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Different Heat Exchanger Options for Natural Draft Cooling Towers

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ABSTRACT

This paper examines the application of different heat transfer enhancement mechanisms to improve the performance of an air-cooled heat exchanger. Specific attention is paid to a natural draft cooling tower which is to be applied as the air-cooled condenser of a geothermal power plant. The conventional method of extending the heat transfer area by means of extruding fins is compared with a modern technique being the application of a metal foam heat exchanger applied as a layer to the outer surface of the tube. Both designs improve the heat transfer rate from the phase-change fluid flowing in the tube bundle albeit at the expense of a higher pressure drop compared to the bare tube as our reference case. Considering the heat transfer enhancement as the benefit and the excess pressure drop as the cost, the two cases are compared against each other. In order to achieve this goal, three different software have been implemented and the results were cross-validated. These include two commercially available software being CFD-ACE and ASPEN B-JAC as well as a FORTRAN code developed in QGECE. A number of correlations are also proposed to predict the cost (excess pressure drop) and the benefit (heat transfer augmentation) of the extended surface.

1. INTRODUCTION

Emission-free electrical power from renewable sources is attracting increasing attention. Hot rock geothermal energy is one of the options being considered in Australia and other places for renewable emission-free base-load generation. Unfortunately, most of the geothermal resources are located in arid areas where there is not enough water to feed wet cooling towers to absorb the cycle waste heat. Air-cooled condensers provide the only economic choice in such places. In such condensers, the cycle fluid condenses inside tubes cooled by air. The tubes have external fins to increase the air-side heat exchange surface. Fins improve the heat transfer performance but at the same time lead to higher pressure drop compared to bare tubes. Hence, as a relatively old engineering practice, the designer has to consider the tradeoff between these two opposing effects. The problem, however, becomes more complicated in case of geothermal power plants where due to their relatively low efficiencies the waste heat generated by them, per MW generated electricity, is almost twice as that of coal-fired power plants operating at higher temperatures. In view of the above, there is an immediate need to improve the heat removal efficiency with a relatively low pressure drop. Reducing the pressure drop is critical if a natural draft cooling tower is contemplated instead of a fan-cooled condenser. Fans can consume a portion of the generated electricity, 0.6% for power plant according to Kroger (2004), on top of maintenance costs. In a natural draft cooling tower, a simple scale analysis shows that, to the first approximation, the pressure difference due to buoyancy (the driving force) is linearly proportional to the height of the cooling tower. This buoyancy-induced pressure difference needs to be large enough to compensate for pressure drop through the tower and the bundles. There is little one can do about the shape of the tower to reduce the pressure drop mainly because of structural issues but it is of particular interest to reduce the pressure drop through the bundles. At the same time, taller towers are more expensive to build. Therefore, it is of paramount importance to minimize the pressure drop through the bundles. Given the fact that fins can increase the pressure drop significantly, the question is to see if a more efficient heat transfer augmentation technique can be found with less pressure drop.

An alternative to finned tubes is a class of porous materials called metal foams. They offer low densities and novel thermal, mechanical, electrical and acoustic properties mainly because the foams are lightweight with high strength and rigidity and high surface area. These help the energy absorption and heat transfer in heat exchangers where the rate of heat transfer is extremely enhanced by conducting the heat to the material struts, which have a large accessible surface area per unit volume, along with high interaction with the fluid flowing through them. As the flow paths through the foams are interconnected the flow will be available in all areas leading to smaller and lighter heat exchangers. Normal foam ligaments in the flow direction result in boundary layer separation and mixing. The flow becomes turbulent and unsteady at pore Reynolds numbers greater than 100. Consequently, the induced turbulence and dispersion cause further heat transfer augmentation and, hence, improve the performance and efficiency of the heat exchanger. As a result, they have already been implemented in such industrial applications as heat exchangers in cryogenics, combustion chambers, cladding on buildings, strain isolation, buffer between a stiff structure and a fluctuating temperature field, petroleum reservoirs, compact heat exchangers for airborne equipments, high power batteries, compact heat sinks for power electronics and electronic cooling, heat pipes and sound absorbers.

