NON-DARCY NATURAL CONVECTION IN A HIGH POROSITY MEDIUM HEATED FROM BELOW: EFFECTS OF THERMAL DISPERSION AND LOCAL THERMAL NON-EQUILIBRIUM - PART 2. NUMERICAL AND EXPERIMENTAL RESULTS

M.S. Phanikumar – phani@msu.edu
Departments of Geological Sciences and Civil & Environmental Engineering
Michigan State University – East Lansing, MI, USA

R.L. Mahajan – mahajan@spot.colorado.edu
CAMPMode (Center for Advanced Manufacturing and Packaging
of Microwave, Optical and Digital Electronics)
Department of Mechanical Engineering
University of Colorado, Boulder, CO, USA

Abstract. *In this two-part paper, we present numerical solutions for buoyancy induced flows in high porosity metal foams heated from below. Experiments conducted under natural convection conditions for the same configuration were used to validate the numerical model. The results show enhancement in heat transfer for different metal foam - fluid combinations. Thermal dispersion effects and the effects of Darcy number on heat transfer are reported. Conditions under which the local thermal equilibrium (LTE) assumption can introduce significant errors are also discussed.*

Keywords: *Metal Foams, Enhancement, Thermal Non-Equilibrium, Dispersion*

1. INTRODUCTION

The aim of this paper is to report numerical and experimental results for natural convection in a metallic foam sample heated from below. The numerical model and details of the problem formulation are presented in part 1 of this paper.

2. EXPERIMENTS

The metal foam heat sinks were tested under natural convection conditions in a horizontal configuration. A schematic of the experimental set-up is shown in Figure 1. It consists of a large Plexiglass housing isolated from the ambient, inside which the sample was stationed. Holes were drilled on the base of the samples to insert Firerod® cartridge heaters. The base of the sample was insulated using low conductivity Styrofoam insulation. The cartridge heaters were powered by a DC power supply. The base and ambient temperatures were monitored using 0.127 mm. T-type thermocouples connected to an Omega DASTC data acquisition system. Experiments were

conducted on foam samples of different porosities and pore densities. For each pore density corresponding to 5, 10, 20 and 40 PPI, two samples of different porosities were chosen. During a typical experimental run, the power to the heaters was varied to achieve different base plate temperatures and hence Raleigh numbers. Due to temperature properties of the Styrofoam insulation, our experiments were restricted to maximum base plate temperatures of 75 ^{o}C.
The heat transfer co-efficient for a typical run was calculated based on

$$h = \frac{Q}{A(T_{base} - T_{amb})} \quad (1)$$

where Q = heat input to the patch heaters

A = area of the heat sink base

T_{base} = average temperature of the heat sink base

T_{amb} = ambient temperature

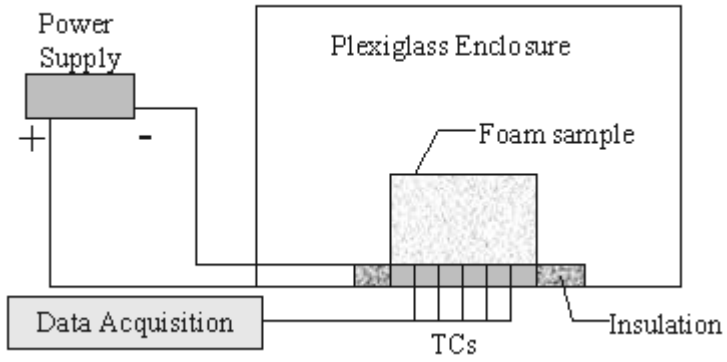


Figure 1. Schematic of the Experimental Setup

3. RESULTS AND DISCUSSION

We have studied three representative combinations of high porosity foam material and fluid in the present work: aluminum foam in water, aluminum foam in air, and Reticulated Vitreous carbon (RVC) foam in air. The properties of these samples used in the computations are listed in Table 1.

