Brazilian Journal
of Chemical
Engineering

ISSN 0104-6632
Printed in Brazil
www.abeq.org.br/bjche

Vol. 27, No. 01, pp. 127 - 135, January - March, 2010

ESTIMATION OF EFFECTIVE THERMAL CONDUCTIVITY ENHANCEMENT USING FOAM IN HEAT EXCHANGERS BASED ON A NEW ANALYTICAL MODEL

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(Submitted: May 5, 2009 ; Revised: October 10, 2009 ; Accepted: October 27, 2009)

Abstract - Thermal performance of open-cell metal foam has been investigated under low Reynolds number by comparing the heat transfer coefficient and thermal conductivity for the flow through a packed channel of high porosity metal foam to that of an open channel. In the case of Al-Air at porosity 0.971, the ratio of heat transfer coefficients is estimated to be 18.5 when the thermal conductivity ratio of foam matrix to fluid conductivity is 130. This demonstrates that the use of foam in the structure of conventional air coolers increases effective thermal conductivity, heat transfer coefficient and thermal performance considerably. To overcome the drawbacks of previous models, a new model to describe the effective thermal conductivity of foam was developed. The model estimates effective thermal conductivity based on a non-isotropic tetrakaidecahedron unit-cell and is not confined only to isotropic cases as in previous models. Effective thermal conductivity is a function of foam geometrical characteristics, including ligament length (L), length of the sides of horizontal square faces (b), inclination angle that defines the orientation of the hexagonal faces with respect to the rise direction (θ), porosity, size, shape of metal lump at ligament intersections and heat transfer direction. Changing dimensionless foam ligament radius or height (d) from 0.1655 to 0.2126 for Reticulated vitreous foam –air (RVC-air) at $\theta=\pi/4$ and dimensionless spherical node diameter (e) equal to 0.339 raises effective thermal conductivity by 31%. Moreover, increasing θ from $\pi/4$ to 0.4π for RVC-air at d=0.1655 and e=0.339 enhances effective thermal conductivity by 33%.
Keywords: Metal foam; Thermal performance; Effective thermal conductivity; Heat transfer coefficient.

INTRODUCTION

In many industrial applications, a large amount of heat must be transported from a fluid to the surrounding air through conventional air coolers. Space limitations have led to the use of novel lightweight materials for improving the thermal management applications. High porosity open-cell metal foam ($\varepsilon \succ 0.85$) has been widely used as one of the most promising materials due to its diverse properties such as high surface area to volume ratio, tortuous flow path to promote mixing, and attractive stiffness-strength properties (Zhao et al., 2004; Bhattacharya and Mahajan, 2002; Boomsma et al., 2003; Lu, et al.,1998). These attractive properties of metal foams have directed researchers to the new class of compact heat exchangers commonly known as foam heat exchangers. In their structure, the high porosity metal foam is combined with tubes and sheets to enhance heat dissipation to the environment (Dukhan and Chen, 2007; Zhao et al., 2006). It has already been pointed out that the use of high porosity

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metal foams can significantly enhance the heat transfer performance of foam heat exchangers compared to conventional air coolers (Lu et al., 2006; Phanikumar and Mahajan, 2002; Ghosh, 2008).

The mechanisms that contribute to the enhanced heat transfer include heat conduction in the metal foam matrix (which is usually several orders of magnitude higher compared to gaseous fluid conductivity), convective heat transfer between the metal foam and cooling fluid and thermal dispersion in the fluid at high velocities cause by eddies shed behind the solid (Phanikumar and Mahajan, 2002; Ghosh, 2008; Calmidi and Mahajan, 2000; Salas and Waas, 2007; Dukhan et al., 2005). Amongst these, thermal dispersion and convection become more important in cases of high fluid velocities, but in the case of natural convection, conduction plays the most important role in enhancing the total rate of heat transfer. Metal foams have also been used successfully as thermal conductivity enhancers for low-Reynolds-number flows. The widespread range of applications of metal foams has led to an increase in the interest of modeling heat transfer phenomena in porous media. The precise calculation of effective thermal conductivity is required for accurate modeling of thermal transport through open-cell metal foam as well as foam heat exchangers. For this reason, many researches are dedicated to analyzing the conduction term in the effective thermal conductivity. By developing a comprehensive and precise model for calculating the effective thermal conductivity contribution and importance of conduction in the total rate of heat transfer can be predicted.

