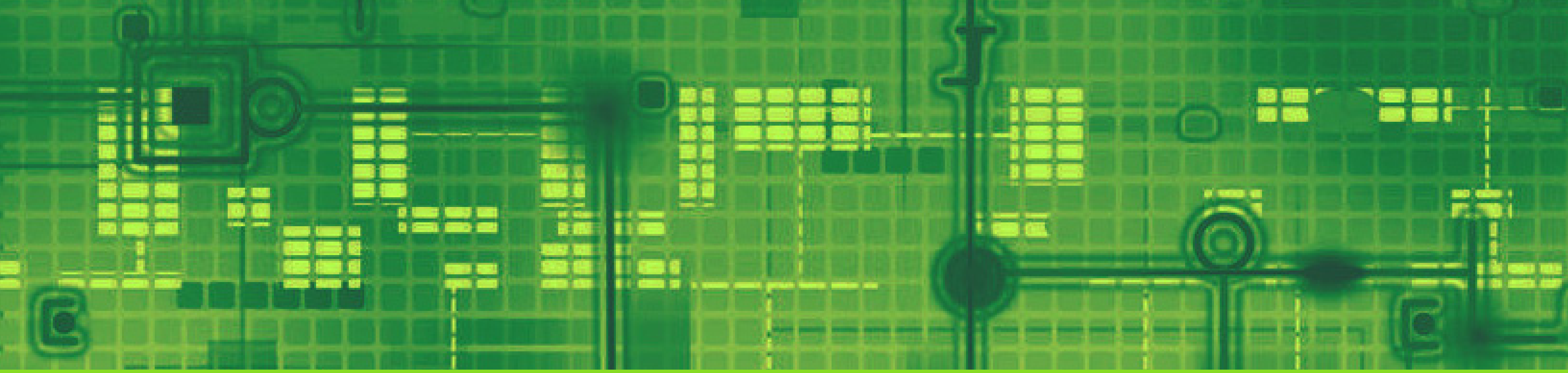

Heat Transfer

in Metal Foam Heat Exchangers

Metal foams have found increasing use in thermal management. Their surface area is much larger than regular finned heat sinks, making the available surface area for heat transfer attractive. On the other hand, their increased pressure drop and interfacial resistance makes them vulnerable to not being useful for some applications. Understanding the underlying physics behind the mesh heat sink and heat exchangers is a necessity for using them effectively in thermal management. Savery [1] used COMSOL [2], which is a commercial CFD finite element software to simulate the natural convection in a metal foam heat sink. Before embarking on presenting the equations that govern the flow and heat transfer in porous media, it is helpful to understand the characteristics of porous media.

A porous material is defined by its porosity and permeability factors. Porosity is the percent of the material that is void. The larger the porosity, the more open area in the material. Permeability is the measure of the ability of the material to let the fluid pass through it. It is essentially the hydraulic resistance of the porous material.

Figure 1 shows two different materials. The one on the left has a porosity of 4 times the one on the right. But, they both have the same permeability, because the 3 holes shaded on the material on the left have been blocked off.

Figure 2 shows two different materials that have the same porosity, because the area of the 4 smaller circles on the left is the same as the area of the large circle on the right. But the 3 small circles on the left have been blocked off, so the resistance, and hence the permeability, of the material on the left is lower than the material on the right.

The governing equation for the energy equation of a flow through a porous material is described as [3]:

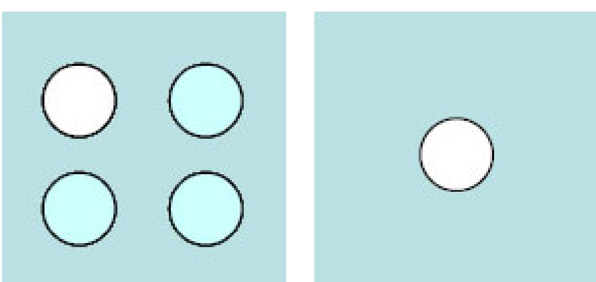

Figure 1. Same Permeability, Different Porosity [1]