Figure 2 shows a comparison of the experimental data with the model predictions of Nusselt number for 5 PPI. An excellent agreement is obtained. Also shown in the figure is the result of a study aimed at understanding the effect of C_T on heat transfer. As C_T increases, the Nu increases as the efficiency of interfacial heat exchange improves. This trend continues till the solid and fluid phase temperatures satisfy local thermal equilibrium at which point the Nusselt number ceases to increase with a further increase in C_T. This occurs at a value of C_T = 0.51.

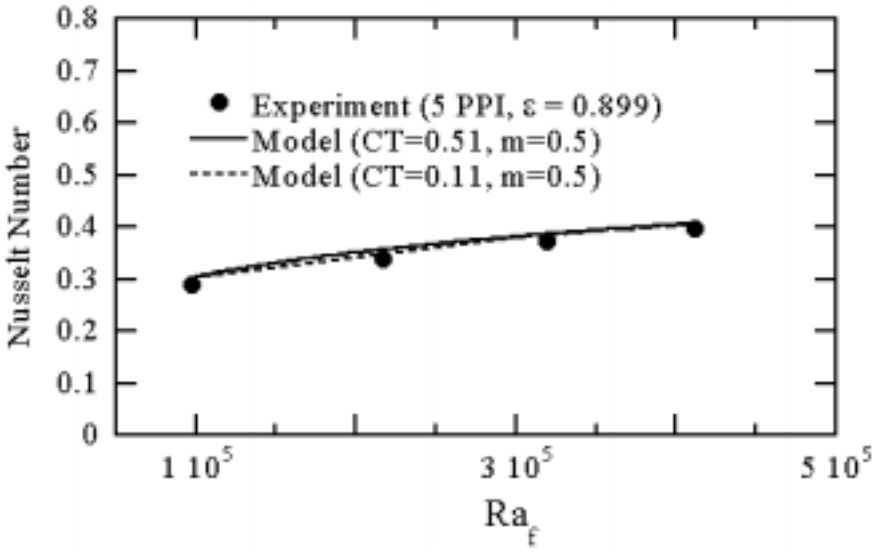


Figure 2. Comparison of the present experimental data for Nusselt number with model predictions for aluminum-air

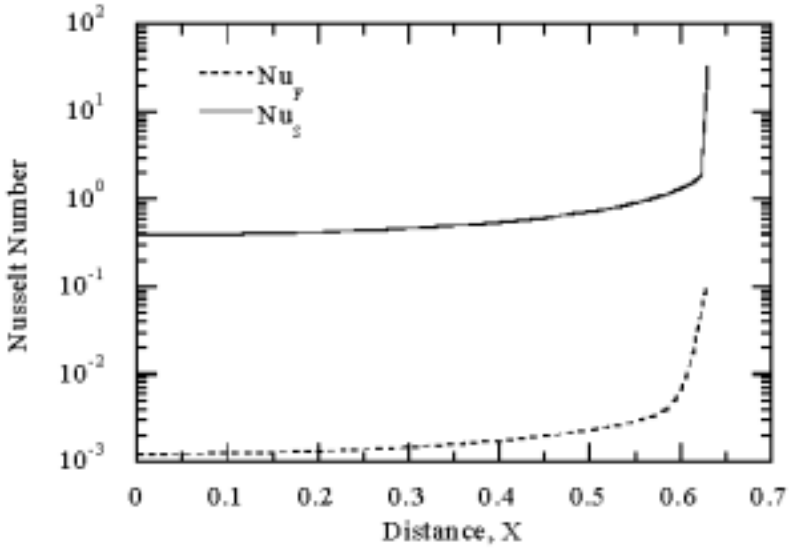


Figure 3. Solid and Fluid (local) Nusselt numbers for 5 PPI aluminum sample in air. Rayleigh number = 411537.9

This value of C_T is used for all the cases reported in this paper. Figure 3 shows the individual (solid and fluid) Nusselt numbers for the two phases. We notice that the fluid phase Nusselt number is orders of magnitude lower compared to its solid phase counterpart and contributes very little to the total Nusselt number. Even though the temperature gradients in the solid and the fluid are comparable, the high value of (k_{se}/k_{eff}) in the definition of Nu increases the contribution from the solid. For the conditions shown in figure 2, the two conductivity ratios are respectively 278 and 0.87 for the solid and the fluid.

