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Heat transfer correlation for high-porosity open-cell foam

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ABSTRACT

Heat transfer correlation has been developed, assuming a simple cubic model of open cell foam. The cross-linked porous structure is equivalent to a bundle of independent, slender tubes with protrusions. It may be presumed that the projected struts either cause an increase in heat transfer coefficient or they act as extended surfaces or fins. The effect of these protruded filaments has been judiciously integrated within the empirical correlation for cross-convective flow over the bank of smooth tubes. An augmentation in foam heat transfer coefficient has occurred due to cross-connections in struts. An excellent agreement has been observed between the predicted correlations and the existing ones which have been established through experimental data.

1. Introduction

Porous media with porosity ranging 0.3 and 0.6 has been researched for many decades and an excellent review of that study has been summarised by Kaviany [1]. However, high-porosity metal foam with porosity more than 0.9 is a relatively new development [2]. Consequently, different models have been evolved to analyse open porous metal foam – a promising alternative as extended heat transfer surface for thermal application.

Many researchers have assumed that the convective fluid and porous matrix are forming a continuum with local equilibrium between them [3–5]. In the recent year, Dukhan et al. [6] have presented a macroscopic lumped-parameter engineering treatment to determine the temperature distribution in open-cell metal foams used in forced convective heat transfer application. The analysis has developed closed form solution for a slice of foam being used as extended surface. However, the validity of the models is restricted to low Reynolds number application where the concept of local equilibrium is relevant [7,8].

In an alternative approach, one has to consider non-equilibrium between the convective fluid and the foam elements in microscopic dimension. When viewed in that scale, the complexity of the foam geometry renders the modelling of heat transfer and fluid flow difficult. "Simple cubic" representation of porous material can introduce simplicity for this complex geometrical structure [7,9,10]. In this model, foam is assumed to be a repetitive cubic block made with slender tubes of diameter d_f and length d_p as shown in Fig. 1. Lu et al. [7] obtained a functional relationship between the cellular structure and the heat-transfer characteristics for the forced convective flow through open-cell metal foams. The analytical model further simplifies cubic cell structure into a bundle of unidirectional tubes without any cross-links. That resembles flow across bank of cylinders. More recently, convective heat transfer in high porosity metal foam has been analysed assuming simple cubic model by Ghosh [11]. The model, in the microscopic level takes account of the forced convective heat transfer coupled with heat conduction through the interconnected foam fibres.

A complementary part of heat transfer modelling is the selection of appropriate thermo-hydraulic data in terms of the Colburn *j*-factor or the Nusselt number. These dimensionless parameters generated using different models, for the obvious reason, are dissimilar. Lu et al. [7] have developed a correlation for the overall heat transfer coefficient using "cubic model". Their predicted average Nusselt number closely resembles the same for cross convective flow through cylinder banks. Fuller et al. [9] have adopted steady-state technique to calculate volumetric heat transfer coefficient. Kim et al. [12] have experimentally determined *j*-factor involving the Darcy number, and foam geometrical parameters viz., length, height of the sample, etc. Hsieh et al. [13] have developed experimentally a set of correlations between Nusselt number and Reynolds number for six types of aluminium-foam heat sinks. Giani et al. [14] conducted mass transfer experiments to generate a correlation for metal foams using Colburn analogy. They are able to re-establish their results in a separate transient heat transfer experiment [15]. Interestingly, the empirical correlations for the dimensionless heat transfer coefficients are often quite similar to that of forced convective fluid flow across bank of tubes.

In simple cubic structure model of metal foam, heat transfer due to *cross-connectivity* in the struts is often neglected. In absence

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Nomenclature

A_c	cross-sectional area of strut
d_f	Strut diameter
d_p	pore size
H	height of foam
h	heat transfer coefficient
h_{tube}	heat transfer coefficient for bank of tubes
k_f	thermal conductivity of foam material
k_∞	thermal conductivity of convective fluid
L	length of foam
m	factor defined in Eq. (3)
M	factor defined in Eq. (6)
Nu	Nusselt number
PPI	pores per inch
P	perimeter of strut
Pr	Prandtl number
q_x, q_y, q_z	heat transfer through strut along x, y, and z *direction,* respectively
q_f	total heat transfer
Re	Reynolds number based on d_f
Sh	Sherwood number
Sc	Schmidt number
T_1	temperature of base plate
T_∞	temperature of convective fluid