Complex geometry and the large difference in thermal conductivity of fluid and solid in the structure of open-cell metal foam present some difficulties in conductivity calculations. The objective of this paper is to compare the effective thermal conductivity (k_{eff}) of foam and conductivity of fluid and to determine the parameters that are effective in enhancing the foam heat exchangers performance by increasing the effective thermal conductivity of foam, especially at low Reynolds numbers. This objective is obtained by developing a new three dimensional effective thermal conductivity model due to the shortcomings of previously developed models, which will be explained in the following section.

Heat exchanger performance improvement assists industries to save on space occupied by the heat exchangers, reduces installation and running costs due to the lighter equipment and, most importantly, saves on energy consumption.

The simplified one-dimensional model for estimating the k_{eff} of foam is expressed as $k_{eff}=\varepsilon k_f+(1-\varepsilon)k_s$ (Boomsma and Poulikakos, 2001; Calmidi and Mahajan, 1999). But this equation is not precise in the case of the vastly different thermal conductivities encountered in the metal foam combinations.

Calmidi and Mahajan (1999) proposed a one-dimensional conductivity model considering the porous medium to be formed of a two-dimensional array of hexagonal cells and using square nodes at the intersection of ligaments. Using a one-dimensional REV for estimating the effective thermal conductivity is not precise. This analytical model was modified to replace the metal lump at the intersection of ligaments with a square shape by a circular one by Bhattacharya et al. (2002). The accurate three-dimensional geometrical description of metal foam is of importance, which was considered in subsequent investigations. Boomsma and Poulikakos (2001) extended the idea of one-dimensional conductivity in a two-dimensional foam structure to the more complex three-dimensional structure. In their proposed model, one-sixteenth of a single tetrakaidecahedron was selected as the representative unit-cell (REV) and was discretised to distinct layers in series. Their research was based on a few geometrical assumptions such as the circular cross section of fibers, a cubic node at the intersection of fiber ligaments and isotropic geometry. But the validity of these is doubtful in many cases, because the cross-section of ligaments is a function of porosity and changes from a circle to an inner concave triangle with increasing porosity. Spherical nodes result in better agreement with experimental data. Also, in many cases the foam structure is elongated in the rise direction causing the foam behavior to be non-isotropic, a phenomenon addressed, for the first time, by the new model presented here, as the previous models have been confined to isotropic cases. In order to overcome these shortcomings and increase the accuracy, a new general model for estimating k_{eff} for non-isotropic open-cell metal foam was developed in this research based on a general description. The equations for effective thermal conductivity are developed in two different directions, z and y. Finally, by using this general structure definition, the relationship between the non-isotropic geometrical characteristics of open-cell metal foam and the thermal conductivity value was studied accurately.

MODEL DEVELOPMENT

Effective Thermal Conductivity

Accurate modeling of metal foam thermal conductivity requires precise definition of the foam structure. The foam matrix is made up of unit-cells in their connected state. The tetrakaidecahedron was identified as the only polyhedron that packs to fill space and minimizes the surface area per unit volume (Boomsma and Poulikakos, 2001; Bhattacharya et al., 2002; Sullivan et al., 2008; Kwon et al., 2003). It is assumed that, the foam structure is organized by repeating these unit-cells. The unit-cell consists of six quadrilateral and eight hexagonal faces with spherical nodes at the intersection of fiber ligaments. The horizontal squares have sides of length b, while the other ligaments have sides of length L. θ denotes the orientation of the hexagonal faces with respect to the rise direction, as shown in Fig. 1. The cross-section of the metal fiber changes from a circle at $\varepsilon=0.85$ to a hypocycloid at higher porosity of about $\varepsilon=0.97$. The radius of the circular or height of the triangular cross-section of the fiber is defined by R and the diameter of the spherical node by r.

In most previously developed models, isotropic geometry is assumed. But, in most real cases, the foam structure has non-isotropic behavior. It can be predicted that the non-isotropic geometry affects the thermal conductivity value. Therefore, effective thermal conductivity should be modeled with regard to the heat transfer direction. In other words, the equations for effective thermal conductivity are developed in two different directions, z and y for open-cell metal foam using a general geometric description.