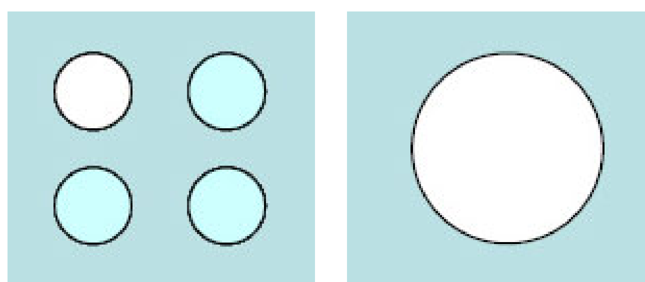

Figure 2. Thermaltake Xpressar Refrigeration System Schematic [2]

$$(\rho C_p)\frac{\partial T}{\partial t} + \rho C_p u \Delta T = \Delta(K_{eq}\Delta T) + \varphi$$

The difference from the simple flow is the replacement with equivalent density, conductivity and heat capacity defined below:

ρ = density (Kg/m^3)
C_p = Heat capacity (J/Kg.K)
T = Temperature (K)
t = Time (sec)
K = Thermal conductivity(W/m.K)
φ = Viscous dissipation
P= Porous material
F = Fluid
θ_P= Porous material volume Fraction
θ_F = Fluid volume fraction

$$K_{eq} = \theta_p K_p + \theta_F + K_F$$

$$(\rho C_p)_{eq} = \theta_P \rho_P C_{p_P} + \theta_F \rho_F C_{p_F}$$

$$\theta_P + \theta_F = 1$$

The momentum equation for the flow in a porous media is the modified Darcy-Brinkman equation as stated in[3]. The geometry of the computational domain analyzed is shown in figure 3.

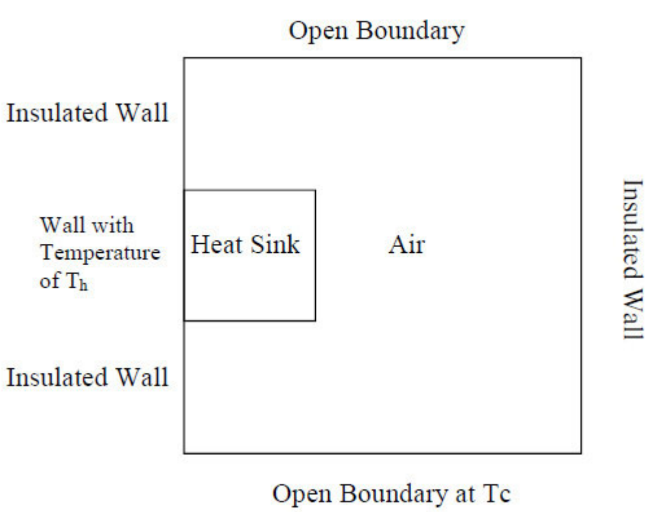

Figure 3. Schematic of the Computational Domain [1]

The following assumptions were made for running the CFD:

- 2-Dimensional
- Steady state
- 50x50 mm domain
- 10x10 mm heat sink
- Inlet and outlet boundaries open
- The left wall was kept at a constant temperature of 310K.

The cases for the heat sink was run for a porous heat sink and a solid heat sink. The case without heat sink was also run to compare the results. Ten different porosities and 3 different permeabilities were simulated to analyze the effect of different parameters. Table 1 shows the results for different combinations of porosity and permeability. The heat removal rate and volumetric flow rate for these combinations are shown in the table. It is to be noted that some of the combinations are purely hypothetical and may not exist in practice.