Greek symbols

ε	porosity
η_f	foam efficiency
$\eta_{1/2}$	efficiency of half strut
θ	temperature differential between foam filament and convective fluid

of inter-connections in the ligaments, foam transforms into familiar geometry of tube-bundles. This transformation enables to make use of the thermo-hydraulic data [16] for convective fluid flow across bank of tubes. However, simplification caused by relaxing one of the key features of porous material leads to erroneous result. The present article assumes porous media as a continuously connected structure while deriving a dimensionless heat transfer correlation for the micro-structure based model proposed in Ref. [11]. The heat transfer correlation has been developed judiciously in terms of the existing empirical equations for the cross-convective flow over the tube bundles. Suitability of the proposed heat transfer correlation for other models has also been verified and a coherent outcome has been observed.

2. Foam heat transfer model

In metal foam continuously connected slender filaments form an open-celled structure. High-porosity metallic foams are characterised by the cell shape, cell size, relative density, properties of the cell wall and the degree of anisotropy. However, manufacturers commonly specify two parameters to characterise the open-cell foam structure; pore size (in pores per inch or PPI) and relative density. The metal foam microstructure described by a complex geometry renders modelling of the heat transfer and fluid flow very complicated. In mathematical models, this can be dealt with by assuming a simplified and repeated configuration (Fig. 1) and describing the microstructure using macroscopic properties like the pore size and relative density. The commonly used geometry for unit foam cell is to consider a cube of slender tubes of diameter d_f, and length d_p, as shown in Fig. 1 [7,9,10]. The strut diameter (d_f) can be related to pore size (d_p) of cubic cell by the following relation [10]:

$$\frac{d_f}{d_p} = 2\sqrt{\frac{1-\varepsilon}{3\pi}}\frac{1}{G} \quad \text{where, } G = 1 - e^{-(1-\varepsilon)/0.04} \tag{1}$$

Assuming the aforesaid simple cubic structure of the foam, a comprehensive analysis of forced convection heat transfer in high porosity metal foam has been presented earlier [11]. The microstructure-based heat transfer model takes account of the heat conduction through filaments of the foam in conjugation with convective heat flow over the struts in all directions. It has been assumed that the open-cell is made of uniformly distributed, equal-sized cubic cells attached to a plate at T_1. Fluid stream temperature, T_∞ remains constant over the element of foam under consideration. The analysis [11] shows that the foam can be imagined as a bunch of several independent x-struts with projections along the y and z directions. The protrusions along the y and z direction are of length ($d_p/2$). Being able to virtually partition y and z-struts into two parts, it has been possible to evaluate the heat transfer occurring through these cross-links. The q_z and q_y are given by the following relation [11]:

$$\begin{aligned} q_z &= q_y = \sqrt{hPk_fA_c}(T_x - T_\infty)\left[\frac{\cosh(md_p)-1}{\sinh(md_p)}\right] \\ &= hPd_p(T_x - T_\infty)\eta_{1/2} \end{aligned} \tag{2}$$

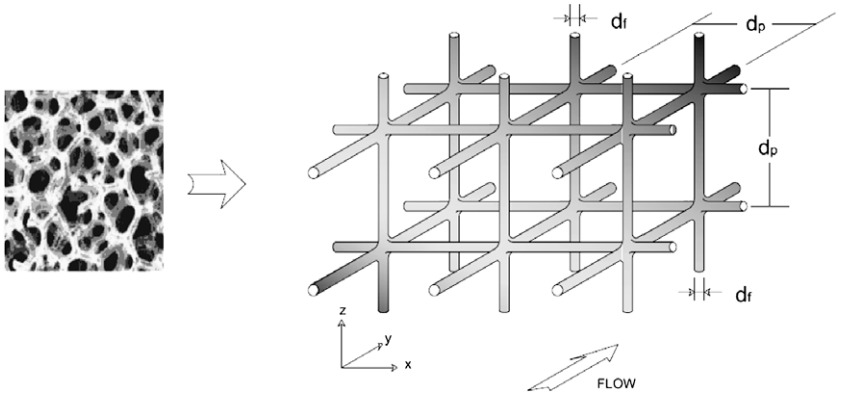

Fig. 1. Simple cubic representation of porous foam.

where,

$$\eta_{1/2} = \frac{\tanh m(d_p/2)}{m(d_p/2)}, \quad P = \pi d_f, \quad A_c = \frac{\pi d_f^2}{4}, \quad m = \sqrt{\frac{4h}{k_f d_f}} \tag{3}$$