In order to model the conductivity of the foam matrix, an appropriate elementary volume should be selected as representative of foam media. This representative volume must reflect all geometrical characteristics of the foam matrix, which are different in the y and z directions. It can be easily shown that a tetrakaidecahedron unit-cell has a symmetrical structure in y and z coordinates even in non-isotropic cases. Meanwhile, the unit-cells are connected to each other with quadrilateral faces and a void fraction exists between the connected cells. Then a cube which encapsulates one half of a single tetrakaidecahedron and part of the void fraction between two neighboring cells was selected as an REV for k_{eff} calculation, as shown in Fig. 2(a) and 2(b). Repeating these selected REV's on both sides forms the foam media.

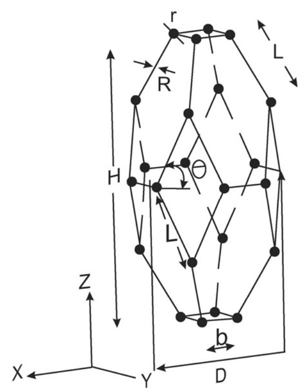

Figure 1: A general tetrakaidecahedron as repeating unit-cell. θ

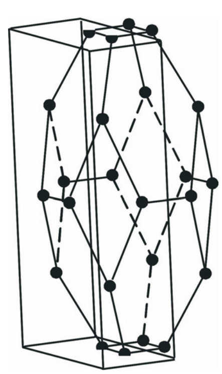

Figure 2(a): Repeating REV in y direction.

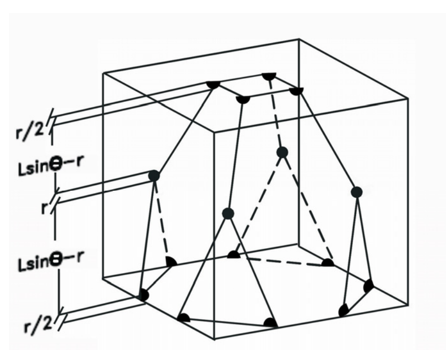

Figure 2(b): Repeating REV in z direction

At first a model for conductivity is developed in the y direction based on an appropriate REV, as shown in Figs. 2(a). The REV is discretised to five parallel layers in the y direction with the lengths $L\cos\theta-\frac{r}{2}$, r, $L\cos\theta-r$, r, and $(b-r)/2$, respectively, and the width and height of each tetrakaidecahedon are defined by $D=2L\cos\theta+\sqrt{2}b$ and $H=4L\sin\theta$. The ligaments of all quadrilateral faces are laid between two adjacent cells. Each node at the fiber intersections is common between two cells. k_{eff} is the summation of the effective thermal conductivity of each layer in series. The effective thermal conductivity of each layer is the weighted arithmetic mean of solid and fluid conductivities as follows:

$$k_{eff} = \varepsilon k_f + (1-\varepsilon) k_s \tag{1}$$

In order to simplify, k_{eff} is formulated based on the following dimensionless relationships:

$$d = R/L$$

$$e = r/L$$

Hence the following general equations were obtained for thermal conductivity of each section in the y direction for high porosity domain:

$$k_1 = \frac{\left(A(1-e)+2A\left(1-\frac{e}{\cos\theta}\right)+\frac{\pi e^3}{6}\right)k_s + \left(\left(\cos\theta-\frac{e}{2}\right)(4\sin\theta)^2 - A(1-e) - 2A\left(1-\frac{e}{\cos\theta}\right)-\frac{\pi e^3}{6}\right)k_f}{\left(\cos\theta-\frac{e}{2}\right)(4\sin\theta)^2} \tag{2}$$

$$k_2 = \frac{\pi e^3/3k_s + \left(e(4\sin\theta)^2 - \pi e^3/3\right)k_f}{e(4\sin\theta)^2} \tag{3}$$

$$k_3 = \frac{8A\left(1-\frac{e}{\cos\theta}\right)k_s + \left((\cos\theta-e)(4\sin\theta)^2 - 8A\left(1-\frac{e}{\cos\theta}\right)\right)k_f}{(\cos\theta-e)(4\sin\theta)^2} \tag{4}$$

$$k_4 = \frac{\left(A\left(\frac{b}{L}-e\right)+\frac{\pi e^3}{2}\right)k_s + \left(e(4\sin\theta)^2 - A\left(\frac{b}{L}-e\right)-\frac{\pi e^3}{2}\right)k_f}{e(4\sin\theta)^2} \tag{5}$$

