11th International Research/Expert Conference
”Trends in the Development of Machinery and Associated Technology”
TMT 2007, Hammamet, Tunisia, 05-09 September, 2007.

THREE DIMENSIONAL HEAT TRANSFER ANALYSIS OF ALUMINIUM METAL FOAM HEAT EXCHANGERS

Kemal Altinişik*, Ahmet Ali Sertkaya*, Alper Altinişik, Şenay Yalçin*****
***Selcuk University, Engineering Faculty, Konya, Turkey**
**** Uzel Mac. İnd., İstanbul, Turkey**
***** Bahcesehir University, Engineering Faculty, İstanbul, Turkey**

ABSTRACT
High porosity metal foams are excellent candidates for high performanced heat exchangers. They are employed in aerospace applications mainly, but their use has been widened to include cooling in Electronic Packaging. Other important applications have been found taking advantage of the thermal properties of the metal foam. These applications include compact heat exchangers for airborne equipment, regenerative and dissipative air-cooled condenser towers, and compact heat sinks for power electronics. The low relative density, open porosity and high thermal conductivity of the cell edges, as well as the large accessible surface area per unit volume, and the ability to mix the cooling fluid, all make metal foam heat exchangers efficient, compact and light weight.
The purpose of this study is to devolop the three dımensıonal heat transfer model for aluminium metal foam heat exchangers. The forced convection and the conduction heat transfer modes for the heat transfer distribution in open cell aluminium metal foam have been concidered. The analysis uses the typical parameters reported by the foam manufacturers such as the porosity and the area density, defined as the ratio of the surface area of the foam to the volume. The simplicity and applicability of the present approach offer a significant advantage over previous models. It eliminates the need for complex microscopic analytical or numerical modeling of the flow and the heat transfer in and around the pores.
Keywords: Aluminium metal foam, heat exchanger , heat transfer

1. INTRODUCTION

Bastawros (1) demostrated the efficiency of metallic foams in forced convection heat removal. Bastawros showed that a high perfoormance cellular aluminum heat sink removed 2-times the usual heat flux removed by a pin-fin array, at a third of weight and with only a moderate increase in the pressure drop. An important application of metal foams, as it has been mentioned in multiple occasions, is to construct compact heat exchangers. An interesting study carried out by Boomsma et al.(2) where open-cell aluminum foams were compressed by various factors and the fashioned in to heat exchangers for electric cooling applications, which dissipate large amount of heat. All the literature reviewed is about thermal and hydraulic characerization of porous media (3,4,5,6,7,8,9,10,11,12). Boomsma, K. And Poukakos, D., (13) developed a one-dimensional heat conduction model for open celled metallic foams. It was based on a three dimensional description of the foam geometry. Dukhan and Quinones (14) used a one-dimensional heat transfer model for open cell metal foam. Aluminium foams with different areas, relative densities, filament diameters and number of pores per inch were analyzed.

All the literature reviewed is about thermal and hydraulic characerization of porous media. Generally, the forced convection and the conduction heat transfer modes for the heat transfer distribution in open cell aluminium metal foam have been considered. However, at the moment there are not any universal correlations to characterize the porous media due to the great variability of geometry and the materials of which they are constructed. The base material of the porous media can be metallic and non-metallic. In this study, aluminium as the base material of the porous media has been used, and the thermal and hydraulic behaviour of this material have been examined.

2. THEORY

In order to do the heat transfer analysis of the aluminium foam the control volume shown in Figure 1, where width of the foam dx, dy and dz are small thickness in the x, y and z directions respectively has been considered.

