BOUNDARY HEAT FLUX BIFURCATION FOR LOW REYNOLDS NUMBER FLUID CONVECTION THROUGH HIGH POROSITY OPEN-CELL METAL FOAM

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ABSTRACT

In this work, the heat flux bifurcation between porous matrix and fluid at the heated boundary for low Reynolds number (Re) convective flow in high porosity open-cell metal foam is studied with pore-scale numerical simulation. The pore structure of metal foam is created by employing the Weaire-Phelan cellular model. Liquid paraffin is taken as the fluid, which flows through the pore space of metal foam and simultaneously receives heat from both heated plate and foam skeleton. By solving Navier-Stokes and energy equations, the effects of pore Re, solid to fluid thermal conductivity ratio (k_s/k_f), thermal resistance (R_t) of heated plate and foam porosity (ε_f) on boundary heat flux bifurcation are addressed. It is found that within the heated plate there exist remarkable tangential heat flows, from the region close to liquid paraffin to the region near foam skeleton. As a result, the heat flux at the heated boundary connected to metal foam is much higher than that in contact with fluid. For relatively lower k_s/k_f and lower Re, the influences of Re and R_t on solid to fluid heat flux ratio (Q_s/Q_f) at the heated boundary are quite insignificant. For relatively higher k_s/k_f and higher Re, Q_s/Q_f increases with the increase of Re and the decrease of R_t. Moreover, Q_s/Q_f increases with the increase of k_s/k_f and the decrease of foam porosity. The present pore-scale results are further compared with available intuitive model in the literature, and the corresponding discrepancy in Q_s/Q_f is observed and discussed. This study can provide guidance to better consider the boundary heat flux information for local thermal non-equilibrium modeling of heated metal foam-fluid system with low Re convection, which finds applications in thermal energy storage with solid-liquid phase change.

Keywords: heat flux bifurcation; thermal boundary; metal foam; low Reynolds number; pore scale.

1. INTRODUCTION

High porosity open-cell metal foam has been found a very effective enhancer for improving thermal behaviour of solid-liquid phase change material (PCM) [1]. As the thermal diffusivity of metal foam skeleton is several orders of magnitude higher than that of PCM and the PCM flow is weak, local thermal non-equilibrium (LTNE) volume-averaging modelling of low Reynolds number (Re, at pore scale) convection in metal foam/PCM composite porous media has attracted increasingly attentions [2, 3]. When using the LTNE model, two energy equations for foam region and fluid region are involved and each energy equation needs a thermal boundary condition. If constant heat flux condition is encountered, the boundary heat flux bifurcation between porous metal foam and PCM regions needs to be carefully considered.

Focusing on LTNE modelling of porous media, Amiri et al. [4] proposed two simplified methods to treat the boundary flux bifurcation. The first one assumed that the heat flux was distributed to the two phases according to their thermal conductivities and wall temperature gradients. The second explicitly considered locally equal heat flux for each phase. Kim and Kim [5], and Alazmi and Vafai [6] examined these two methods and pointed out their limitations, which indicated that the mechanisms of heat flux bifurcation with a constant heat flux wall condition should be investigated in depth to consider more influencing factors besides thermal conductivities and wall temperature gradients of solid and liquid phases. To better understand how the heat flux bifurcates, pore-scale simulation of fluid flow and heat

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transfer in porous structures proved to be effective. Jiang and Lu [7] performed a pore-scale numerical study on the thermal boundary characteristics of convection in packed bed, which showed the contact conditions had great influence on boundary heat flux bifurcation. Through direct simulation at pore scale, Imani et al. [8] estimated the heat flux bifurcation at the heated boundary of a porous media with fin array filled in a channel. The results indicated that the foam porosity, Reynolds number and solid to fluid thermal conductivity ratio all had evident influences on the boundary flux bifurcation. By comparing pore-scale computation results with those from macro-scale LTNE modeling, Ouyang et al. [9] validated a thermal boundary condition model considering tangential thermal resistance in the solid wall for LTNE thermal analysis of convection in a channel filled with inline pin-fins. However, it should be pointed out that the above works have not addressed the influencing factors responsible for the metal foam/PCM system, in which the foam pore structure and low Re convection are no doubt key aspects requiring to be explored at pore scale.