Applying energy balance at the nodal point N_0 (Fig. 2), the governing differential equation for the temperature difference (θ) between the solid and the convective fluid can be finally written as

$$\frac{dq_x}{dx} + hP(T_x - T_\infty)\left(1 + 4\eta_{1/2}\right) = 0 \tag{4}$$

Substituting, $q_x = -k_f A_c \frac{dT_x}{dx}$ and putting, $\theta = (T_x - T_\infty)$ in Eq. (4) one can obtain,

$$\frac{d^2\theta}{dx^2} - M^2\theta = 0 \tag{5}$$

where,

$$M = \sqrt{\frac{hP}{k_f A_c}\left(1 + 4\eta_{1/2}\right)} = m\sqrt{(1 + 4\eta_{1/2})} \tag{6}$$

Differential Eq. (5) is analogous to the ordinary fin equation, except that the value of 'm' has been increased by $\sqrt{1 + 4\eta_{1/2}}$. This factor is due to the heat transfer through cross-connecting y and z-struts.

Eq. (5) can be solved assuming adiabatic boundary condition, $(d\theta/dx) = 0$ at $x = L$ and a general boundary condition $\theta = (T_1 - T_\infty) = \theta_1$ at $x = 0$

$$\theta = \theta_1 \frac{\cosh M(L - x)}{\cosh ML} \tag{7}$$

Eq. (7) has been used to estimated the amount of heat getting transferred from the foam base

$$q_f = \left(\frac{H}{d_p}\right)\left(\frac{W}{d_p}\right)\sqrt{hPk_f A_c(1 + 4\eta_{1/2})}\theta_1 \tanh(ML) \tag{8}$$

where, $\left(\frac{H}{d_p}\right)\left(\frac{W}{d_p}\right)$ factor stands for the total number of struts within height, H and width, W of the foam.

The efficiency of foam as extended heat transfer surface with adiabatic condition at the tip becomes [11]

$$\eta_f = \tanh(ML)/ML \tag{9}$$

Based on the abovementioned analysis, a parametric study involving variation of Nusselt number, pore density (PPI), and porosity (ε) has been reported earlier in Ref. [11]. But, whenever an exact heat transfer solution is sought, Nusselt number must be evaluated using an appropriate expression. The following analysis attempts to establish a correlation for this dimensional heat transfer coefficient.

In contrast to this microstructure based model, the '*lumped parameter*' engineering treatment considers foam as a continuum made of porous construction filled with convective fluid. The effective thermal conductivity of such formation can be represented as $(1 - \varepsilon)\ k_f + \varepsilon k_\infty$. The temperature variation in the metal foam, according to their analysis, is obtained through the following relation [6].

$$\theta = \theta_1 \frac{\cosh m_{fm}(L - x)}{\cosh m_{fm} L} \tag{10}$$

where, $m_{fm} = \sqrt{\frac{h_{fm}\sigma}{(1-\varepsilon)k_f + \varepsilon k_\infty}}$ has been defined as the 'foam parameter'.

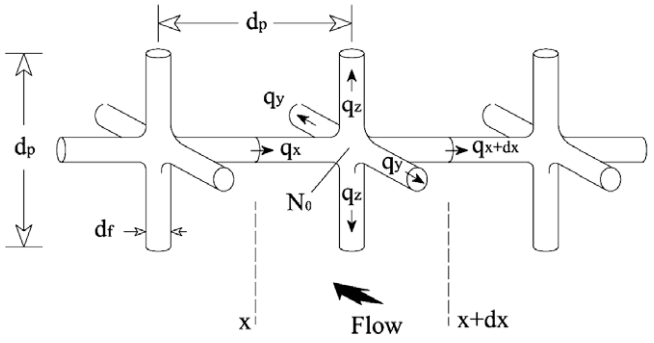

Fig. 2. Differential element for deriving governing foam equation.

3. Predicted heat transfer correlation

In general, heat transfer coefficient, being a thermo-physical property, should be unique as long as the geometry of the flow conduit and the flow rate remain constant. However, these parameters (or the correlations) may have dissimilar values because they are generated using different models. Thus, one must be careful while selecting appropriate heat transfer coefficient.