$$k_5 = \frac{3/2\left(\frac{b}{L}-e\right)k_s + \left(\left(\frac{b}{2L}-\frac{e}{2}\right)(4\sin\theta)^2 - 3/2\left(\frac{b}{L}-e\right)\right)k_f}{\left(\frac{b}{2L}-\frac{e}{2}\right)(4\sin\theta)^2} \tag{6}$$

$$k_{eff} = \frac{\left(2\cos\theta+\frac{b}{2L}\right)}{\frac{(\cos\theta-e/2)}{k_1}+\frac{e}{k_2}+\frac{(\cos\theta-e)}{k_3}+\frac{e}{k_4}+\frac{(b-e)/2}{k_5}} \tag{7}$$

Then the conductivity model is developed in the z direction. The assumed REV is discretised to five distinct layers in series in the z direction with lengths $r/2$, $L\sin\theta - r$, r, $L\sin\theta - r$, and $r/2$, respectively. As in the y direction, each solid node is common between two cells. But only one-fourth of nodes contribute at the lower cross-section of first layer, as shown in Fig. 2(b). Details of the effective thermal conductivity model in the z direction were presented by Haghighi and Kasiri (2009). This model is a function of geometrical parameters of the foam matrix. The overall formula for conductivity in the z direction is as follows:

$$k_{eff} = \frac{2L\sin\theta}{\dfrac{\frac{r}{2}}{k_1} + \dfrac{(L\sin\theta - r)}{k_2} + \dfrac{r}{k_3} + \dfrac{(L\sin\theta - r)}{k_4} + \dfrac{\frac{r}{2}}{k_5}} \quad (8)$$

The non-dimensional geometrical parameters d and e are substituted into the above analytical equations and θ is assumed to be 45°. k_{eff} in the z direction is then given by the following simplified formula:

$$k_{eff} = \frac{1}{\dfrac{\frac{1}{2\sqrt{2}}e}{k_1} + \dfrac{\left(0.5 - \frac{1}{\sqrt{2}}e\right)}{k_2} + \dfrac{\frac{1}{\sqrt{2}}e}{k_3} + \dfrac{\left(0.5 - \frac{1}{\sqrt{2}}e\right)}{k_4} + \dfrac{\frac{1}{2\sqrt{2}}e}{k_5}} \quad (9)$$

A is the cross-sectional area of fibers which is replaced by πR^2 in the case of a circular cross-section and $\sqrt{3}/3R^2$ for a triangular cross-section. This model is the most general form of the effective thermal conductivity, which permits one to predict results precisely for any structure of the foam matrix. This extended equation reveals that conductivity is a function of the geometric characteristics of the foam such as L, b, r, R, cross-sectional shape of fiber ligaments, node shape at the intersection of ligaments and inclination angle θ that defines the orientation of the hexagonal faces with respect to the rise direction.

These geometric parameters can be expressed in term of porosity. In the next section, porosity is represented as a function of geometrical specifications of the foam media.

Porosity

Porosity is defined as the ratio of volume of a selected REV that is occupied by saturating fluid to the total volume of the REV. A cube that encompasses one unit-cell is used as REV for calculating the porosity. The total volume of this REV is $V_{tot} = HD^2$. Sixteen ligaments with length L and 8 ligaments with length b contribute to the quadrilateral faces and 24 nodes exist in one unit cell, which are common between neighbor cells. Therefore, only one-half of their volume is included in the calculation.

$$V_s = 16AL + 8Ab + 2\pi r^3 \quad (10)$$

And porosity can be given in general form (Haghighi and Kasiri, 2009):

$$\varepsilon = \frac{16AL + 8Ab + 2\pi r^3}{4L\sin\theta\left(2L\cos\theta + \sqrt{2}b\right)^2} \quad (11)$$

RESULTS AND DISCUSSION

The Effect of Geometric Characteristics on Effective Thermal Conductivity

To validate the developed model, the results are compared to another three-dimensional model proposed by Boomsma and Poulikakos (2001) and experimental data reported by Bhattacharya et al. (2002). The width of the unit-cell D is assumed to be equal to the pore diameter d_p that was obtained by Bhattacharya et al. (2002). Based on this assumption, L is estimated by:

$$L = \frac{\left(D - \sqrt{2}b\right)}{2\cos\theta} = \frac{\left(d_p - \sqrt{2}b\right)}{2\cos\theta} \quad (12)$$

e is assumed to be equal to 0.339 as proposed by Boomsma and Poulikakos (2001). The allowable range for inclination angle θ and the ratio b/L were determined by Sullivan et al. (2008).