In this research, the boundary heat flux bifurcation between metal foam and PCM at the heated wall for low Re convection of liquid PCM in metal foam will be investigated through pore-scale direct simulation. A relatively realistic foam model represented by the Weaire-Phelan structure is adopted to consider the structural influence of metal foam. And low velocity liquid PCM flowing through the porous metal foam attached to a heated metal plate is modelled by solving conservation equations of fluid flow and heat transfer. The effects of pore Reynolds number, foam porosity, foam material and thermal resistance of heated plate on the boundary heat flux bifurcation are emphatically investigated. This work is expected to provide guidance to reasonably consider the boundary heat flux conditions when modelling heated metal foam-PCM system based on LTNE volume-averaging method.

2. PHYSICAL MODEL AND METHOD

Figure 1 shows the pore-scale physical problem corresponding to heat flux bifurcation at the heated boundary for low Re flow of liquid PCM in metal foam. The metal foam structure (5.08 mm ×5.08 mm ×5.08 mm) constructed based on Weaire-Phelan model [10] is attached to a heated plate (HP, 5.08 mm ×5.08 mm × 0.5 mm) with a prescribed constant heat flux at the bottom surface. Liquid PCM flows through the pore space of metal foam and receives heat along the fluid path. Due to bifurcation of heat flux, different portions of total heat are transported to metal foam skeleton (Q_s) and fluid (Q_f), respectively. The ratio of Q_s to Q_f will be investigated under different conditions, including various pore Re, foam-to-fluid thermal conductivity ratio (k_s/k_f), thermal resistances (R_t) of HP and foam porosities (ε_f). Assuming that the flow is laminar and incompressible, and the thermophysical properties of fluid/solid phases are temperature-independent, the above-mentioned problem can be described by the following governing equations:

-Continuity equation:

$$\nabla \cdot \boldsymbol{v} = 0 \tag{1}$$

-Momentum equation:

$$\rho \nabla \cdot (\boldsymbol{v}\boldsymbol{v}) = -\nabla p + \mu \nabla^2 \boldsymbol{v} \tag{2}$$

-Energy equation for fluid:

$$\rho c_p \nabla \cdot (\boldsymbol{v} T_f) = k_f \nabla^2 T_f \tag{3}$$

-Energy equation for metal foam and HP:

$$k_s \nabla^2 T_s = 0 \tag{4}$$

where $\boldsymbol{v}$ is fluid velocity; ρ is fluid density; p is fluid pressure; μ is dynamic viscosity; c_p is the specific heat at constant pressure of fluid; T_f is fluid temperature; T_s is temperature of solid; k_f and k_s are thermal conductivities of fluid and solid phases. At the inlet, the velocity-inlet boundary condition with parametrically changed uniform velocity (v_{in} = 10^{-9} - 10^{-3} m/s) and constant uniform temperature (305 K) is prescribed. The corresponding pore Re (= $\rho v_{in} d/\mu$, where d is equivalent diameter of foam ligament)

is 10^{-7} - 10^{-1}.The outflow boundary condition is prescribed at the fluid outlet. Constant heat flux (q = 10000 W/m^2) is prescribed at the bottom surface of HP while the surrounding surfaces of HP are thermally insulated. The symmetry boundary conditions are set for the remaining outer surfaces to ensure zero flux of both heat and mass transfer. Paraffin is selected as the PCM. Steel alloy, nickel, aluminium and copper are adopted as the foam and HP materials. Thermophysical properties of associated materials can be found in [11]. The foam-to-fluid thermal conductivity ratio (k_s/k_f) is from 82 to 2005, besides the thermal resistance (R_t) of HP varies from 3.12 × 10^{-2} to 1.27 × 10^{-4} m^2·K/W. Moreover, different foam porosities of 0.917, 0.949 and 0.974 are considered.