Our goal in this paper is to examine the potential for metal foam heat exchangers in designing natural-draft dry cooling towers, an application where a favorable trade-off between the pressure drop and the heat transfer rate is of critical importance. In this application we propose the use of metal foams to replace fins at the outer layer of tubes in the air-cooled condenser of a geothermal power plant. In order to do so, we analyzed metal foams as porous media with typically high porosity consisting of tortuous, irregular shaped flow passages with different characteristics from packed beds and granular porous media. For instance, experiments with metal foams have indicated that, at the same Reynolds number, the pressure drop resulting from foam matrices is much lower than that by granular matrices, Liu, Wu et al. (2006). As the internal structure of metal foams is very complex, in addition to the random orientation of the solid phase, pore scale simulation is almost impossible. Metal foams include small continuously connected filaments in an open-celled foam structure. The cells are usually polyhedrons of 12–14 faces in which each face has a pentagonal or hexagonal shape (by five or six filaments). Consequently, the best can be done is to propose accurate volume-averaged models. This work will use such volume-averaged correlations based on previous theoretical and experimental studies.

Compared to plain tubes, metal foam filled tubes have significantly higher (up to 40 times) heat transfer performance. A metal foam tube heat exchanger is modeled analytically and pertinent correlations for friction factor, pressure drop and heat transfer coefficient (overall Nusselt number) are presented in Lu, Zhao et al. (2006); Zhao, Lu et al. (2006). The Brinkman and local thermal non-equilibrium models where used to analyze the fluid and heat transfer. Three different heat exchangers can be modeled with these correlations: the case where the inner tube of the heat exchanger is filled by the metal foam, Lu, Zhao et al. (2006); the case where the inner tube is surrounded by a metal foam ,Zhao, Lu et al. (2006); and the case where the inner tube is filled and surrounded by metal foams. The heat flux through the tube inner wall is assumed to be constant and uniform. Although the pressure drop correlation is arranged based on the heat exchanger tube diameter, the results indicate that the pressure drop is mainly caused by the foam solid structure rather than the pipe wall. Klett, Stinton et al. (2001) showed that solid foam radiators can transfer heat an order of magnitude better than the fin radiators. Boomsma and Poulikakos et al. (2003) indicated that the thermal resistances generated by the compressed open-cell aluminum foam heat exchangers were two to three times lower than the commercially available heat exchangers, while requiring about the same pumping power. Kim, Paek et al. (2000); Kim, Kang et al. (2001) experimentally examined heat transfer through aluminum foams inserted between two isothermal plates. Their results show that the foam material have better heat transfer performance compared to the conventional array fins, but subject to a greater pressure drop. This work focuses on the use of metal foams as external fins and reports approximate results for a simple geometry. Previously obtained results from commercially available software CFD-ACE and ASPEN B-JAC, for finned tube heat exchangers are presented here for comparison purpose.

2. ANALYSIS

Results from a previous study by Hooman and Gurgenci (2008) for a geothermal power plant that produces 50 MW electricity, with Isopentane as the working fluid in a binary cycle, is used as a basis for this work. There, based on results from ASPEN B-JAC, the well-established industrial software used by air-cooled heat exchanger designers, it was shown that the 283MW of the plant waste heat , Ejlali, Ejlali et al. (2008); Hooman and Gurgenci (2008), can be dumped, at the design-point ambient air temperature of 35^{o}C, by installing 288 tube bundles with 132 tubes in each bundle of 6mx0.2mx2.5m dimensions. The tubes are 6m long and have an outside diameter of 30mm. They are set in 33 columns and 4 rows. The physical model, as depicted in Fig. 1a, justifies the applied boundary conditions outlined in Fig. 1b. Further details of the system are not reported for the sake of brevity.