We now present results for flow and heat transfer for the case of no dispersion (i.e. $C_D = 0$). Figure 4 shows the streamfunction and isotherm plots for a 5 PPI aluminum metal foam with air

as the saturating fluid. We notice from this figure that there is a strong entrainment from the air domain on the right hand side and into the metal foam near the bottom of the porous block. The entrained fluid reaches the heated surface and moves up due to thermal buoyancy. The isotherm plot confirms the upward movement of fluid in a plume shaped region around the symmetry line at the left. The heat transfer results for this case are shown earlier in figure 5 using the solid and fluid phase Nusselt numbers.

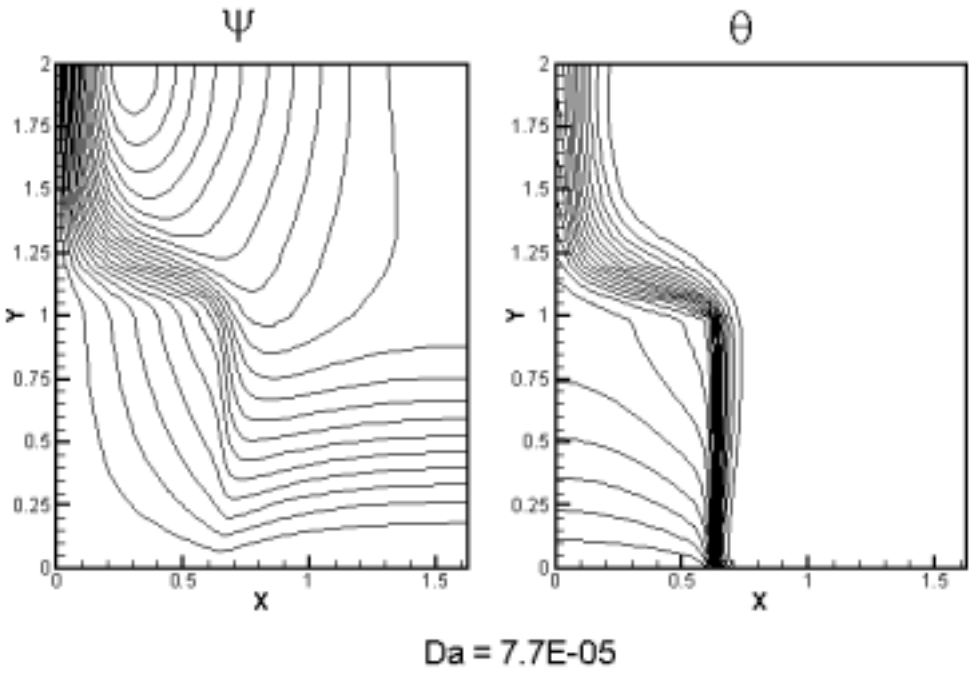


Figure 4. Streamfunction and Isotherm plots for Aluminum-air (5PPI), Ra = 411537.9

To estimate the heat transfer enhancement resulting from the use of the metal foam, we compared the Nusselt numbers to the case in which there is no metal foam. The Nusselt number in the absence of the metal foam is calculated using the following correlation proposed by Lloyd and Moran (1974) for a heated surface facing upward:

$$Nu_{L^*} = 0.54 Ra_{L^*}^{0.25}$$

where $L^* = A/p$ is a characteristic length of the plane surface. Figure 5 shows the enhancement as a function of Rayleigh number for aluminum-air. Detailed comparisons of enhancement for all the metal foam – fluid combinations will be presented in a future paper. We note that for a meaningful comparison the Nusselt numbers for both cases (with and without metal foam) should be based on the thermal conductivity of the fluid (k_f) rather than the effective conductivity see equations 16 in part 1. For the aluminum-air combination shown in figure 5, the heat transfer enhancement due to the presence of the metal foam is significant –about 400 percent. For the case of carbon-air, and aluminum-water, the enhancements were found to be much smaller compared to the case of aluminum-air (not shown). This wide range of values for the enhancement can be understood by examining the solid to fluid thermal conductivty ratios (λ_s) for these cases (Table 2). The high value of