Form	Porosity	Permeability	Heat Removal Rate (W/m)	Volumetric Flow Rate (m^2/s)
1a	0.05	1.00E-08	3.1391	0.0011
1b	0.1544	1.00E-08	3.1443	0.0011
1c	0.2589	1.00E-08	3.1456	0.0011
1d	0.3633	1.00E-08	3.1452	0.0011
1e	0.4678	1.00E-08	3.1443	0.0011
1f	0.5722	1.00E-08	3.1429	0.0011
1g	0.6767	1.00E-08	3.1406	0.0011
1h	0.7811	1.00E-08	3.1361	0.0011
1i	0.8856	1.00E-08	3.1233	0.0011
1j	0.99	1.00E-08	2.8752	0.001
2a	0.05	5.05E-07	7.6455	0.002
2b	0.1544	5.05E-07	11.5721	0.0027
2c	0.2589	5.05E-07	10.6431	0.0025
2d	0.3633	5.05E-07	10.04	0.0024
2e	0.4678	5.05E-07	9.7644	0.0024
2f	0.5722	5.05E-07	9.6461	0.0024
2g	0.6767	5.05E-07	9.585	0.0024
2h	0.7811	5.05E-07	9.5285	0.0024
2i	0.8856	5.05E-07	9.4072	0.0023
2j	0.99	5.05E-07	7.5302	0.0021
3a	0.05	1.00E-06	7.9756	0.002
3b	0.1544	1.00E-06	14.0391	0.0031
3c	0.2589	1.00E-06	14.9183	0.0033
3d	0.3633	1.00E-06	15.1979	0.0033
3e	0.4678	1.00E-06	15.1892	0.0033
3f	0.5722	1.00E-06	15.0854	0.0033
3g	0.6767	1.00E-06	14.9662	0.0033
3h	0.7811	1.00E-06	14.8254	0.0033
3i	0.8856	1.00E-06	14.543	0.0033
3j	0.99	1.00E-06	10.7887	0.0028

Table 1. Heat Removal Rate and Volumetric Flow Rate Per Unit Width for Different Combinations of Porous Materials [1]

Figure 4 shows the heat removal of a porous material as a function of porosity for different permeabilities. The figure shows that near the extremes, the heat removal decreases fast. In the case of very low porosity on the left, it resembles a solid block. The solid block does not allow the flow to go through the heat sink; hence, no convection heat transfer occurs. On the other hand, at very high values of porosity at the extreme right, the conductivity of the porous material reaches the conductivity of the fluid, which is much lower; hence, minimizing the heat transfer rate. The blue line for permeability of 10^{-8} m^2 shows the heat removal rate of a solid fin, which is about 3.1 W/m per unit width. It is evident from the figure that, at very low permeabilities, let's say around 10^{-6}, the heat removal has increased to 15 W/m, which is higher than the solid fin by a factor of 5. This figure also shows that porosity does not have much effect on the heat removal capability far from the extremes.

Figure 5 shows the heat removal capacity as a function of permeability for different porosities. The figure shows that the heat removal capacity increases with an increase of permeability. The bottom line is the heat rate curve for a plane wall at 1.5 W/m per unit width. The second line from the bottom is the heat rate curve for a solid fin block at 2.9 W/m heat rate. The calculated results show a drastic improvement in heat transfer for a foam heat sink.

Figure 6 shows the heat removal rate as a function of permeability. The figure shows that, after a

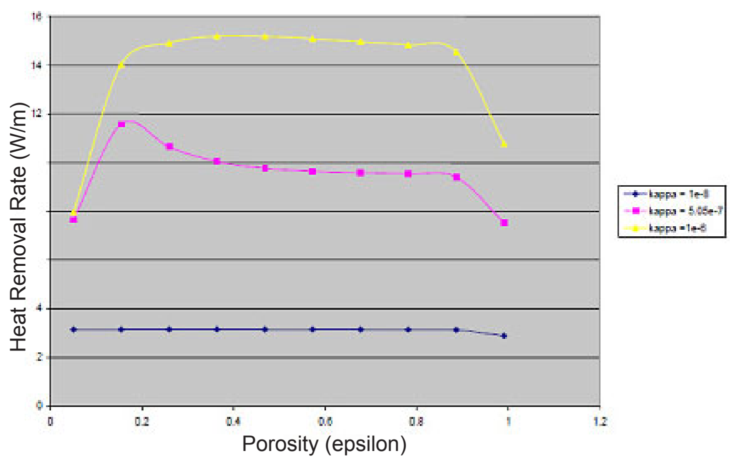

Figure 4. Heat Removal as a Function of Porosity for Different Permeabilities [1]