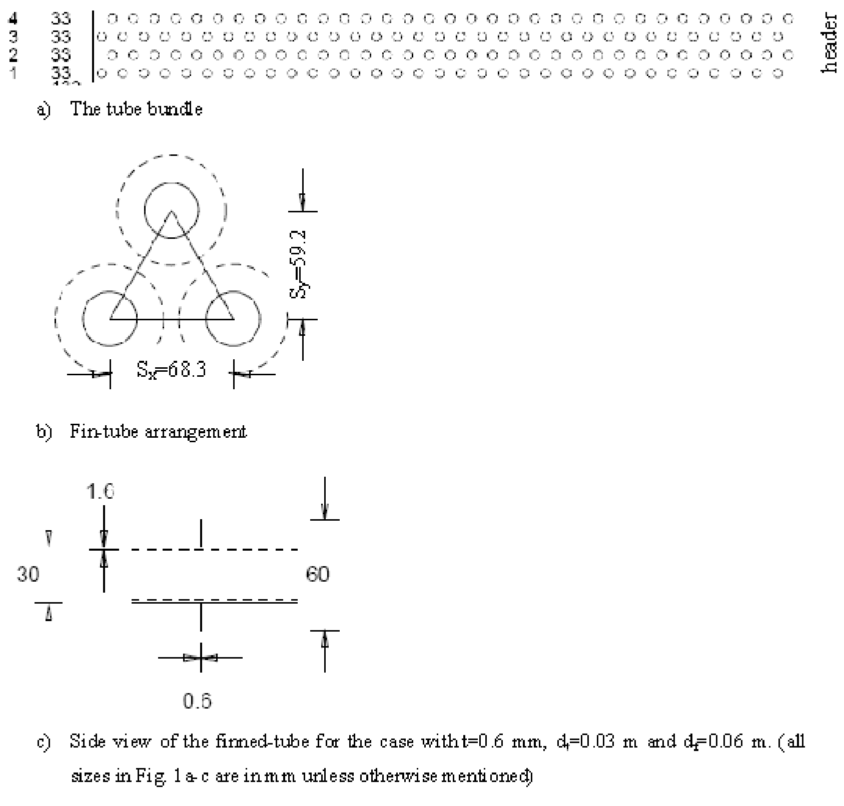

Figure 1: Schematic view of the finned-tube bundle considered here.

Table 1 Summary of the governing equations with $\nu_T=0.09k^2/\varepsilon$.

Equations	ϕ	Γ_ϕ	S_ϕ
Continuity	1	0	0
x-momentum	u/φ^2	$(\nu+\nu_T)/\varphi$	$-\frac{1}{\rho}\frac{\partial p}{\partial x}-\frac{(\nu+\nu_T)u}{K}-\frac{C_F u\sqrt{u^2+v^2}}{K^{1/2}}+\frac{\partial}{\partial x}((\nu+\nu_T)\frac{\partial u}{\partial x})+\frac{\partial}{\partial y}((\nu+\nu_T)\frac{\partial v}{\partial x})$
y-momentum	v/φ^2	$(\nu+\nu_T)/\varphi$	$-\frac{1}{\rho}\frac{\partial p}{\partial y}-\frac{(\nu+\nu_T)v}{K}-\frac{C_F v\sqrt{u^2+v^2}}{K^{1/2}}+\frac{\partial}{\partial x}((\nu+\nu_T)\frac{\partial u}{\partial y})+\frac{\partial}{\partial y}((\nu+\nu_T)\frac{\partial v}{\partial y})$
Energy	T	$\alpha+\nu_T/Pr_T$	$q_b/(\rho c_p)$
Turbulent energy*	K	$\nu+\nu_T$	$\nu_T\left(2(\frac{\partial u}{\partial x})^2+2(\frac{\partial v}{\partial y})^2+(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x})^2\right)-\varepsilon$
Turbulent dissipation*	E	$\nu+0.77\nu_T$	$\frac{\varepsilon}{k}\left(1.44\nu_T\left(2(\frac{\partial u}{\partial x})^2+2(\frac{\partial v}{\partial y})^2+(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x})^2\right)-1.92$

* For clear fluid only; in porous layer we have $k=1.5\varphi^2(u^2+v^2)^{1/2}/10^4$ and $\varepsilon=1.643k^{3/2}$.