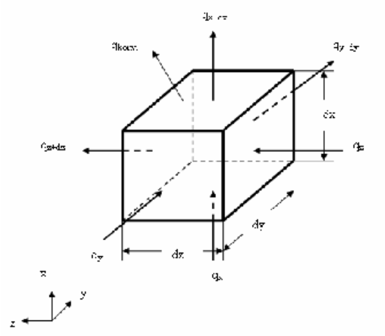

Figure 1. The control volume for the aluminum foam

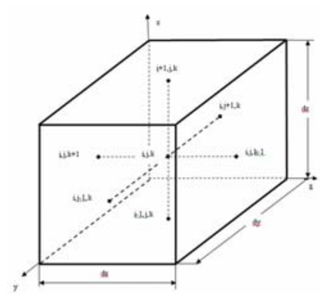

Figure 2. Finite difference form for the control volume

The law's Conservation of energy has been applied to the control volume. For the energy fluxes, the conduction and convection heat transfer have been considered. The following equation for the energy balance can be written

$$q_x+q_y+q_z = q_{x+dx}+q_{y+dy}+q_{z+dz}+q_{conv.} \tag{1}$$

The Fourier 's law of heat conduction and Newton 's law of heat losses for the aluminium foam having the solid parts and holes are given as follows

$$q_x = -k_s\, dy\, dz\, (1-\varepsilon) \frac{\partial T_{fm}}{\partial x} - k_f\, dy\, dz\, \varepsilon \frac{\partial T_{fm}}{\partial x} \tag{2}$$

$$q_y = -k_s\, dx\, dz\, (1-\varepsilon) \frac{\partial T_{fm}}{\partial y} - k_f\, dx\, dz\, \varepsilon \frac{\partial T_{fm}}{\partial y} \tag{3}$$

$$q_z = -k_s\, dx\, dy (1-\varepsilon) \frac{\partial T_{fm}}{\partial z} - k_f\, dx\, dy\, \varepsilon \frac{\partial T_{fm}}{\partial z} \tag{4}$$

$$q_{x+dx} = -k_s dydz(1-\varepsilon)\frac{\partial T_{fm}}{\partial x} - k_s dxdydz(1-\varepsilon)\frac{\partial^2 T_{fm}}{\partial x^2} - \varepsilon k_f dydz\frac{\partial T_{fm}}{\partial x} - \varepsilon k_f dxdydz\frac{\partial^2 T_{fm}}{\partial x^2} \tag{5}$$

$$q_{y+dy} = -k_s dxdz(1-\varepsilon)\frac{\partial T_{fm}}{\partial y} - k_s dxdydz(1-\varepsilon)\frac{\partial^2 T_{fm}}{\partial y^2} - \varepsilon k_f dxdz\frac{\partial T_{fm}}{\partial y} - \varepsilon k_f dxdydz\frac{\partial^2 T_{fm}}{\partial y^2} \tag{6}$$

$$q_{z+dz} = -k_s dxdy(1-\varepsilon)\frac{\partial T_{fm}}{\partial z} - k_s dxdydz(1-\varepsilon)\frac{\partial^2 T_{fm}}{\partial z^2} - \varepsilon k_f dydz\frac{\partial T_{fm}}{\partial z} - \varepsilon k_f dxdydz\frac{\partial^2 T_{fm}}{\partial z^2} \tag{7}$$

and for the solid parts and holes, the following equations are written

$$A_{cond}=A_c-A_p \tag{8}$$

$$A_{cond}=A_c(1-\varepsilon) \tag{9}$$

$$dA_{conds}=(1-\varepsilon)\, dy\, dz \tag{10}$$

$$dA_{condf}=\varepsilon\, dy\, dz \tag{11}$$

$$dA_{conds}=(1-\varepsilon)\, dx\, dz \tag{12}$$

$$dA_{condf}=\varepsilon\, dx\, dz \tag{13}$$

$$dA_{conds}=(1-\varepsilon)\, dx\, dy \tag{14}$$

$$dA_{condf}= \varepsilon\, dx\, dy \tag{15}$$

ε is surface porosity and it can be given as follows, $\varepsilon = 1-\left(\frac{V_s}{V_{tot}}\right)$ (16)