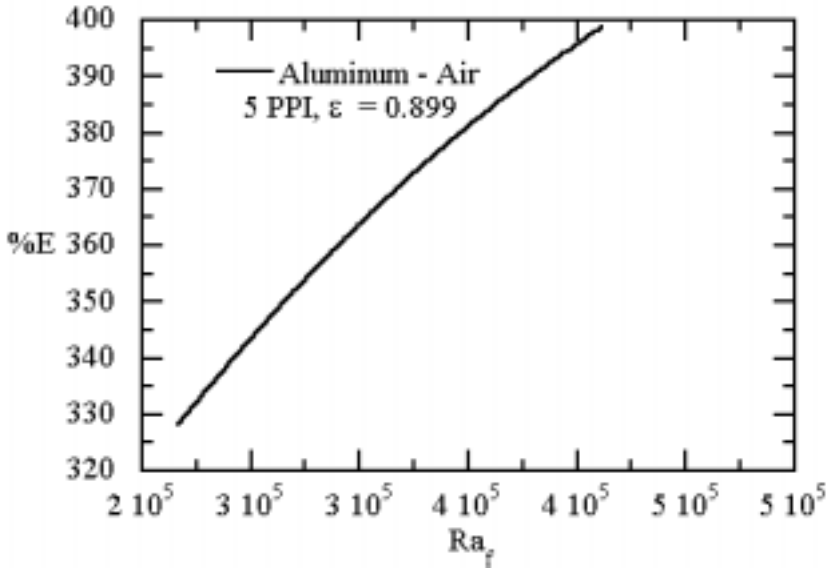
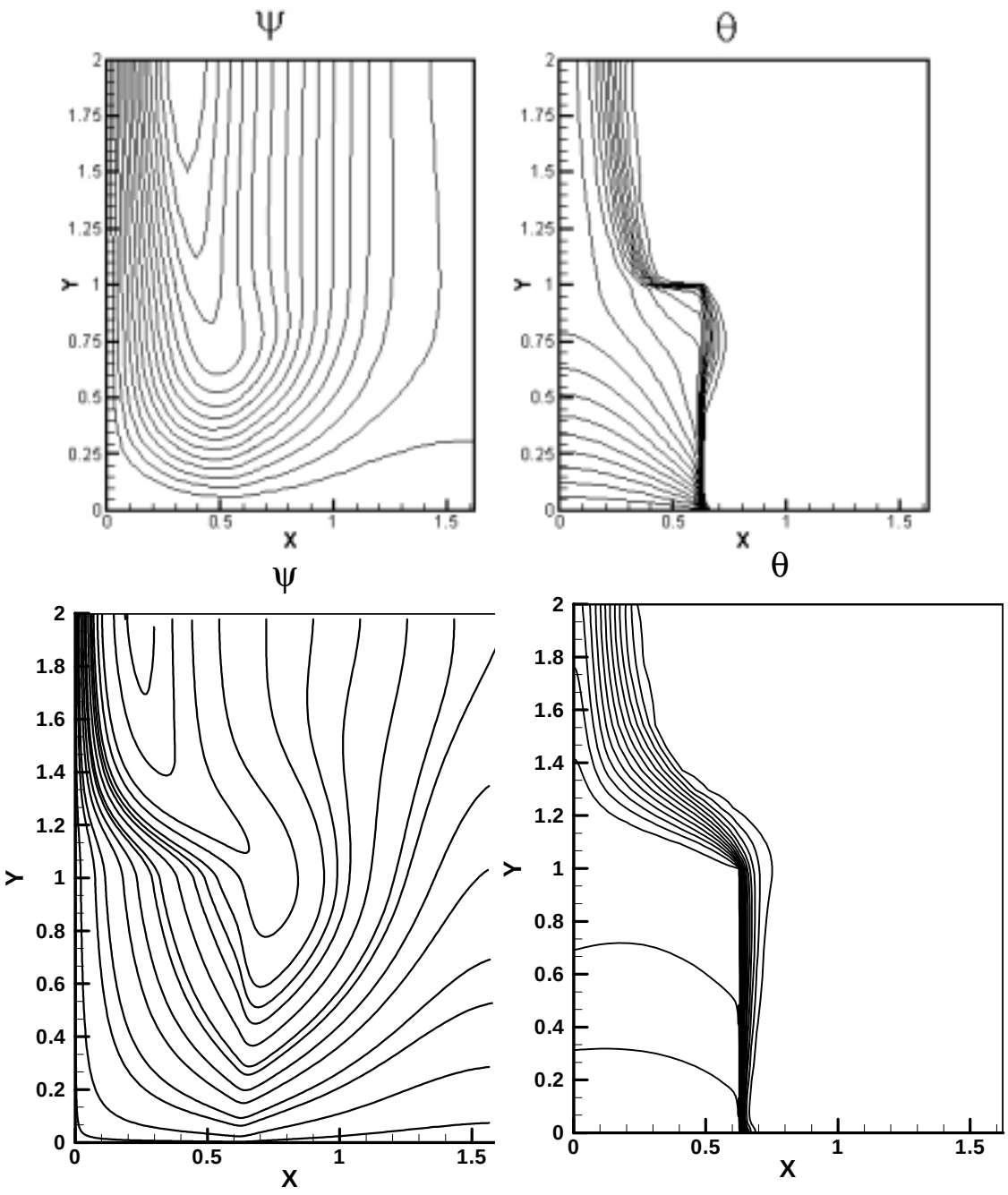