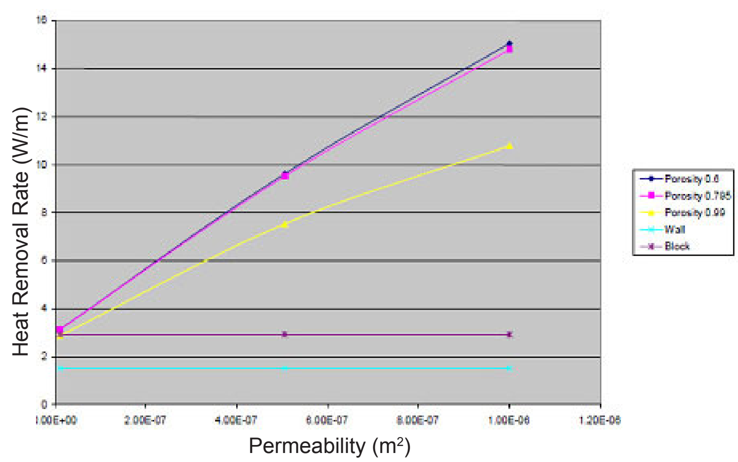

Figure 5. Heat Removal as a Function of Permeability for Different Porosities [1]

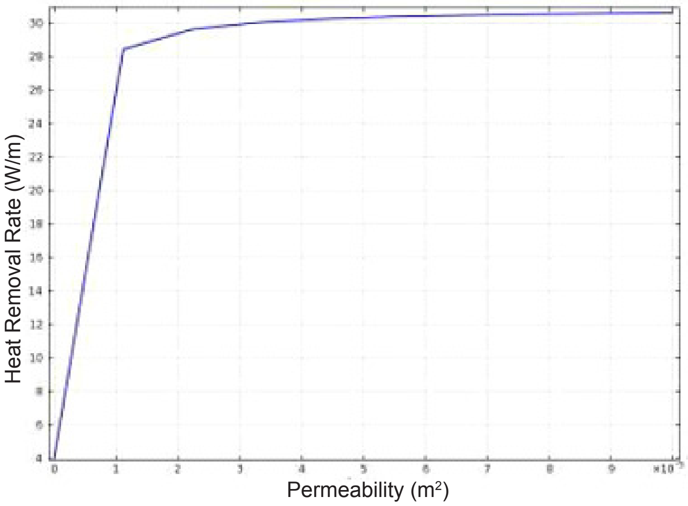

Figure 6. Heat Removal Rate as a Function of Permeability [1]

permeability of 2 m^2, the curve has reached a plateau and increasing the permeability further does not affect the heat rate. A graph of volumetric flow rate, as a function of permeability, shows that the flow rate reaches a maximum at K=2 m^2; hence, the heat rate stays the same. It should be noted that there is a difference between porosity and permeability. High values of permeability do not mean that the conductivity of the material has been compromised. It only shows that the material has less resistance to the fluid flow through it.

In another experiment, Haak et. al [4] conducted experiments on a high temperature iron-based alloy foam. The experiment was done on two pieces of 127 x 127 x 12 mm foam bonded to a copper sheet (1 mm thick) using brazing technology. Figure 7 shows the average Nusselt number as a function of the Reynolds number, based on the permeability defined as :

$$Re_K = \rho U \sqrt{K} / \mu$$

Where U is the velocity of the flow in the porous material and K is the permeability.