In a subsequent study, we have used the rating mode of the software with V=5.45 m/s as the face velocity and T_{in}=35°C as the environment temperature and analyzed the heat exchanger performance for fixed tube dimensions (length and diameter), size, and number of the tube bundles, tube arrangement, inlet temperature, and the air mass flow rate but at different fin number densities and fin sizes so that the porosity and permeability alter. Results obtained from this software were compared against numerical simulations obtained from the commercially available CFD package CFD-ACE, which implements porous media transport equations given by the generic form of equation 1, Hooman and Gurgenci (2009):

$$\frac{\partial(u\phi)}{\partial x}+\frac{\partial(\mathrm{v}\phi)}{\partial y}=\frac{\partial}{\partial x}(\Gamma_\phi\frac{\partial\phi}{\partial x})+\frac{\partial}{\partial y}(\Gamma_\phi\frac{\partial\phi}{\partial y})+S_\phi \quad (1)$$

with φ=1 and $K\to\infty$ for non-porous regions while for the tube bundle the porosity can be determined as the volume occupied by the fluid divided by the total (bundle) volume

$$\varphi=1-\frac{V_f+V_t}{V_b} \quad (2)$$

where V_f, V_t, and V_b are volumes occupied by the fins, bare tubes, and the (total) bundle, respectively. These are obtainable as

$$V_t=N_t L\pi\frac{d_t^2}{4}$$
$$V_f=N_t N_f Lt\pi\frac{d_f^2-d_t^2}{4} \quad (3a\text{-}c)$$
$$V_b=L\left((N_x-1)S_x+2d_f\right)\left((N_y-1)S_y+d_f\right)$$

where t is the fin thickness, S and N are the pitch and number (of the tubes) with the subscripts x,y showing the directions and t,f,b representing the tubes, fins, and the bundle, respectively.

Thus, the porosity is given by

$$\varphi=1-\frac{\pi d_t^2 N_t}{4}\frac{1+N_f t\left(1-(d_f/d_t)^2\right)}{\left((N_x-1)S_x+2d_f\right)\left((N_y-1)S_y+d_f\right)} \quad (4)$$

In this problem, we have

$$\varphi=1-0.24\frac{1+0.00058N_f\left(1-(d_f/0.03)^2\right)}{\left(1+0.915d_f\right)\left(1+5.63d_f\right)} \quad (5)$$

Following Bejan and Morega (1993), the permeability for the tube bundle is defined as

$$K=\frac{d_t^2\varphi^3}{100(1-\varphi)^2} \quad (6)$$

A sample of our results is presented in figure 2 where excess pressure drop and heat transfer, respectively, compared to no fin case, defined as

$$e_p=\frac{\Delta p-\Delta p_{\text{no fin}}}{\Delta p_{\text{no fin}}} \quad (7\text{-a})$$

$$e_q=\frac{Q-Q_{\text{no fin}}}{Q_{\text{no fin}}} \quad (7\text{-b})$$

are depicted where they are best fitted by

$$e_p=22.19(0.78-\varphi) \quad (8\text{-a})$$

$$e_q=2.41-\frac{0.03}{0.796-\varphi} \quad (8\text{-b})$$

The highest porosity, 0.78, is associated with the no fin case selected as our reference state in equations 7-8. As seen, increasing the fin number or density the porosity decreases.