Figure 5. Enhancement in Heat Transfer as a function of Rayleigh number for aluminum – air.

enhancement obtained for the case of aluminum-air is due to the high value of the solid phase conductivity compared to that of the fluid (λ_s = 278.2). For aluminum-water, and carbon-air the values of λ_s are respectively, 11.94 and 10.80. The dominant mode of heat transfer for these cases is conduction in the metal foam matrix as already noted earlier in Figure 3. This explains why such relatively high values of enhancement are obtained for the case of Al-air.

3.1 Effect of Darcy Number

To study the effect of permeability of the metal foam on the flow and heat transfer we study the effect of Darcy number. Figures 6(a)and 6(b) show the streamfunction and isotherm plots for a high value of Darcy number (Da = 1) and for a very low value (Da = 10^{-8}) respectively. It can be seen from the streamfunction as well as the isotherm plots that increasing the Darcy number (permeability) makes the flow penetrate deeper into the porous layer. This is consistent with the well known fact that for high values of Da, the Darcy term becomes very small and the flow in the porous layer becomes essentially similar to a flow without the porous medium and one that is governed by Navier-Stokes equations with the inertia terms retained. For very low values of Da, the Brinkman term in the momentum equation becomes small in comparison with the other terms and may be dropped. For this case, there is a considerable resistance to the flow in crossing the interface and entering the porous layer. As a result, the flow will not be able to penetrate the porous layer for all practical purposes and the entire flow is essentially confined to the air domain. In this case, if the Rayleigh number is sufficently high, a strong boundary layer develops adjacent to the porous layer. This situation is qualitatively similar to the case of flow over an impermeable solid block. Indeed, as can be seen from figure 6(b) there is a strong boundary layer type of flow near the porous block. The horizontal velocity is found to be highest near the bottom (y=0) of the metal foam where air gets entrained from the surrounding region. This fluid comes in contact with the heated surface, moves upward and eventually leaves the porous block at the top on the left side in a plume region.