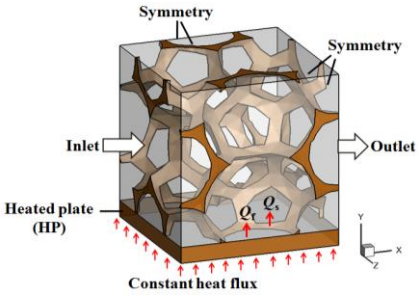

Figure 1. Physical problem at pore scale.

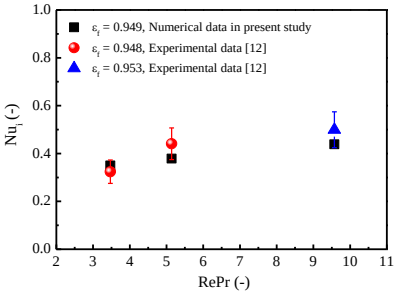

Figure 2. Comparisons of Nu$_i$.

The computational domain of the aforementioned physical problem is first spatially meshed using 3D unstructured grids. Afterwards, the governing equations are discretized with finite volume method. The computational residuals for momentum and energy are less than 10^{-5} and 10^{-9}, respectively. The conjugated fluid flow and heat transfer with specified boundary conditions are solved through a commercial finite-volume CFD solver: ANSYS FLUENT 6.3. Grid independency is tested to find the appropriate mesh size. To validate the present pore-scale computation framework, the interstitial Nusselt number (Nu$_i$) for convective flow of air in nickel foam with porosity around 0.95 is calculated and compared with the experimental results in [12] (see Figure 2).The comparison shows that the present numerical simulation predicts the Nu$_i$ with deviation less than 15 %, i.e., our pore-scale modelling of heat transfer in foam is relatively accurate.

3. RESULTS

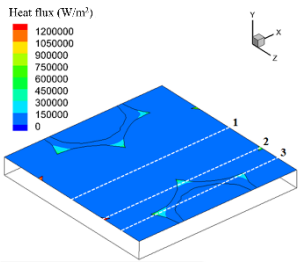

Figure 3. Heat flux distributions at top surface of HP.

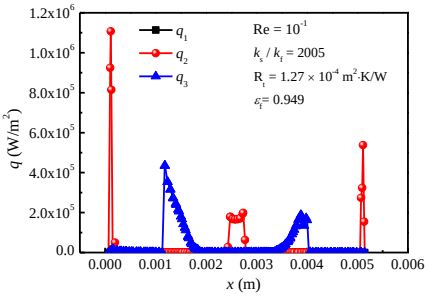

Figure 4. Heat flux along line 1, line 2 and line 3 (depicted in Fig. 3).

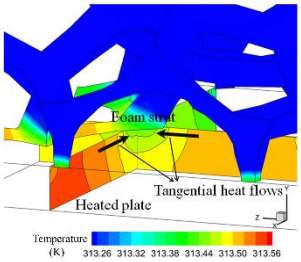

Figure 5. Tangential heat flows relative to the x-z plane in HP.

Figure 3 shows the heat flux distribution on top surface of the heated plate (HP) when Re = 10^{-1}, k_s/k_f = 2005, R_t = 1.27 × 10^{-4} m^2·K/W and ε_f = 0.949. As it can be seen, though a uniform heat flux is prescribed at the bottom surface of HP, the thermal flux on the top surface of HP bound to metal foam and PCM is not even. This non-uniformity can further be revealed in Fig. 4, showing the fluxes along line 1, line 2 and line 3 as depicted in Fig. 3. It is obvious that the heat flux at the metal foam-HP interface is much higher than that at fluid-HP interface, as the thermal resistance of metal foam is much lower than that of liquid paraffin (with low thermal conductivity). Fig. 5 shows the temperature fields at two cutting planes of HP, which intersect at a position under the metal foam-HP interface. The

temperature contours indicate that there are tangential heat flows (relative to the *x*-*z* plane) in HP transferred from the region under fluid-HP interface to the region under foam-HP interface (as shown by the thick arrows). This finding can give more intuitional explanation that why the heat flux at the foam-HP interface is much higher than that at the fluid-HP interface.