This, in turn, increases the pressure drop as the resistance to fluid flow is boosted. At the same time, the heat transfer augmentation is more pronounced mainly because of surface extension. Note that while the cost, excess pressure drop, increases monotonically, according to equation 8-a linearly, the heat transfer shows an asymptotic increase signaling that from a certain point on there is no point in adding to the solidity as the gain is less than the cost. After that point, any marginal improvement in the heat transfer rate may not be worth the cost paid in extra pressure drop for porosities below 0.72. For example, the improvement in heat transfer is only 10% against a 200% increase in pressure drop for a porosity reduction from 0.72 to 0.64. This non-uniform behavior of heat transfer augmentation can be attributed, at least in part, to the change in the fluid flow behavior due to the presence of more fins. That is, a less porous bundle tends to damp the turbulence and also to limit the access of flow to the high temperature region (heat source, i.e. the tubes).

Let us mention that this threshold value of porosity is regarded to as the optimum design point by the software (ASPEN B-JAC). This optimum point can be referred to as the point where the maximum heat transfer augmentation is obtained at the least possible pressure drop.

It has also been shown that for φ=0.64-0.78 the form drag coefficient is best fitted by

$$C_F = 0.55\left[9.887(1-\varphi)(\varphi-0.323) \text{ -}0.8443\right] \tag{9}$$

The total pressure drop is then given by

$$\frac{\Delta P}{L} = \frac{\mu V}{K} + \frac{C_F \rho V^2}{\sqrt{K}} \tag{10}$$

where μ and ρ are the fluid (here air) viscosity and density, respectively. For this specific case the viscous drag (Darcy) term is negligible compared to the form drag term so the approximate pressure drop is given by

$$\frac{\Delta P}{L} \cong \frac{C_F \rho V^2}{\sqrt{K}} \tag{11}$$

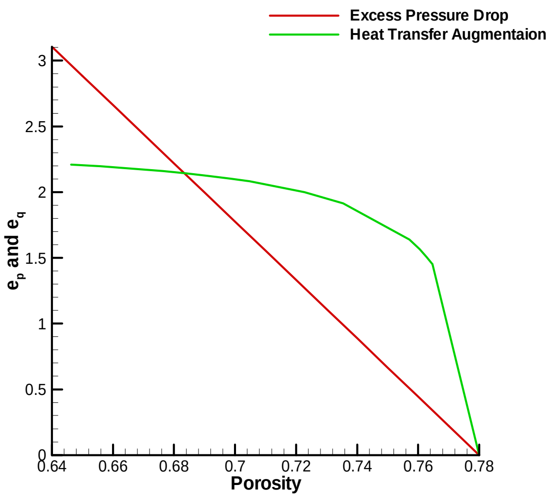

Figure 2: Excess pressure drop and heat transfer versus porosity.

Table 2: Comparison between the performance of metal foams, finned-tubes, and plain channels.

Parameter	Plain channel	Metal foam	Finned tube	Metal foam/Fin
f	0.045	2.96	0.92	3.22
h(W/m^{2}K)	13.36	683.6	344.45	1.99

If rearranged in terms of a friction coefficient, the above equation reads

$$f = \frac{2\Delta P}{\rho V^2} \cong \frac{2C_F L}{\sqrt{K}} \tag{12}$$

2.1 A case study

In what follows the heat transfer and pressure drop of a metal foam is compared with those of fins in the tube bundle shown in Figure 1. According to Hooman and Gurgenci (2009) when φ=0.72 about 6% of the total volume is occupied by the fins. With Aluminum foam of 93% porosity, nearly the same mass is required. According to Lefebvre, Banhart et al. (2007), Alcoa (USA) estimates the price at 5 US$/kg for large-scale production based on the continuous casting of the foam using $CaCO_3$ as blowing agent; a technique developed in 2006.

Assuming that the fins and the foam cost almost the same, we can proceed with examining the heat transfer and pressure drop performance of the two. A sample of metal foam with $\varphi=0.93$, $K=4.2x10^{-8}m^2$, and $C_F=0.0076$ Jin and Leong (2008) is hypothetically inserted on the outer layer of a tube to fill the space in between the tubes. This allows us to use a porous-saturated conduit assumption. The performance of this channel, in view of heat removal and pressure drop, is compared with a finned tube from the designed heat exchanger and the results are presented in Table 2. Pertinent correlations are taken from Mahjoob and Vafai (2008).