Figures 6(a) (Top) and 6(b) (Bottom) Streamfunction and Isotherm plots for (a) Da = 1.0 and (b) Da = 1.0E-08

Figure 7 shows the effect of Darcy number on the heat transfer. The Nusselt number increases with the Darcy number and reaches a constant value around a Darcy number of 1.0. Similarly the Nu approaches a constant value for very low values of Da.

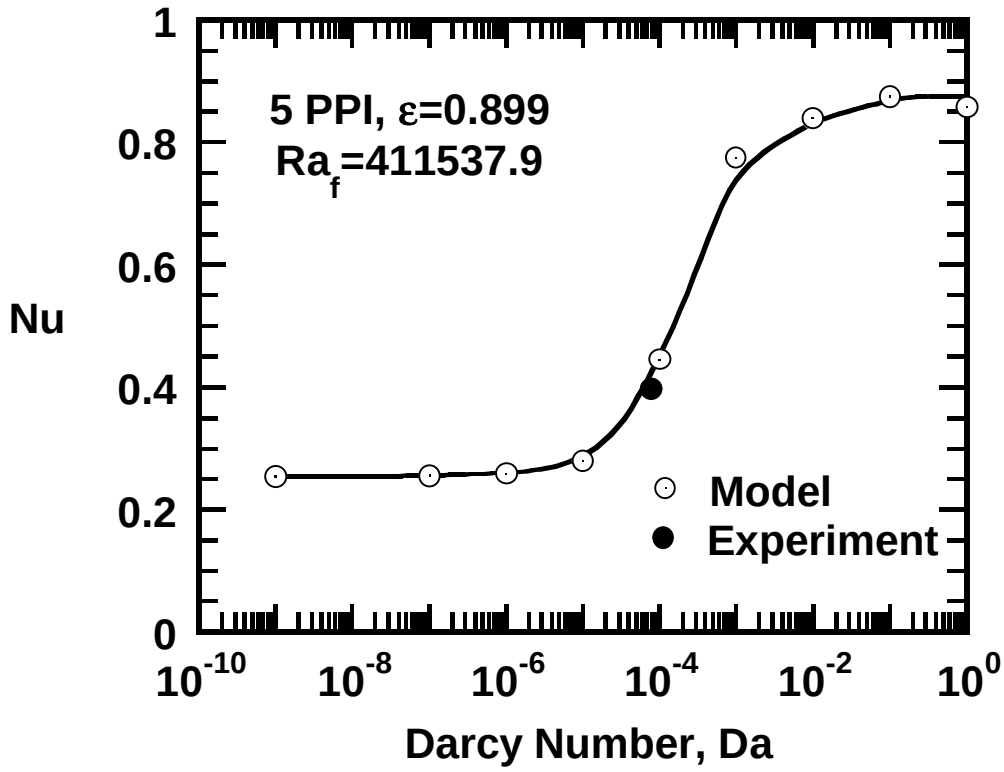


Figure 7. Effect of Darcy number on heat transfer for aluminum – air, $C_D = 0$

3.2 Effects of Thermal Dispersion

We now study the effects of thermal dispersion on flow and heat transfer. For the aluminum – air combination, the enhancement in heat transfer due to thermal disperion is studied for C_D values ranging from 0.1 to 0.3 and found to be negligible for the experimental conditions summarized in Table 2. The value of C_D = 0.1 was obtained based on order of magnitude estimates using the experimental work of Hunt and Tien(1988), the theoretical work of Koch and Brady (1986) and the numerical work of Jiang et al (1999) – all for forced convection. For thermal dispersion to become significant, the contribution due to dispersion should be comparable to the effective stagnant conductivity, k_{eff}. Equation (15) in part 1 of this paper can be rewritten in a dimensionless form as:

$$\frac{k_d}{k_{eff}} = \frac{C_D\sqrt{Da}}{\lambda_{eff}}\left|\bar{\mathbf{v}}\right| \qquad (2)$$