$$\mathrm{Arc}\sin\left\{\frac{\sqrt{2}b}{5L} + \frac{\sqrt{2}}{10}\sqrt{10 - \frac{b^2}{L^2}}\right\} \prec \theta \prec \frac{\pi}{2} \quad (13)$$

$$0 \prec b/L \prec 2\sqrt{2} \quad (14)$$

In non-isotropic geometry, conductivity in the y and z directions is different. As experimental data is obtained in the z direction, then the developed model in the z direction is compared to the other models which are demonstrated in Table 1. In this table, k_{eff}

values at different parameter combinations are reported using the new analytical model, that of Boomsma and Poulikakos (2001) and also the experimental data of Bhatacharya et al. (2002). k_{eff} values are estimated for high porosity metal foam $(\varepsilon \succ 0.95)$ at defined geometric values $b/L=1$, $\theta=\pi/4$, isotropic unit-cell and e=0.339, the same as the experimental data. The results show that the new analytical model follows the experimental data better than the previous model of Boomsma and Poulikakos (2001).

The results show that the proposed model predicts effective thermal conductivity better than the model proposed by Boomsma and Poulikakos (2001). On the other hand, the norm of deviation of results of the proposed model and the experimental data is lower than the norm of deviation of the results of the Boomsma-Poulikakos model and the experimental data.

This is due to the accuracy in the definition of the REV. They assumed a cube that encapsulates one sixteenth of a single tetrakaidecahedron as an appropriate REV. But in the present model, one half of a tetrakaidecahedron and the void fraction between two neighbor cells is proposed as the REV, which reflects the geometrical characteristics of the foam matrix more precisely than the REV proposed by Boomsma and Poulikakos (2001). The proposed model considered the effect of changing the inclination angle θ and fiber cross-section and heat transfer direction on effective thermal conductivity. The model of Boomsma and Poulikakos (2001) cannot predict these effects on effective thermal conductivity. The incentive behind this investigation is the study of the characteristics of the novel high porosity metal foam material applied in the structure of foam heat exchangers in order to increase the efficiency of conventional heat exchangers. Although the metal most usual in foam heat exchangers is aluminum, the results of the proposed model are valid for other high porosity materials. To assess this, the results are plotted for RVC-air and RVC-water as well.

As shown in Figs. 3(a) and 3(b), the effective thermal conductivity increases with decreasing porosity and increasing

This is due to the fact that, by increasing the porosity and decreasing θ, the effective fluid thermal conductivity (εk_f) in each layer increases, the effective solid thermal conductivity $(1-\varepsilon)k_s$ decreases and, consequently, the overall effective thermal conductivity is reduced.

Another geometric parameter that affects k_{eff} considerably is d. Increasing d raises the effective thermal conductivity, which is apparent in Fig. 4. The effect of these parameters (θ and d) on the effective thermal conductivity reveals that θ and d play an important role in raising k_{eff} and so the conduction rate. This is of significance in foam design problems, allowing the designer to come up with foams with larger effective thermal conductivity. Consequently, this can lead to a higher conduction rate and higher overall heat transfer rate of foam heat exchangers. Table 2 shows the influence of both aforementioned geometric parameters on conductivity.

From Fig. 5, it is seen that the effective thermal conductivity is several orders of magnitude higher than fluid thermal conductivity and is decreased slowly by increasing porosity.

Table 1: Comparison between k_{eff} for the proposed model, the Boomsma and Poulikakos model (2001) and the experimental data measured by Bhattacharya et al.(2002).