The quantity relative density is defined as the ratio between the density of the foam material to the density of the solid material. This figure shows that at the same ppi (pores per inch), increasing the relative density increases the average Nusselt number. Figure 8 shows that, at a relative density fixed at 15%, a 60 ppi foam has a higher Nusselt number than the 10 ppi foam. But this happens at the expense of higher pumping power. In fact, the authors claim that, at the same pumping power, the highest Nusselt number obtained for the 10 ppi, 15% relative density is twice that obtained from the 60 ppi, 15% relative density. The larger hole size creates a lesser pressure drop; hence, less pumping power. This figure also shows that the 10 ppi foam with 15% relative density has a higher Nusselt number than the 30 ppi, 10% relative density. The data shows that relative density is more important than ppi.

More data and experimentation, especially for forced convection, is needed to obtain a better handle on the design of foam heat sinks. The data will be more meaningful if the Nusselt numbers are plotted as a function of pumping power rather than

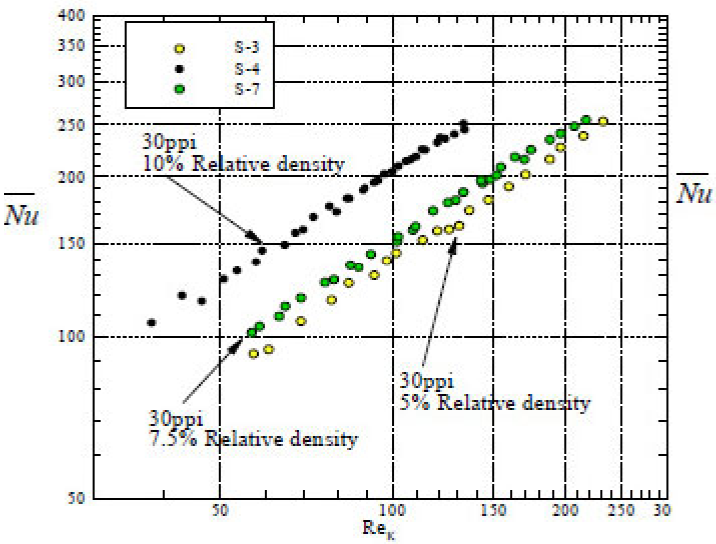

Figure 7. Average Nusselt Number as a Function of Reynolds Number for Different Porosities and Relative Densities. [4]

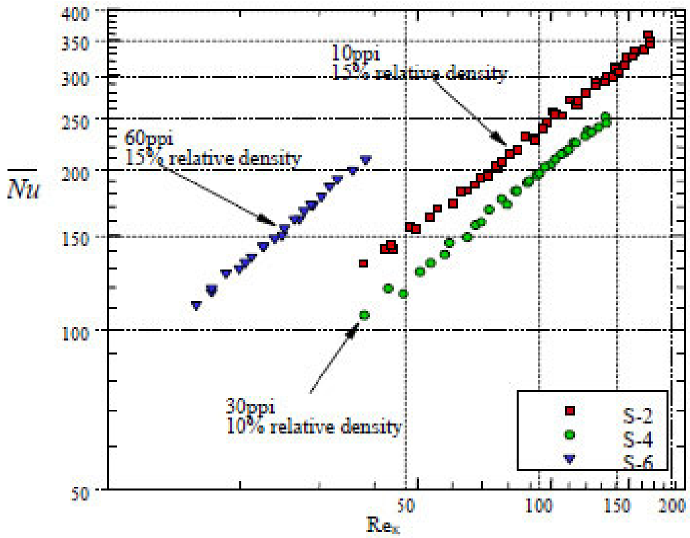

Figure 8. Average Nusselt Number as a Function of Reynolds Number for Different Porosities and Relative Densities. [4]

as a Reynolds number. The foam heat sinks show promising results in certain applications, such as high performance heat exchangers, assuming the pumping power and the fouling are resolved.

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