In the previous section, it has been envisaged that the highly interconnected foam structure can be disintegrated into a group of independent x-struts with protrusions of length $(d_p/2)$ along y and z directions. Essentially, when foam becomes a cluster of several uneven micro-tubes, it may be presumed that the projected struts are either causing an increase in heat transfer coefficient or they are acting as extended heat transfer surfaces. When the projected filaments act as fins, the heat transfer through the individual struts has to be considered independently, as it is done in the 'microstructure' based model. Alternatively, the heat transfer coefficient (h_{tube}) for the cross-convective flow over the bank of smooth tubes has to be modified suitably; so that the effect of roughness caused due to protruding struts is taken care. If h_{tube} could be corrected, transportation of heat through the protrusions need not be considered separately. In this situation, foam can be treated as a 'bulk body' with known convective heat transfer coefficient. An attempt is made in the following section to amend h_{tube}, so that it includes the effect of protruding struts.

3.1. Enhanced heat transfer coefficient approach

Fig. 3 shows the partial view of a section of unit foam cell. Imagine that the extreme left end of the foam structure is attached to an isothermal surface at T_1. If the fluid flow is occurring along the y direction, the x- and z-struts are at right angle to the direction of flow. The y-strut, on the other hand, is aligned to the flow direction. Therefore, if one thinks of using h_{tube} *as the foam heat transfer coefficient*; there are contributions from the x- and z-struts only. The y-strut being parallel to fluid flow does not have any role. On the other hand, when fluid is flowing over the tubes, the x- and z-struts are all alike. Fluid can not differentiate between them because both of them are aligned perpendicular to the flow. Since, x and z micro-tubes are indistinguishable by the fluid, it can be imagined that there is an inflated single tube at right angle to its path. If the cross-sectional area (and hence the total heat transfer surface) of the filaments remain invariant during this transformation, the diameter of the resulting single-strut becomes effectively $\sqrt{2}d_f$. When this exaggerated micro-tube is subjected to cross-convective flow, the amount of heat dissipation becomes

$$h_{tube}P'(T_x - T_\infty) \tag{11}$$

where, $P' = \pi(\sqrt{2}d_f)$, is the perimeter of the inflated micro-tube. Substituting the value of P' in Eq. (11), one obtains

$$\sqrt{2}h_{tube}P(T_x - T_\infty) \tag{12}$$

The use of heat transfer coefficient h_{tube} in open-cell metal foam thus needs amplification by a factor of $\sqrt{2}$. It arises due to

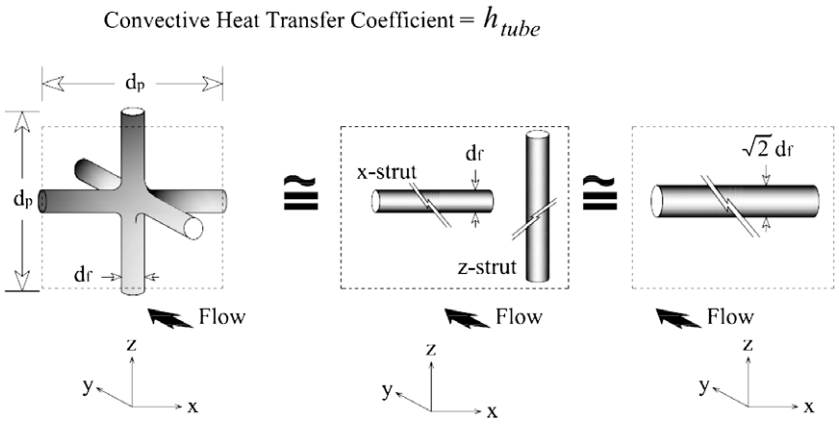

Fig. 3. Increase in the effective strut diameter.

cross-connecting struts. Once the contribution of the protruded struts has been included within the heat transfer coefficient; foam can be treated as a 'bulk' medium having effective heat transfer coefficient given by $\sqrt{2}h_{\text{tube}}$. The empirical equations relating h_{tube} are well-established and documented [16] and can be used in the appropriate Reynolds number range.

3.2. Extended heat transfer surface approach

When the protruding struts are viewed as extended surfaces (fins), there is additional convective heat transfer due to cross-connecting ligaments. Microstructure based model [11] is able to quantify the total convective heat dissipation occurring through *all* the struts in the unit block (Fig. 2)

$$\left(1+4\eta_{1/2}\right)hP(T_x-T_\infty) \tag{13}$$

where, h is the interstitial heat transfer coefficient.