The heat transfer by convection is $q_{conv.}=h_{fm}\, \sigma\, dx\, dy\, dz\, (T_{fm}-T_\infty)$ (17)

and the following expression for **σ** is given by the producer firms. The **σ is**

$$\sigma = \frac{A_{conv.}}{dx\, dy\, dz} \tag{18}$$

If the expression given above are substituted into the eqution (1), the following form can be obtained

$$k_s (1-\varepsilon)\left(\frac{\partial^2 T_{fm}}{\partial x^2}+\frac{\partial^2 T_{fm}}{\partial y^2}+\frac{\partial^2 T_{fm}}{\partial z^2}\right)+\varepsilon\, k_f\left(\frac{\partial^2 T_{fm}}{\partial x^2}+\frac{\partial^2 T_{fm}}{\partial y^2}+\frac{\partial^2 T_{fm}}{\partial z^2}\right)-h_{fm}\, \sigma\, (T_{fm}-T_\infty) =0 \tag{19}$$

or

$$\nabla^2 T_{fm} = \frac{h_{fm}\sigma}{k_s(1-\varepsilon)+\varepsilon k_f}(T_{fm}-T_\infty) \tag{20}$$

For the constant values in equation (20), the following description can be done

$$m^2_{fm}=\frac{h_{fm}\sigma}{k_s(1-\varepsilon)+\varepsilon k_f} \quad (21)$$

According to this description, the equation (20) takes the following form

$$\nabla^2 T_{fm} = m^2_{fm}(T_{fm}-T_\infty) \quad (22)$$

or $$\frac{\partial^2 T_{fm}}{\partial x^2}+\frac{\partial^2 T_{fm}}{\partial y^2}+\frac{\partial^2 T_{fm}}{\partial z^2} - m^2_{fm}(T_{fm}-T_\infty)=0 \quad (22a)$$

In order to put the dimensionless form, the following expressions can be written

$$X=\frac{x}{L}, \qquad Y=\frac{x}{L}, \qquad Z=\frac{z}{L} \quad (23)$$

For dimensionless temperature is $\theta_{fm}=\frac{T_{fm}-T_\infty}{T_b-T_\infty}$ (24)

Here, T_{fm} is $T_{fm}=\theta_{fm}(T_b-T_\infty)+T_\infty$ (25)

L is the length of metal foam and M^2 is the dimensionless constant. According to this definition for M^2 the following equation can be written

$$M^2=L^2 m^2_{fm} \quad (26)$$

After doing the necessery arrangements, the result equation takeds the following form

$$\frac{T_b-T_\infty}{L^2}\left(\frac{\partial^2\theta}{\partial X^2}+\frac{\partial^2\theta}{\partial Y^2}+\frac{\partial^2\theta}{\partial Z^2}-m^2_{fm}L^2\theta\right)=0 \quad (27)$$

or $$\frac{T_b-T_\infty}{L^2}(\nabla^2\theta-M^2\theta)=0 \quad (28)$$

or $$\nabla^2\theta-M^2\theta=0 \quad (29)$$

This equation can be rewritten as follows,

$$\frac{\partial}{\partial X}\left(\frac{\partial\theta}{\partial X}\right)+\frac{\partial}{\partial Y}\left(\frac{\partial\theta}{\partial Y}\right)+\frac{\partial}{\partial Z}\left(\frac{\partial\theta}{\partial Z}\right)-M^2\theta=0 \quad (30)$$

For solution this expression, the control volume given in Figure 1 can be considered as given in Figure 2. The expression (30) can be given in integral form as follows

$$\int_{k-1}^{k+1}\int_{j-1}^{j+1}\int_{i-1}^{i+1}\frac{\partial}{\partial X}\left(\frac{\partial\theta}{\partial X}\right)dXdYdZ+\int_{k-1}^{k+1}\int_{i-1}^{i+1}\int_{j-1}^{j+1}\frac{\partial}{\partial Y}\left(\frac{\partial\theta}{\partial Y}\right)dYdXdZ$$

$$+\int_{i-1}^{i+1}\int_{j-1}^{j+1}\int_{k-1}^{k+1}\frac{\partial}{\partial Z}\left(\frac{\partial\theta}{\partial Z}\right)dZdYdX-\int_{k-1}^{k+1}\int_{j-1}^{j+1}\int_{i-1}^{i+1}M^2\theta\, dXdYdZ=0 \quad (31)$$