For the aluminum metal foam sample in air, typical values of Da and $\left|\bar{\mathbf{v}}\right|$ (for a Rayleigh number of 411537.93) are respectively 7.7×10^{-5} and 300.0 respectively. For C_D values of 0.1 and 0.3 , and $\lambda_{eff} = 273.2$, the dispersion accounts for 0.09% and 0.29% of the total stagnant conductivity. Consequently, we expect the effects of thermal dispersion to be negligible for this

case. This was indeed borne out by our numerical solutions and hence these results are not shown here.

Table 1
Properties of Typical Foam Samples Used in the present work

Material	ε	d_p (m)	K x 10^7 (m^2)	C	k_{se} W/m.K	k_{fe} W/m.K	k_{eff} W/m.K
Aluminum -air	0.899	0.00422	1.989	8.75E-02	7.28	0.0229	7.318
carbon-air	0.899	0.00422	1.989	8.75E-02	0.104	0.0251	0.1336
Aluminum -Water	0.899	0.00422	1.989	8.75E-02	7.29	0.538	8.13

Table 2
Effects of Thermal Dispersion on Heat Transfer

#	Material	Ra	C_D	Da	Nu	% Increase
1	Al - Water	411537.9	0.0	0.01	1.590	--
2	Al - Water	411537.9	0.1	0.01	1.604	0.89
3	Al - Water	411537.9	0.2	0.01	1.624	2.14
3	Al - Water	411537.9	0.3	0.01	1.642	3.27
4	Al - air	411537.9	0.0	0.01	0.800	--
5	Al - air	411537.9	0.1	0.01	0.871	8.92
6	Al - air	411537.9	0.2	0.01	0.876	9.50
7	Al - air	411537.9	0.3	0.01	0.880	10.0

However, as shown in Table 2, if we increase the Darcy number (for example, by decreasing H), then the effects of dispersion become important. For example, for a Darcy number of 0.01, thermal dispersion can account for a 10% increase in heat transfer for C_D = 0.3. Still higher increases in heat transfer have been obtained for higher Ra. For the aluminum-Water combination, we increased the Darcy number to 10^{-2} while keeping all other properties constant. For this case, dispersion accounts for less than 5% of the total heat transfer. Since increasing the Darcy number improves flow penetration and results in higher flow velocities, $|\bar{\mathbf{v}}|$ increases therby increasing the contribution from dispersion. However, the net increase is governed by the ratio of the stagnant to the dispersion conductivity. The stagnant conductivity is lower for the aluminum – air combination compared to the aluminum – water combination. Even though the velocities are higher for the Al-water case, the relative increase is higher for the case of Al-air. The effects of dispersion were found to be highest for the aluminum metal foam due to its high permeability compared to other cases (e.g. carbon).

3.3 Effects of Local Thermal Nonequilibrium

For the cases presented above, an examination of the effects of local thermal non-equilibrium was carried out by comparing the fluid and solid temperatures at every point in the porous medium. The departure of the solid temperature from its fluid counterpart is taken as a measure of the local thermal non-equilibrium as defined below (Amiri and Vafai, 1994):

$$\%\text{LTNE} = \left|\theta_{\text{f}} - \theta_{s}\right| \times 100$$

Figure 8(a) shows the variation of %LTNE as a function of space for aluminum-air. Figure 8(a) shows the %LTNE for a low Rayleigh number while 8(b) shows the same for a high Ra. First for the low Ra, we notice that the levels of LTNE are much lower compared to those for a higher Ra. Sincethe local thermal equilibrium is assumed for all the right, top, and bottom surfaces, we expect the %LTNE to be small around these regions. This was found to be true (Figure 8(a)). Secondly, we notice that as the Rayleigh number increases the region of high %LTNE values is associated with high-velocity regions. Figure 8(b) shows this effect at a Rayleigh number of 10^6 and Da = 10^{-2}. We notice that, for this case, the effects of LTNE are beginning to become significant and that they are mostly confined to two high-velocity regions: (1) a region near the lower portion of the metal foam where there is a relatively strong entrainment of fluid from the air domain and (2) a region near the plume on the left hand side.