Eqs. (12) and (13) signify the same convective heat transfer in different form. Equating them, one can obtain the heat transfer coefficient for the microstructure model

$$h=\left(\sqrt{2}h_{\text{tube}}\right)\Big/\left(1+4\eta_{1/2}\right) \tag{14}$$

If the foam ligaments were not interlinked, the simple cubic structure transforms into a bundle of x cylinders. As a result, transverse flow of heat, q_y and q_z do not appear in the original energy balance and the additional term $4\eta_{1/2}$ also does not exist in Eq. (14). In absence of cross-links, enlargement of the strut diameter does not occur. Consequently, the interstitial heat transfer coefficient (h) gets modified into a similar expression for tubular fin (x-struts only) subjected to cross-flow

$$h\rightarrow h_{\text{tube}} \tag{15}$$

The validity of the proposed heat transfer coefficient, derived through the microscopic model of foam and 'bulk' model, has been verified with the available experimental correlations of similar kind. Literature review reveals that Giani et al. [14,15] separately conducted heat and mass transfer experiments to determine correlations for heat transfer. They have obtained the following correlation for Sherwood number (Sh) through the mass transfer investigation [14]:

$$Sh=1.1Re^{0.43}Sc^{1/3}\ \text{for}\ 10<Re<100 \tag{16}$$

Using Colburn analogy, the Nusselt number can be written as

$$Nu=1.1Re^{0.43}Pr^{1/3}\ \text{for}\ 10<Re<100 \tag{17}$$

In a separate experimental heat transfer test setup, they have also obtained [15] the following correlation for Nusselt number

$$Nu=1.2Re^{0.43}Pr^{1/3}\ \text{for}\ 20<Re<240 \tag{18}$$

In the similar Reynolds number range, the correlation relating h_{tube} can also be obtained from Ref. [16]

$$Nu_{\text{tube}}=0.8Re^{0.4}Pr^{1/3}\ \text{for}\ 10<Re<100 \tag{19}$$

It may be noted that the Reynolds number in Eqs. (17) and (18) are based on the superficial or the free stream gas velocity and the strut diameter, while the same in Eq. (19) is estimated on the basis of interstitial fluid velocity and the strut diameter.

The effective heat transfer coefficient, in the dimensionless form, can be written as:

$$Nu_{\text{eff}}=\sqrt{2}Nu_{\text{tube}} \tag{20}$$

When Nu_{tube} from Eq. (19) is substituted in Eq. (20), the resulting equation become

$$Nu_{\text{eff}}=1.13Re^{0.4}Pr^{1/3}\ \text{for}\ 10<Re<100 \tag{21}$$

Eq. (21) clearly shows that the predicted heat transfer correlation, in the dimensionless form, is remarkably similar to that obtained through experimental results (Eqs. (17) and (18)). Fig. 4 shows a comparison between the experimental and the predicted correlations. The predicted results are close to those obtained in mass transfer experiments.

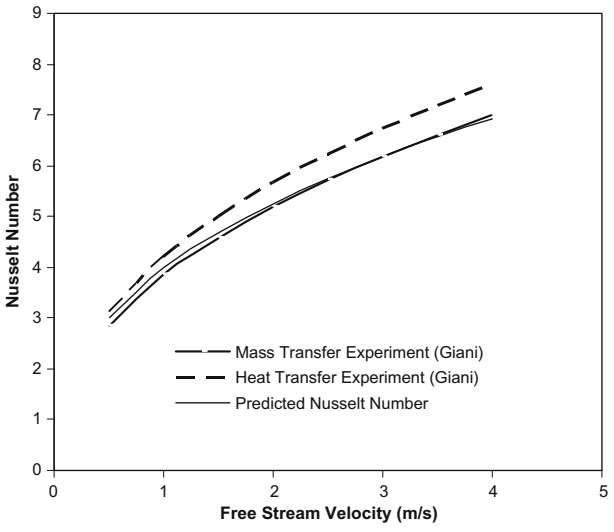

Fig. 4. Comparison between the experimental and predicted Nusselt number.

4. Results and discussions

The micro-structure model with appropriate heat transfer correlation is suitable for parametric studies involving variation in temperature profile in the metallic foam and the total amount of heat being transferred from the foam base. Variables for the analysis include foam porosity (ε), pore density (in PPI) and the velocity of air, being considered as the convective fluid.

Assuming that the foam is made of aluminium, the thermal conductivity of foam strut can be taken as 216 W/m K. Length (L) of the foam block is an additional parameter necessary to estimate the temperature profile, (θ/θ_1) or the total amount of heat transfer (q_f) occurring through the foam base. The length (L) has been taken arbitrarily as 10 mm.