If the boundary conditions are considered and after that the eqution is integrated, the following statement can be obtained

$$\left[\frac{d\theta}{dX}\Big|_{i+1}-\frac{d\theta}{dX}\Big|_{i-1}\right]\Delta Y\Delta Z+\left[\frac{d\theta}{dY}\Big|_{j+1}-\frac{d\theta}{dY}\Big|_{j-1}\right]\Delta X\Delta Z$$

$$+\left[\frac{d\theta}{dZ}\Big|_{k+1}-\frac{d\theta}{dZ}\Big|_{k-1}\right]\Delta X\Delta Y-\int_{k-1}^{k+1}\int_{j-1}^{j+1}\int_{i-1}^{i+1}M^2\theta\, dXdYdZ=0 \quad (32)$$

For the numerical derivative the following equation is given

$$\left[\frac{\theta_{i+2}-\theta_i}{(\delta X)_{i+1}}-\frac{\theta_i-\theta_{i-2}}{(\delta X)_{i-1}}\right]\Delta Y\Delta Z+\left[\frac{\theta_{j+2}-\theta_j}{(\delta Y)_{j+1}}-\frac{\theta_j-\theta_{j-2}}{(\delta Y)_{j-1}}\right]\Delta X\Delta Z$$

$$+\left[\frac{\theta_{k+2}-\theta_k}{(\delta Z)_{k+1}}-\frac{\theta_k-\theta_{k-2}}{(\delta Z)_{k-1}}\right]\Delta X\Delta Y-\int_{k-1}^{k+1}\int_{j-1}^{j+1}\int_{i-1}^{i+1}M^2\theta\, dXdYdZ=0 \quad (33)$$

For the last term in equation (33), the expression given below is written

$$\int_{k-1}^{k+1}\int_{j-1}^{j+1}\int_{i-1}^{i+1}M^2\theta\, dXdYdZ=\overline{S}\,\Delta X\Delta Y\Delta Z \quad (34)$$

Here, $\overline{S}$ is the mean value of linearized source term and it is given as follows

$$\overline{S}=Sc+Sp\,\theta \quad (35)$$

Here, Sc is the constant value of lineraized source term and Sp is the coefficient of temperature. In this case, equation (33) takes the following form

$$\left[\frac{\theta_{i+2}-\theta_i}{(\delta X)_{i+1}}-\frac{\theta_i-\theta_{i-2}}{(\delta X)_{i-1}}\right]\Delta Y\Delta Z+\left[\frac{\theta_{j+2}-\theta_j}{(\delta Y)_{j+1}}-\frac{\theta_j-\theta_{j-2}}{(\delta Y)_{j-1}}\right]\Delta X\Delta Z$$

$$+\left[\frac{\theta_{k+2}-\theta_k}{(\delta Z)_{k+1}}-\frac{\theta_k-\theta_{k-2}}{(\delta Z)_{k-1}}\right]\Delta X\Delta Y-(Sc+Sp\ \theta)\Delta X\Delta Y\Delta Z=0 \tag{36}$$

In order to make suitable for the computer the equation (36) can be written in compact form as follows

$$a_{Pi,j,k}.\theta_{i,j,k}=a_{Ei,j,k}.\theta_{i+2,j,k}+a_{W\,i,j,k}.\theta_{i-2,j,k}+a_{N\,i,j,k}.\theta_{i,j+2,k}+a_{S\,i,j,k}.\theta_{i,j-2,k}+a_{T\,i,j,k}.\theta_{i,j,k+2}+a_{Bi,j,k}.\theta_{i,j,k-2}+b \tag{37}$$

$$a_{P\,i,j,k}=a_{E\,i,j,k}+a_{W\,i,j,k}+a_{N\,i,j,k}+a_{S\,i,j,k}+a_{T\,i,j,k}+a_{B\,i,j,k}+Sp \tag{38}$$

3. CONCLUSION

This sudy was realized theoritically. The mathematical model was made suitable for computer and the program written was run. The experimental study related to this research has been gone on. In numerical study, it was found that the best way to find the temperature distribution inside of the foam is to set the boundary condition where the air leaves the sample as a zero slope condition. This means that the temperature is increasing along the z direction and it reaches its maximum value at the boundary where the air leaves the foam. We would like to propose this work to be applied to practical situtions where the metal foam are strongly used in many kind of heat exchangers.

