compared with equation (3) provided byBhattacharya *et al*.[5]

$$-\frac{dp}{dx} = \frac{\mu u}{K} + \frac{\rho_a ff}{\sqrt{K}} u^2 \qquad (7)$$

***Fig. 4: Pressure drop per unit length (measured and calculated* Bhattacharya *et al.* [5])**



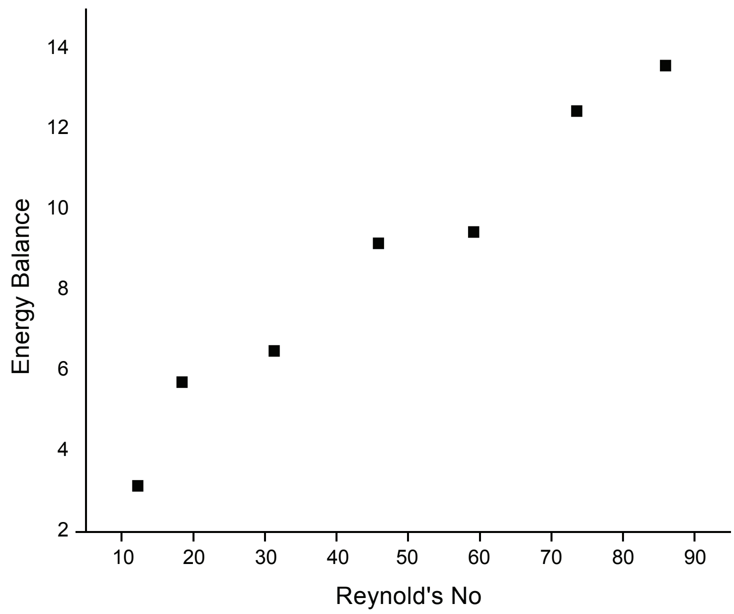

The relationship is widely used to estimate the pressure drop through porous media, but the experimental results show that the difference varies from 76% (213 Pa/m calculated 49 Pa/m measured) at low face velocity to 17 % (5629 Pa/m calculated 6739 Pa/m measured) at high face velocity. The uncertainty in the measuring device for pressure difference is

## *Uncertainty Analysis:*

The uncertainties in various instruments were reported in Table 1. While conducding the wind tunnel experiments the first critical step is to achieve the energy balance on the air side and on the coolant side. Such a balance supports the veracity of the data.. Even with significant care, these problems are unavoidable and resulting a slight heat transfer mismatch on air and coolant sides. The problems become rather severe at high face velocities. If the energy balance is defined by the following relationship, then its variation with Reynolds No based on the ligament diameter is given as follows (equation (8)).

$$\dot{Q}_{balance} = \left| \dot{Q}_{air} - \dot{Q}_{coolant} \big/ \dot{Q}_{avg} \right| \tag{8}$$

***Fig 5: Energy balance against Reynold's no.***




The trend in the energy balance show that as the air mass flow rate is increased, the problems like air leakage become more significant as compared to they were at low air mass flow rates. The uncertainty for the heat transfer coefficient varies from 5.7 % at lowest face velocity (0.6 m/s) to 13.7 % at the highest face velocity (4.3 m/s).

## *Future work and challenges:*

Current results are based on the experiments performed under dry conditions for just one specimen, which is actually a drop-in-replacement configuration for some current heat exchangers (louver fins are replaced with metal foam). In order to fully exploit the performance-enhancing characteristics of metal foams, further experimental work is required. For that purpose new samples with different configurations and having different foam properties need to be tested. Bonding the metal-foam fins to the base surface is perhaps the most challenging task while manufacturing and also the most costly process if one wants the ideal thermal joint (soldering or brazing). Thermal epoxies can also work well if they have good thermal conductivity, but the performance of the brazed metal foam should be tested along with the specimen employing thermal epoxy as adhesive. Furthermore the performance of the materials should be tested under wet conditions, where there is condensation on the surface, to see the water drainage behavior during operational conditions.

## *Conclusion:*

Metal foams are novel materials which can perform well if they are used in compact heat exchangers. For the two parameters, heat transfer performance and pressure drop, one has to come up with some ideal conditions in terms of geometry and flow conditions to get the best performance. There are correlations available to calculate the heat transfer coefficient and pressure drop. The relationship for the heat transfer coefficient by Calmidi and Mahajan [2] is accurate enough as the experimental values are close enough. For the pressure drop performance the relationship by Bhattacharya [5] work well at high velocities.

## *Nomenclature:*

$ff$ inertial coefficient
$\eta_o$ fin efficiency
$h$ convective heat transfer coefficient, W/m$^2$K
$Pr$ prendtl No
$K$ permeability, m$^2$
$k$ thermal conductivity of the fin, W/m.K
$u$ velocity, m/s
$\mu$ dynamic viscosity, N/ms
$d_f$ fiber diameter,m
$d_p$ pore diameter,m
$\rho$ density, kg/m$^3$

## *Subscript:*

$a$ air side
$c$ coolant side
$m$ mean value
$h,i$ hot inlet
$h,o$ hot outlet
$c,i$ cold inlet
$c,o$ cold outlet

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