Results obtained for the variation in metal–fluid temperature differential and the total heat transfer occurring through the foam have also been compared with the same obtained with '*lumped-parameter engineering*' model proposed by Dukhan et al. [6].

In a porous media, the interstitial velocity of the convective fluid is higher than the superficial velocity. The variation in the former velocity type depends on the shape and size of the body across which fluid is flowing. It is because of this reason, the superficial velocity has been considered as the variable to compare different models. However, the calculation of Reynolds numbers (Re) is based on the velocity recommended with the empirical relations. As for example, the correlation in Eqs. (17) and (18) are based on superficial velocity, while the same in Eq. (19) has the basis of the interstitial one.

The variation in normalised temperature difference (θ/θ_1) with superficial fluid velocity at ($x/L = 0.5$) has been plotted in Fig. 5. Predictions of the micro-structure model using the proposed heat transfer correlation in Eq. (14) are in good agreement with the same obtained through 'lumped-parameter engineering' model using the experimental Nusselt number expression in Eq. (17).

Since the porosity and pore density (PPI) are kept constant, the variation in normalised temperature differential decreases with convective fluid flow. Enhanced fluid flow augments heat transfer (shown in Fig. 6) causing a fall in the temperature differential between foam and convective fluid.

The variation of foam-efficiency (η_f) with superficial fluid velocity has been plotted in Fig. 7. It decreases monotonically with the superficial velocity and almost follows the nature of variation of (θ/θ_1).

The effect of pore density (PPI) on heat transfer has been analysed keeping the length (L) of the foam and its porosity (ε) constant. Fig. 8 shows the variation of normalised temperature differential (θ/θ_1) in foam matrix as a function of pore density (PPI) assuming that the length of the foam, $L = 0.01$ m, $x = L/2$ and foam density, $\varepsilon = 0.85$. The figure shows that the temperature ratio (θ/θ_1) steadily decreases with pore density (PPI). An increase in pore density (PPI) can be characterised by reduction in pore diameter (d_p) and a corresponding augmentation in heat transfer area density. An enhancement in heat transfer area causes more heat exchange through the foam base and hence the temperature differential between the gas and solid phase falls. Similar variation of temperature difference between the solid and the gas phase with

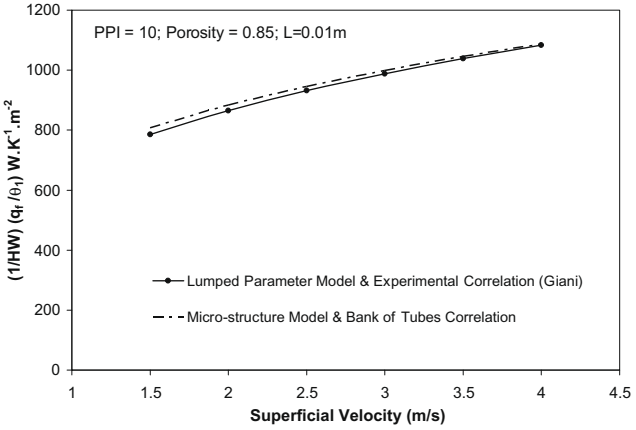

Fig. 6. Variation in heat flux with superficial velocity.

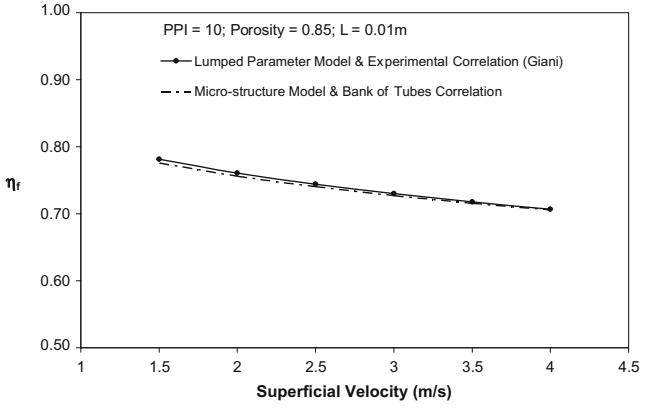

Fig. 7. Variation of foam-efficiency with superficial velocity.