4. REFERENCES

[1] Bastawros, A. F., 1998, "Effectiveness of Open-Cell Metallic Foams for High Power Electronic Cooling", Proceeding Symposium on the Thermal Management of Electronics, IMECE. Anaheim, CA.

[2] Boomsma, K., Poulikakos, D. and Zwick, F., 2003, "Metal Foams as Compact High Performance Heat Exchangers", Mechanics of Materials. Vol. 35, pp. 1161-1176.

[3] Zhao, C.Y., Lu, T.J., Hodson, H.P. and Jackson, J.D., 2004, "The Temperature Dependence of Effective Thermal Conductivity of Open-Celled Steel Alloy Foams", Vol. 367, pp. 123-131.

[4] Vafai, K. and Tien, C. L., 1982, "Boundary and Inertia Effects on Convective Mass Transfer in Porous Media", International Journal of Heat and Mass Transfer. Vol. 25, pp. 1183-1190.

[5] Writz, R. A., 1997, "A Semi-Empirical Model for Porous Media Heat Exchanger Design", Proceedings of the American Society of Mechanical Engineers National Heat Transfer Conference, Baltimore, MD.

[6] Lu, T.J., Stone, H.A. and Ashby, M.F., 1998, "Heat Transfer in Open Cell Metal Foams", Acta Metallurgica Inc. Vol. 46, No. 10, pp. 3619-3635.

[7] Kiwan, S. and Al-Nimr, M.A., 2001, "Using Porous Fins for Heat Transfer Enhancement", Journal of Heat Transfer. Vol. 123, pp. 790-795.

[8] Younis, L.B. and Viskanta, R., 1993, "Experimental determination of the Volumetric Heat Transfer Coefficient Between Stream of Air and Ceramic Foam", International Journal of Heat and Mass Transfer. Vol. 36, pp. 1425-1434.

[9] Decker, S., Möβbauer, S., Nemoda, S., Trimis, D. and Zapf, T., 2000, "Detailed Experimental Characterization and Numerical Modeling of Heat and Mass Transport Properties of Highly Porous Media for Solar Receivers and Porous

[10] Burners", Lehrstuhl für Strömungsmechanik Universität Erlangen-Nürnberg Cauerstr. 4, D-91058 Erlangen, Germany. 102

[11] Phanikumar M. S. and Mahajan R.L., 2002, "Non-Darcy Natural Convection in High Porosity Metal Foams", International Journal of Heat and Mass Transfer. Vol. 45, pp. 3781-3793.

[12] Pan, H.L., Pickenacker, O., Pickenacker, K., Trimis, D., Moβbauer, S. Wawrzinek, K. and Weber, T., 1999, "Experimental Determination of Effective Heat Conductivities of Highly Porous Media", Institute of Fluid Mechanics,

[13] University of Erlangen-Nuremberg Cauerstr. 4, D-91058 Erlangen, Germany.

[14] Bastawros, A.F., Evans A.G. and Stone H.A., 1999, "Evaluation of Cellular Metal Heat Transfer Media", Division of Engineering and Applied Sciences. Harvard University. Cambridge, MA 02138.

[15] Boomsma, K. and Poulikakos, D., 2000, "On the Effective Thermal Conductivity of a Three-Dimensionally Structured Fluid-Saturated Metal Foams", International Journal of Heat and Mass Transfer. Vol. 44, pp. 827-836.

[16] Dukhan, N. and Quinones, P., 2003, "Convective Heat Transfer Analysis of Open Cell Metal Foam for Solar Air Heaters", Proceeding International Solar Energy Conference, Hawaii, USA.