Sample combination	Porosity	k_{eff} (W/mK^{-1}) Experimental data Bhattacharya et al. (2002)	k_{eff} (W/mK^{-1}) (New Model)		k_{eff} (W/mK^{-1}) Boomsma and Poulikakos Model (2001)	
Al-air	0.971	2.7	3.4361	1.1096 (deviation from experimental data)	2.0288	1.2566 (deviation from experimental data)
Al-air	0.972	2.5	3.3032		1.9276	
Al-air	0.978	2.2	2.4102		1.3050	
Al-water	0.971	3.7	4.0967	0.7808 (deviation from experimental data)	2.9428	1.1831 (deviation from experimental data)
Al-water	0.972	3.3	3.9684		2.8469	
Al-water	0.978	3.05	3.1241		2.2620	
RVC-air	0.9615	0.17	0.2047	0.0436 (deviation from experimental data)	0.1532	0.0582 (deviation from experimental data)
RVC-air	0.9664	0.164	0.1840		0.1355	
RVC-air	0.9681	0.16	0.1764		0.1292	
RVC-air	0.9724	0.15	0.1555		0.1133	
RVC-water	0.9615	0.7430	0.8071	0.1081 (deviation from experimental data)	0.8687	0.2260 (deviation from experimental data)
RVC-water	0.9664	0.73	0.7869		0.8462	
RVC-water	0.9681	0.727	0.7797		0.8384	
RVC-water	0.9724	0.722	0.7615		0.8187	

Deviation from experimental data was estimated based on the Least Square Method

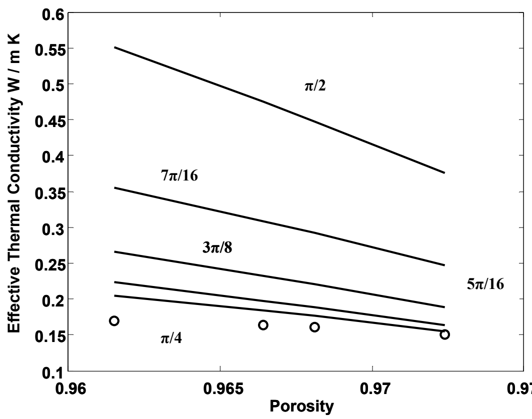

Figure 3(a): The effect of porosity on k_{eff} as predicted by the proposed model for RVC-air at b/L=1 and e=0.339 (with θ as parameter). Open symbols refer to experimental points.

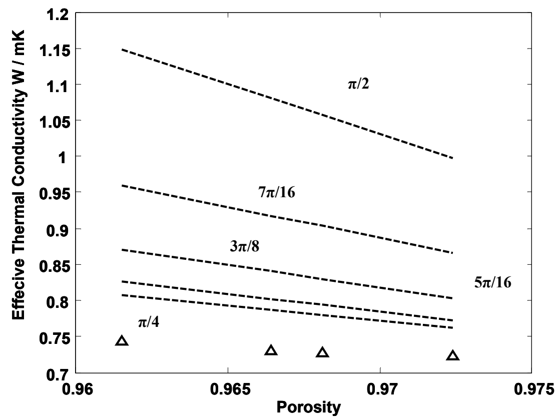

Figure 3(b): The effect of porosity on k_{eff} as predicted by the proposed model for RVC-water at b/L=1 and e=0.339 (with θ as parameter). Open symbols refer to experimental points.

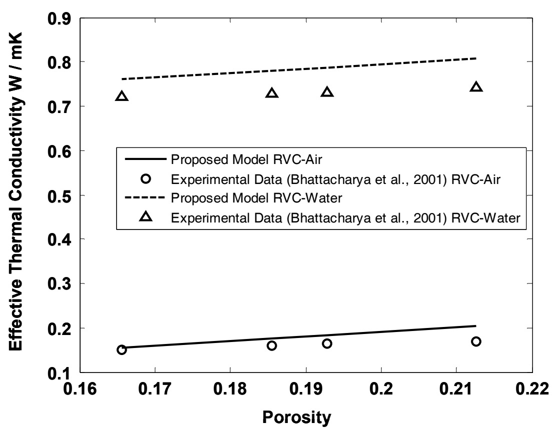

Figure 4: Effect of d on k_{eff} for RVC-Air and RVC-Water at b/L=1, $\theta = \pi/4$ and e=0.339

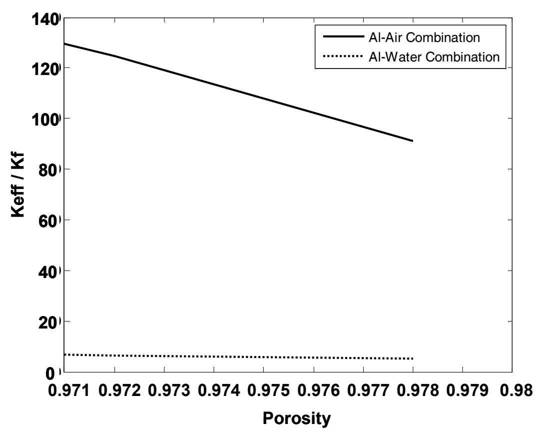

Figure 5: Effect of porosity in enhancing the effective thermal conductivity for Al-air and Al-water at b/L=1, $\theta = \pi/4$ and e = 0.339