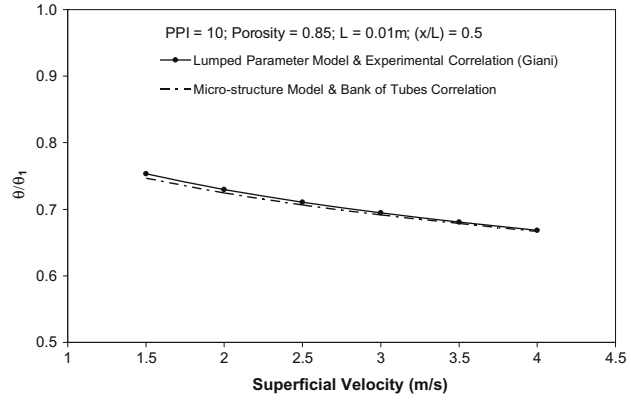

Fig. 5. Normalised temperature variation with superficial velocity.

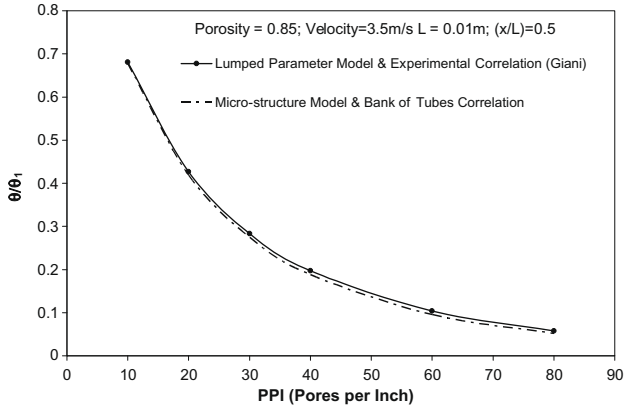

Fig. 8. Normalised temperature variation with pore density (PPI).

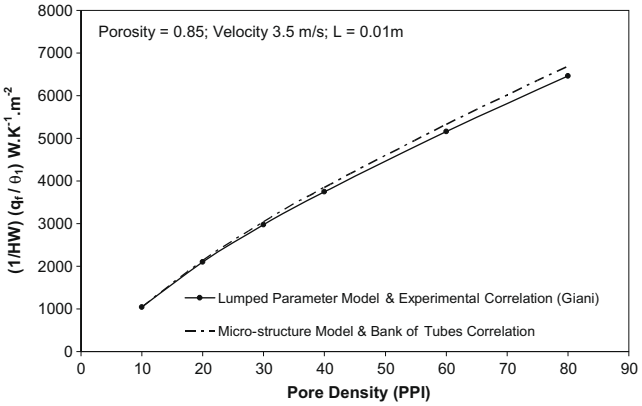

Fig. 9. Variation in heat flux with pore density (PPI).

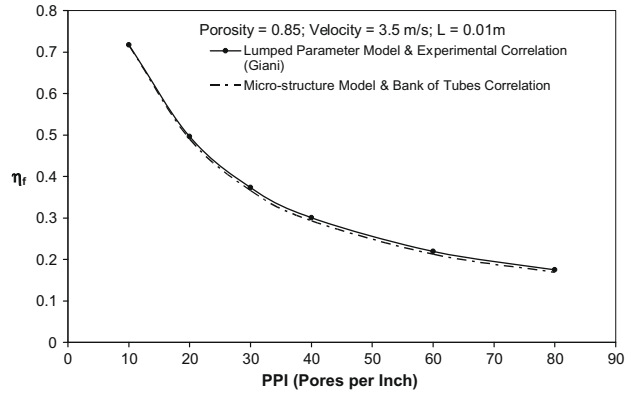

Fig. 10. Variation of foam-efficiency with pore density (PPI).

pore density has been experimentally reported in Ref. [13]. Results obtained using different models are almost alike.

Variation in heat flux through the foam base and the foam efficiency as a function of pore density (PPI) are shown in Figs. 9 and 10, respectively. Excellent match between the predicted results from different heat transfer models is discernible.