As seen, both finned tube and metal foam show superior heat transfer performance compared to plain channels albeit at the expense of a higher pressure drop. The metal foam examined here leads to 99% improvement in heat transfer at the expense of about 222% higher pressure drop compared to a finned-tube heat exchanger. It should be noted that this is just a sample of our data and by no means represents the optimal design of the aluminum foam whereas the finned tube design corresponds to the optimum design point found by the software. It is known, based on previous studies, that applying lower velocities to a metal foam heat exchanger can lead to lower pressure drop even those comparable to that of finned tubes. This is in line with our earlier observation where a metal foam heat exchanger cooled by a laminar jet was compared to a plate fin counterpart where at the same pressure drop higher heat transfer rate was observed as a result of the application of the metal foam, see Ejlali, Ejlali et al. (2009).

According to Hooman and Gurgenci (2009), it can be shown that the pressure drop by fins only, can be given by

$$\Delta P_{fins} \cong \frac{e_p}{1+e_p} \frac{C_F \rho V^2}{\sqrt{K}} \tag{13}$$

where that of a metal foam can be predicted as

$$\Delta P_{mf} = \left(\frac{C_F \rho V^2}{\sqrt{K}}\right)_{mf} \tag{14}$$

The ratio then is given by

$$\frac{\Delta P_{fins}}{\Delta P_{mf}} \cong \frac{e_p}{1+e_p} \frac{C_F V^2}{\sqrt{K}} \frac{1}{\left(\frac{C_F V^2}{\sqrt{K}}\right)_{mf}} \tag{15}$$

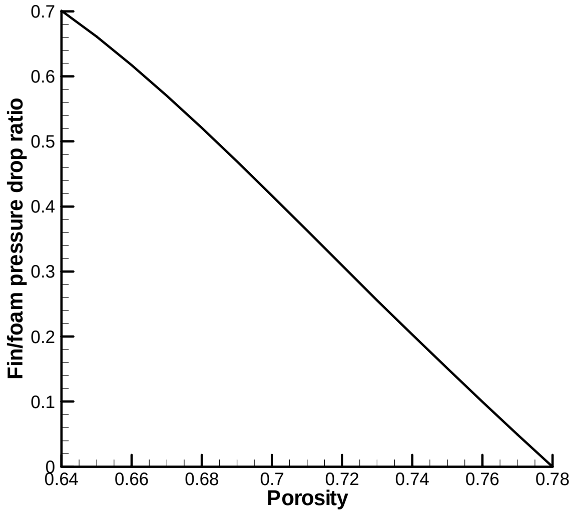

Figure 3: Fin to metal foam excess pressure drop ratio versus porosity

Assuming the same velocity and density for both systems, we varied the porosity of the finned-tube bundles and fixed that of metal foams to those from Jin and Leong (2008), with for $\varphi=0.93$, $K=4.2x10^{-8}m^2$, and $C_F=0.0076$, and made use of our equations 8-a and 8-b, i.e.

$$\frac{\Delta P_{fins}}{\Delta P_{mf}} \cong \frac{10^{-6}(1-\varphi)^{1.5}}{2\left(\varphi+0.045\varphi(0.78-\varphi)^{-1}\right)} \left(9.887(1-\varphi)(\varphi-0.332)-0.8443\right) \quad (16)$$

The result from the above equation is plotted versus the porosity as depicted by figure 3. As seen, increasing the porosity decreases the pressure drop ratio because the highest porosity is associated with the no fin or bare tube case. Similar graph can be generated once the metal foam is selected and when it comes to analyze the pressure drop performance.