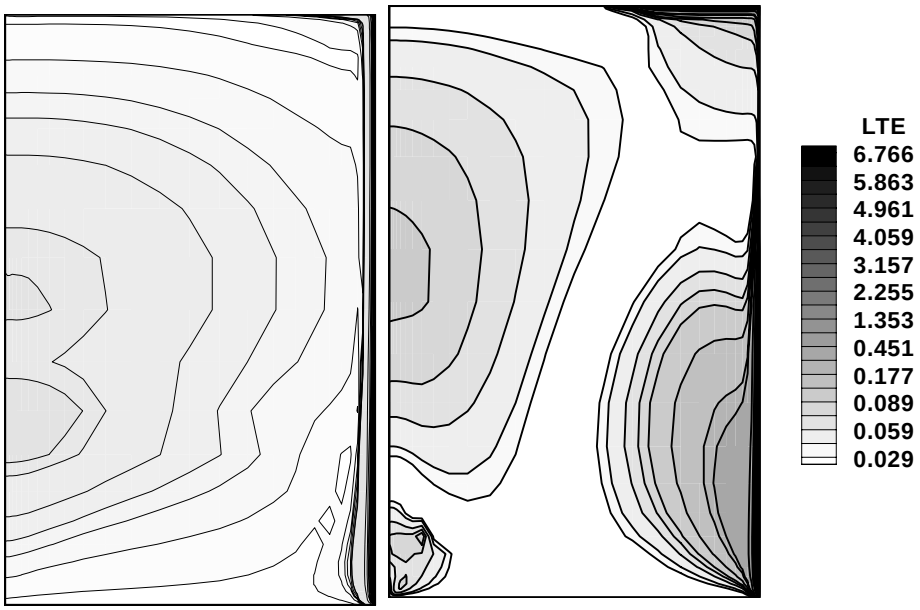


Figure 8(a) (Left) and (b) (Right). Local Thermal Non-Equilibrium (%LTNE) as a function of space for aluminum – air. (a) Ra = 411537.9, Maximum %LNTE = 2.5 (b) Ra = 1.0E+06, Maximum %LTNE = 6.7

This can be explained by examining the matching conditions for the interfacial heat flux, equation (18) of part1 of this paper. We note that the LTE assumption is more appropriate at a no-slip wall where the convective flux vanishes resulting in a local thermal equilibrium condition. In the present problem, however, the velocity at the porous-fluid interface is not zero. Indeed, a local maximum in the horizontal velocity (u) occurs near the lower portion of the metal foam (near y = 0). Hence a deviation from LTE can be expected in this region. Similarly, hot fluid rises along the symmetry line at the left giving rise to high velocities at the top interface near x = 0. The LTE assumption is therefore not justified there as shown in figure 8(b). We found that the validity of the LTE assumption rapidly decreases as the Darcy number is increased for a given Rayleigh number. Similarly , it was also found that if the Rayleigh number is increased keeping the Darcy number constant, the LTE assumption likewise becomes less justifiable.

4. CONCLUSIONS

In the present work we examined the flow and heat transfer that results when a metal foam sample is heated from below. Significant enhancements in heat transfer were obtained. In some cases, the enhancement is more than a factor of four compared to the case in which there is no metal foam. Even though dispersion did not play a role for the experimental conditions described in this paper, we showed that significant errors (e.g., 10%) can be made if dispersion is neglected at high Darcy numbers. Local thermal non-equilibrium effects were found to be significant at high Rayleigh and Darcy numbers and significant errors can be potentially introduced if a single-equation approach is adopted. Consequently, the two-equation model for energy is a better model for metal foams.

5. ACKNOWLEDGEMENTS

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