The heat flux bifurcation between metal foam and fluid can be evaluated by Q_s/Q_f, where Q_s and Q_f are heat flows through the metal foam-HP interface and fluid-HP interface, respectively. Fig. 6 shows the variations of Q_s/Q_f with Re for different k_s/k_f at foam porosity of 0.949. It is found that for a given k_s/k_f, Q_s/Q_f is almost unvaried at a lower Re ($< 10^{-2}$) because the heat transfer is dominated by thermal diffusion. When Re is higher than 10^{-2}, there is a significant increase of Q_s/Q_f with Re for higher k_s/k_f. It is due to the fact that the convection gets stronger at higher Re especially for higher k_s/k_f case, which enhances the interstitial heat transfer between metal foam and liquid paraffin, thus more heat is distributed to foam skeleton at the heated boundary. To more clearly understand the effect of Re on heat transfer, the temperature fields on z = 2.54 mm plane for diffusion-dominated heat transfer at a lower Re and for convection-dominated heat transfer at a higher Re are presented in Fig. 7. It is shown that, only for the higher Re case, evident thermal wakes can be found at the downstream of foam skeleton, indicating the convection dominated thermal transport. For relatively low k_s/k_f case in Fig. 6, the variation of Q_s/Q_f with Re is insignificant even at relatively higher Re region, because foam skeleton with low conductivity limits the interstitial heat transfer between metal foam and liquid paraffin. However, with the increase of k_s/k_f, Q_s/Q_f can be enhanced more than five times, benefiting from the foam skeleton of higher thermal conductivity which absorbs more heat from the heated boundary through conduction.

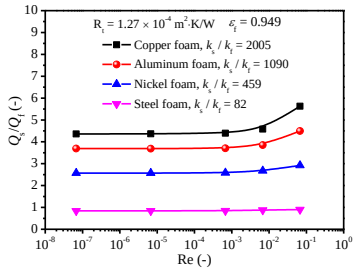

Figure 6. Variations of Q_s/Q_f with pore Re for different k_s/k_f.

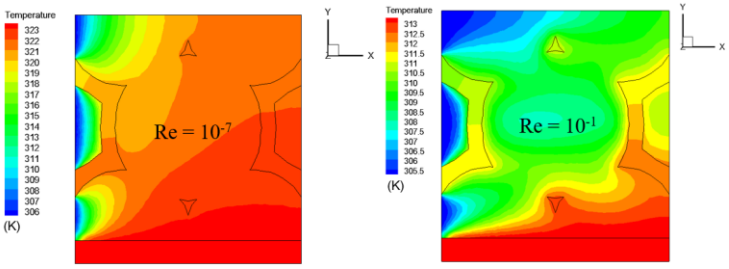

Figure 7. Diffusion dominated temperature field and convection dominated temperature field on z = 2.54 mm plane.

Figure 8 shows the variations of Q_s/Q_f with thermal resistance of HP (R_t) for copper foam and steel foam with porosity of 0.949 at Re = 10^{-1} and 10^{-7}, respectively. A wide range of R_t is considered corresponding to different HP materials with distinct thermal conductivities. It is found that for copper foam with relatively high k_s/k_f, Q_s/Q_f decreases with the increase of R_t at both higher Re and lower Re, because higher R_t impedes heat distribution to foam skeleton from the heated boundary. Thus, reducing R_t by use of high conductive HPs such as copper and aluminium plates is required to enable a higher Q_s/Q_f. For steel alloy foam, as it has relatively low k_s/k_f which limits the thermal transport in foam skeleton, there is no evident influence of R_t on Q_s/Q_f at both higher Re and lower Re. Fig. 9 shows the variations of Q_s/Q_f with Re for copper foams with different porosities. For a given porosity, the variation trend of Q_s/Q_f with Re is the same as that discussed in Fig. 6. For a given Re, it is found that Q_s/Q_f significantly increases with the decrease of foam porosity. The reason is that metal foam with lower porosity has thicker foam skeleton (i.e., higher effective thermal conductivity and larger heat transfer area), which enhances the heat transfer from metal foam to paraffin, and meanwhile more heat can be drawn to metal foam from the heated boundary. Fig. 10 shows a comparison of Q_s/Q_f between results predicted by the model 1F (i.e., $Q_s/Q_f = k_s(1-\varepsilon_f)/(k_f\varepsilon_f)$) of Alazmi and Vafai [6] and results from present pore-scale study. Model 1F of Alazmi and Vafai is selected because it is a combination of models 1D and 1E in [6], which are all extensions of the first method for heat flux bifurcation at the heated wall of porous medium mentioned in the introduction. As can be seen from Fig. 10, though both results show the same overall trend of variation in Q_s/Q_f, the predictions by the model 1F of Alazmi and Vafai are