Table 2: Comparison of the effect of parameters θ and d on effective thermal conductivity

Sample Combination	Effect of d on k_{eff} at $\theta = \pi/4$				Effect of θ on k_{eff} at d=0.1655			
	d	percentage increase in d	k_{eff} W/mK	Percentage increasing in k_{eff}	θ	percentage increase in θ	k_{eff} W/mK	percentage increase in k_{eff}
RVC-air	0.1655 0.2126	28%	0.1555 0.2047	31%	$\pi/4$ 0.4π	60%	0.1555 0.2071	33%
RVC-water	0.1655 0.2126	28%	0.7615 0.8071	5.9%	$\pi/4$ 0.4π	60%	0.7615 0.8235	8%

Effect of Porosity on the Heat Transfer Coefficient

By applying the new analytical solution for effective thermal conductivity, the thermal performance of novel open-cell metal foam can be investigated under low Reynolds numbers. The heat transfer coefficient will be chosen to evaluate the enhancement of the heat transfer rate. The research of Bhattacharya and Mahajan (2002) shows that the ratio of the heat transfer coefficient for the flow through a packed channel of high porosity metal foam to that of an open channel can be written as:

$$h_o/h_c = \left(k_{eff}/k_f\right)^{0.6} \tag{15}$$

For the case of Al-air at porosity 0.971, the thermal conductivity ratio of the foam media to fluid is 130, and the ratio of heat transfer coefficients is predicted as 18.5 by eq. (13). As a result, by increasing porosity, the ratio of heat transfer coefficients is decreased. The variation of h_o/h_c with porosity for Al-air and Al-water samples is plotted in Fig. 6.

These results show that, compared to different air cooler designs, the capability of foam heat exchangers in increasing the heat transfer rate is superior.

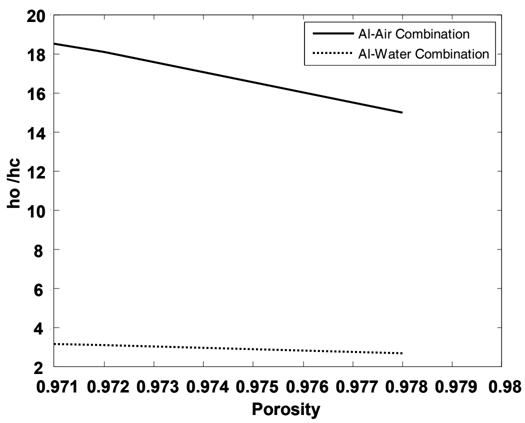

Figure 6: Variation of h_o/h_c versus ε for Al-Air and Al-Water samples, foam characteristics b/L=1, $\theta=\pi/4$ and e = 0.339

CONCLUSIONS

A new analytical model was developed for effective thermal conductivity calculation of non-isotropic open-cell metal foam based on half a tetrakaidecahedron as the appropriate REV. The results show that k_{eff} of the foam is a function of three independent dimensions, L, b, θ, porosity and diameter of the spherical solid lump. Fiber cross-section varies with increasing porosity from a circle to triangle. This factor was incorporated into the new proposed model. The previous model proposed by Boomsma and Poulikakos (2001) is confined to isotropic geometry ($\theta=45^o$) by assuming equal sizes of all ligaments of the unit-cell (b=L) and circular shape of fiber cross section in the porosity range of $0.88 \prec \varepsilon \prec 0.98$, which is not strictly correct for all cases. These deficiencies are resolved in the proposed model, in addition to increasing accuracy relative to previous model of Boomsma and Poulikakos (2001). It was found that k_{eff} increases significantly with decreasing porosity and increasing geometrical parameters d and θ. The ratio of the heat transfer coefficient for flow through a channel packed with open-cell metal foam to an open channel is a function of the ratio of the effective thermal conductivity of foam to fluid conductivity. For the case of Al-air at porosity 0.971, the thermal conductivity ratio of foam media to fluid is 130, and the ratio of heat transfer coefficients is 18.5.