Finally, a parametric variation of porosity (ε) on heat transfer in metal foam has been analysed. The foam density (PPI) and the superficial velocity have been kept at 10 PPI and 3.5 m/s, respectively. The ratio (θ/θ_1), heat flux and foam efficiency (η_f) have been found to decrease with porosity (ε). Higher porosity foam has slightly thinner strut diameter (d_f), if the other parameters remain

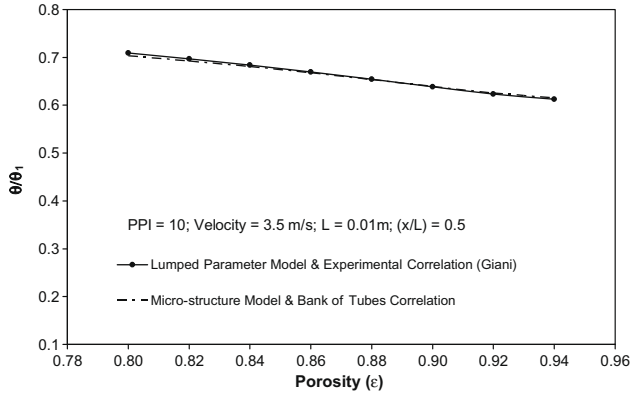

Fig. 11. Normalised temperature variation with porosity (ε).

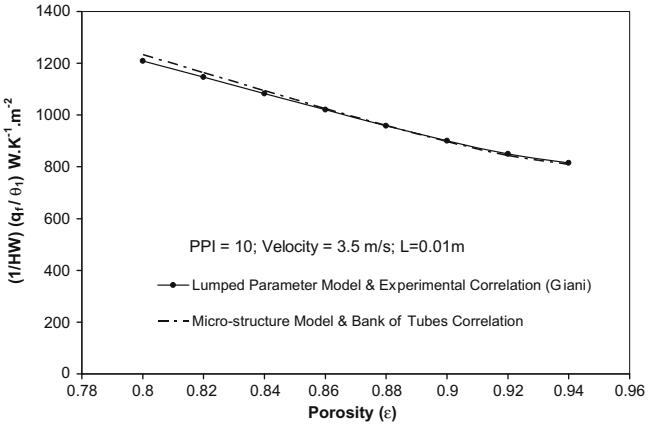

Fig. 12. Variation in heat flux with porosity (ε).

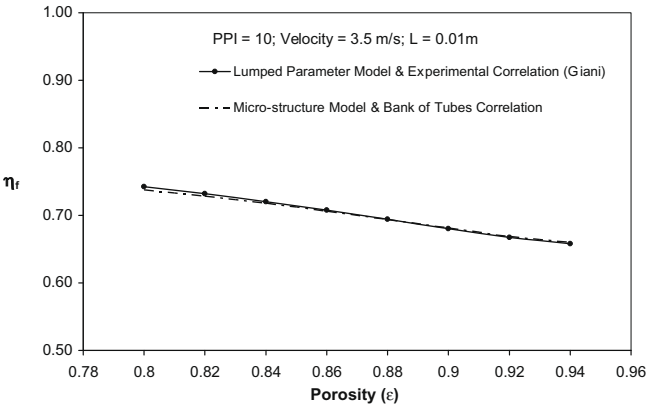

Fig. 13. Variation of foam-efficiency with porosity (ε).

constant. It may be noted from Eq. (3) that the reduction in d_f can enhance the factor 'm' and consequently transportation of heat. Therefore, the dimensionless temperature differential (θ/θ_1) reduces with the high porosity metal foam (Fig. 11). However, an increase in porosity is also associated with an overall decrease in heat transfer area density causing a reduction in heat flux through the foam (Fig. 12). Finally the variation of foam efficiency as a function of porosity has been shown in Fig. 13.

5. Conclusions

An in-depth analysis with simple cubic structure of open-cell foam provides a perceptive view of the heat transfer mechanism in the porous medium. Complexity of the foam heat transfer modelling has been significantly reduced using this simplified model.

Since it is possible to virtually disintegrate highly connected foam structure into a group of independent x-struts with protrusions along the y and z directions [11], the projected filaments can either cause an increase in overall heat transfer coefficient or they may act as extended heat transfer surface area. When protruded struts act as fins, there is an overall increase in heat transfer by a factor of $(1 + 4\eta_{1/2})h$; where the heat transfer coefficient is given by $(\sqrt{2}h_{\text{tube}})/(1 + 4\eta_{1/2})$.

In an alternative approach, the effect of roughness caused due to projected micro-tubes can be included within overall heat transfer coefficient. In order to do that, the heat transfer coefficient (h_{tube}) for the cross-convective flow over the bank of smooth tubes can be modified suitably. Effectively, there is an increase in heat transfer coefficient by a factor of $\sqrt{2}$ over the same for smooth

tubes. However, the final outcome from both the models is effectively the same. A very good agreement between the predicted results and the available experimental data has also been observed.

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