3. CONCLUSION

Application of metal foams to the outer layer of tubes in an air cooled condenser is examined. The results for a case study were compared to conventional finned-tube bundle design. Both of the above heat transfer augmentation techniques lead to, as expected, higher heat transfer rates compared to plain channel heat exchangers of course at higher pressure drops. The optimal design of a finned-tube bundle showed less pressure drop and lower heat transfer compared to a random sample of a metal foam heat exchanger. The results of this study indicate a potential for improving the performance of natural-draft dry cooling towers by using metal foam heat exchangers in place of conventional finned tube bundles. Since heat exchange components may amount to over 60% of the capital cost for a natural draft cooling tower, improving the heat exchanger performance would have a substantial influence in economic feasibility of natural draft towers for geothermal power applications.

REFERENCES

Bejan, A. and A. M. Morega (1993). "Thermal Contact Resistance Between 2 Flat Surfaces That Squeeze A Film of Lubricant." Journal of Heat Transfer-Transactions of the Asme 115(3): 763-767.

Boomsma, K., D. Poulikakos, et al. (2003). "Simulations of flow through open cell metal foams using an idealized periodic cell structure." International Journal of Heat and Fluid Flow 24(6): 825-834.

Ejlali, A., A. Ejlali, et al. (2008). A comparative study on dry cooling of different working fluids for geothermal applications. Australian Geothermal Energy Conference H. Gurgenci and A. Budd. Melbourne, Australia.

Ejlali, A., A. Ejlali, Hooman, K., Gurgenci, H. (2009). "Application of high porosity metal foams as air cooled heat exchangers to high heat load removal systems." International Communications in Heat and Mass Transfer 36(7): 674-679.

Hooman, K. and H. Gurgenci (2008). Air cooled porous matrix heat exchangers for geothermal applications. Australian Geothermal Energy Conference H. Gurgenci and A. Budd. Melbourne, Australia.

Hooman, K. and H. Gurgenci (2009). "Porous Medium Modeling of Air-Cooled Condensers " Transport in Porous Media submitted.

Jin, L. W. and K. C. Leong (2008). "Pressure drop and friction factor of steady and oscillating flows in open-cell porous media." Transport in Porous Media 72(1): 37-52.

Kim, S. Y., B. H. Kang, et al. (2001). "Forced convection from aluminum foam materials in an asymmetrically heated channel." International Journal of Heat and Mass Transfer 44(7): 1451-1454.

Kim, S. Y., J. W. Paek, et al. (2000). "Flow and heat transfer correlations for porous fin in a plate-fin heat exchanger." Journal of Heat Transfer-Transactions of the Asme 122(3): 572-578.

Klett, J., D. Stinton, et al. (2001). Heat exchangers/radiators utilizing graphite foams, Oak Ridge National Laboratory, US Department of Energy.

Kroger, D. (2004). Air-Cooled Heat Exchangers and Cooling Towers, Penn Well.

Lefebvre, L. P., J. Banhart, et al. (2007). Porous Metals and Metallic Foams: Current Status and Recent Developments. 5th Biennial International Conference on Porous Metals and Metallic Foams, Montreal, CANADA.

Liu, J. F., W. T. Wu, et al. (2006). "Measurement and correlation of friction characteristic of flow through foam matrixes." Experimental Thermal and Fluid Science 30(4): 329-336.

Lu, W., C. Y. Zhao, et al. (2006). "Thermal analysis on metal-foam filled heat exchangers. Part I: Metal-foam filled pipes." International Journal of Heat and Mass Transfer 49(15-16): 2751-2761.

Mahjoob, S. and K. Vafai (2008). "A synthesis of fluid and thermal transport models for metal foam heat exchangers." International Journal of Heat and Mass Transfer 51(15-16): 3701-3711.

Zhao, C. Y., W. Lu, et al. (2006). "Thermal analysis on metal-foam filled heat exchangers. Part II: Tube heat exchangers." International Journal of Heat and Mass Transfer 49(15-16): 2762-2770