With regard to the aforementioned results, it can be demonstrated that inclusion of high porosity open-cell metal foam into the structure of commercially available air coolers enhances the effective thermal conductivity, the heat transfer coefficient and the thermal performance considerably. The relationship between conductivity and foam geometrical specifications is very important for efficient design of foam heat exchangers. Gaining in-depth knowledge of the effect of the geometric parameters of foam on conductivity is important in enhancing thermal performance and efficient design of foam heat exchangers.

ACKNOWLEDGEMENTS

The authors wish to express their thanks and gratitude to the National Iranian Oil Refining & Distribution Company, NIORDC, and its management for their financial support, which made the completion of this work possible.

NOMENCLATURE

A	Edge cross-sectional area
b	Length of the sides of horizontal square faces
D	The width of the unit cell
d	Dimensionless foam ligament radius or height
d_p	Pore diameter
e	Dimensionless spherical node diameter
H	Cell height
k_f	Thermal conductivity of fluid
k_s	Thermal conductivity of solid matrix
k_{eff}	Effective thermal conductivity of the porous medium
L	Ligament length

r	Diameter of spherical nodes
R	Radius or height of cross-section foam ligament
V_{tot}	›lume of selected REV

Greek Symbols

ε	Porosity of porous media
θ	The inclination angle that defines the orientation of the hexagonal faces with respect to the rise direction

REFERENCES

Bhattacharya, A. and Mahajan, R. L., Finned metal foam heat sinks for electronics cooling in forced convection. J. Electronic Packaging, 124, 155-163 (2002).

Bhattacharya, A., Calmidi, V. V. and Mahajan, R. L., Thermophysical properties of high porosity metal foams. International Journal of Heat and Mass Transfer, 45, 1017-1031 (2002).

Boomsma, K. and Poulikakos, D., On the effective thermal conductivity of a three dimensionally structured fluid-saturated metal foam. International Journal of Heat and Mass Transfer, 44, 827-836 (2001).

Boomsma, K., Poulikakos, D. and Zwick, F., Metal foams as compact high performance heat exchangers. Mechanics of Materials, 35, 1161–1176 (2003).

Calmidi, V. V. and Mahajan, R. L., The effective thermal conductivity of high porosity fibrous metal foams. Transactions of the ASME, 121, 466-471 (1999).

Calmidi, V. V. and Mahajan, R. L., Forced Convection in High Porosity Metal Foams. Journal of Heat Transfer, 122, 557-565 (2000).

Dukhan, N., Quinones-Ramo, P. D., Cruz-Ruiz, E., Velez-Reyes, M. and Scott, E. P., One-dimensional heat transfer analysis in open-cell 10-ppi metal foam. International Journal of Heat and Mass Transfer, 48, 5112-5120 (2005).

Dukhan, N. and Chen, K., Heat transfer measurements in metal foam subjected to constant heat flux. Experimental Thermal and Fluid Science, 32, 624-631 (2007).

Ghosh, I., Heat transfer analysis of high porosity open-cell metal foam. Journal of Heat Transfer, 130, 1-6 (2008).

Haghighi, M. and Kasiri, N., A three dimensional analytical model for the effective thermal conductivity and porosity in high porosity metal foam using a non-isotropic unit-cell. J. Enhanced Heat Transfer (2009). (In press).

Kwon, Y. W., Cooke, R. E. and Park, C., Representative unit-cell for open-cell metal foams with or without elastic filler. Material Science and Engineering, A343, 63-70 (2003).

Lu, T. J., Stone, H. A. and Ashby, M. F., Heat transfer in open-cell metal foams. Acta Mater, 4 (10), 3619-3635 (1998).

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

Phanikumar, M. S. and Mahajan, R. L., Non-Darcy natural convection in high porosity metal foams. International Journal of Heat and Mass Transfer, 45, 3781-3793 (2002).

Salas, K. I. and Waas, A. M., Convective heat transfer in open cell metal foams. Journal of Heat Transfer, 129, 1217-1229 (2007).

Sullivan, R. M., Ghosn, L. J. and Lerch, B. A., A general tetrakaidecahedron model for open-cell foams. Int. J. of Solid and Structures, 45, 1754-1765 (2008).

Zhao, C. Y., Lu, T. J., Hodson, H. P. and Jackson, J. D., The temperature dependence of effective thermal conductivity of open-cell steel alloy foams. Material Science & Engineering, A 367, 123-131 (2004).

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