much higher than that of present study. The noticeable difference is mainly because the model 1F of Alazmi and Vafai does not consider multiple influences including complex structure of metal foam, interstitial convective heat transfer between metal foam and fluid, and the thermal resistance of boundary wall with certain thickness. This indicates that, pore-scale simulation is indispensable to extract the accurate heat flux bifurcation information at boundary wall, required for local thermal non-equilibrium (LTNE) volume-averaging modelling of metal foam/fluid system.

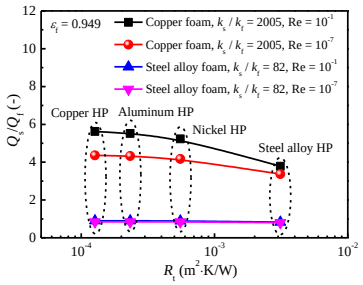

Figure 8. Variations of Q_s/Q_f with R_t.

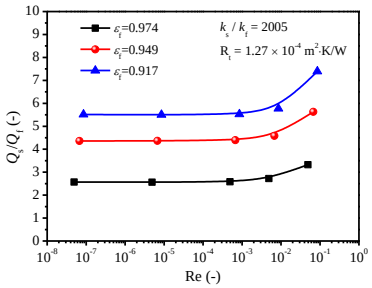

Figure 9. Variations of Q_s/Q_f with Re for different ε_f.

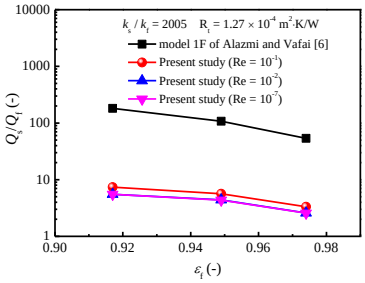

Figure 10. Comparisons of Q_s/Q_f between model [6] and present study.

4. CONCLUSIONS

In this work, we employ pore-scale direct simulation to investigate the heat flux bifurcation between metal foam and fluid at the heated boundary for low Reynolds number convection of liquid paraffin in open-cell metal foam. The Weaire-Phelan cellular topology is adopted to construct the porous structure of metal foam. Through pore-scale modelling and analysis, the influences of Reynolds number (Re), foam to fluid thermal conductivity ratio (k_s/k_f), thermal resistance (R_t) of heated plate (HP) and foam porosity (ε_f) on heat flux bifurcation are revealed. The present pore-scale results are also compared to a literature reported intuitive model. The main conclusions are as follows:

(1) There exist remarkable tangential heat flows within the heated plate, which transfer from the region close to liquid paraffin to the region near foam skeleton. As a result, the heat flux at the heated boundary near metal foam region is much higher than that close to liquid paraffin region.

(2) For steel alloy foam with relatively lower k_s/k_f and at lower Re, the influences of Re and R_t on the heat flux ratio (Q_s/Q_f) between metal foam and fluid regions at the heated boundary are weak. For copper, aluminium and nickel foams having relatively higher thermal conductivities and at higher Re region, Q_s/Q_f increases with the increase of Re and the decrease of R_t. Moreover, Q_s/Q_f increases with the increase of k_s/k_f and the decrease of ε_f. The variation range of Q_s/Q_f in this study is from 0.8 to 7.4 depending on different conditions.

(3) The explicit model, ignoring multiple influences including complex structure of metal foam, interstitial convective heat transfer between metal foam and fluid, as well as the thermal resistance of boundary wall with certain thickness, is not accurate to predict the heat flux bifurcation. Pore-scale simulation is indispensable to extract the boundary heat flux bifurcation information required for local thermal non-equilibrium (LTNE) volume-averaging modelling of metal foam/fluid system.

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