1 Introduction

Buoyancy-induced convection plays an important role in the cooling of electronic components. Despite significantly lower heat transfer coefficients compared to forced convection or immersion boiling, buoyancy-induced convection in air is preferred for low-end applications, such as switching devices, consumer electronics, and avionics packages, due to their simplicity and reliability. Further, any increase in heat transfer rate that might be achieved through innovative designs comes at no additional penalty (unlike that of a pressure drop that is experienced in forced convection). In this paper, we evaluate the thermal performance of metal foam and finned metal foam heat sinks in buoyancy-induced convection. The focus in the first part is on metal foam heat sinks, while the second half deals with natural convection in finned metal foam heat sinks.

2 Buoyancy-induced Convection in Porous Media

Fundamental investigations on natural convection in saturated porous media started with the linearized stability theory applied to an infinite horizontal layer heated from below [1,2]. These studies developed the criterion for the onset to convection, which was later experimentally verified by other researchers [3]. Many other experimental and analytical studies have also been reported in the literature which deal with buoyancy-induced convection through saturated porous media in various geometries, including rectangular cavities heated from bottom or sides, and horizontal and vertical annuli [4]. Extensive reviews of these studies can be found in [4–6].

Most of the earlier theoretical and numerical studies on buoyancy-induced convection in a porous medium are based on empirical models using the classical Darcy formulation. The experimental studies are relatively few in number and mostly deal with a packed bed of spherical beads. They show significant discrepancies with the analytical or numerical results based on the Darcy model. The numerical models are found to consistently underpredict the experimental results.

Non-Darcian effects in such flows have been investigated by a number of researchers including Beckermann et al. [7], Lauriat and Prasad [8], Poulikakos and Bejan [9], Prasad and Tuntomo [10], Tong and Subramaniam [11], and Beji and Gobin [12]. It is reported that the non-Darcy terms contribute toward decreasing the superficial velocities and heat transfer. Recalling that the Darcy formulation results in an underprediction of the heat transfer rate, incorporation of the non-Darcy terms leads to even larger differences. Thus, other physical phenomena need to be included at the macroscopic level to account for these discrepancies. David et al. [13] and Cheng et al. [14] suggested that the variation of porosity at the walls of the enclosures is likely to induce higher velocities at the boundary layers [15]. Their numerical results ex-
hbit a significant increase in heat transfer when this porosity variation is taken into account. Another factor that is expected to enhance heat transfer is thermal dispersion, models of which have been proposed by Hong and Tien [16], Georgiadis and Catton [17], and Hsu and Cheng [18]. Beji and Gobin [15] numerically solved the Darcy-Brinkman-Forchheimer equations (incorporating Darcy, viscous, and inertial effects) including the effect of thermal dispersion. Their results show a considerable increase in the overall heat transfer and better agreement with the experimental data of Inaba et al. [19]. However, they note that such an improvement is limited to porous media for which the solid and fluid phase thermal conductivities are comparable.

Similar to forced convection, modeling of natural convection in porous media has been traditionally studied with the local thermal equilibrium (LTE) assumption. Consequently, a single energy equation describes the temperature field in the porous medium. While this assumption holds well when the thermal conductivities of the constituent phases of the porous medium are comparable, its validity cannot be taken for granted, so a priori, for a medium, whose constituent phases are in thermal equilibrium. While this assumption holds well when the thermal conductivities of the constituent phases of the porous medium are comparable, its validity cannot be taken for granted, a priori, for a medium, whose constituent phases are in thermal equilibrium. However, they note that such an improvement is limited to porous media for which the solid and fluid phase thermal conductivities are comparable.

3 Buoyancy-Induced Convection in Metal Foams—Experiments

Experimental studies were conducted on aluminum metal foam samples to evaluate their thermal performance in buoyancy-induced convection. A schematic of the experimental setup is shown in Fig. 1. The sample is stationed inside a large Plexiglass housing (60 cm × 60 cm × 40 cm) isolated from the ambient. Holes were drilled on the base of the samples to insert Fierro® cartridge heaters (see Fig. 2). 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 measured using 0.127 mm (36 AWG) 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 (see Table 1). During a typical experimental run, the power to the heaters was varied to achieve different base plate temperatures and, hence, Rayleigh numbers. Limited by the allowable operating temperature of the styrofoam insulation, our experiments were restricted to maximum base plate temperatures of 75°C. The heat transfer coefficient for a typical run was calculated as follows:

$$h = \frac{Q}{A_h(T_b - T_{amb})}$$

4 Error Analysis

The main uncertainties in this experiment are due to errors in measurements of power, thermocouple readings, and physical dimensions. The maximum error in the multimeter readings for the voltage and resistance measurements is 0.5% (manufacturer’s data sheet). This results in an error of 1.5% in the estimation of the heat input Q. The combined error (resolution of the data acquisition unit, thermocouple calibration) in the estimation of the temperature difference under the given conditions is 0.2°C. For the temperature differences observed in our experiments (of the order of 30°C), the error in our measurement is thus 0.67%. The error in measurement of the cross-sectional area is estimated to be 0.75%. The heat losses through the styrofoam insulation (thermal conductivity is 0.03 W/m K), as well as the conduction losses through the thin thermocouple wires (diameter of 0.127 mm), are neglected. Based on the preceding errors, the total error in the measurement of the effective thermal conductivity can be calculated using [26]

$$\frac{\Delta h}{h} = \sqrt{\left(\frac{\Delta Q}{Q}\right)^2 + \left(\frac{\Delta A_h}{A_h}\right)^2 + \left(\frac{\Delta(T)}{T}\right)^2}$$

and was found to be ±1.8%. However, it may be noted that the above analysis does not take into account the error due to thermal contact resistance introduced by the epoxy used to bond the metal foam to the fins and the base.

Table 1 Metal foam samples used in the current study

<table>
<thead>
<tr>
<th>Sample No.</th>
<th>PPI</th>
<th>Porosity</th>
<th>Permeability (×10^{-2})</th>
<th>Inertial coefficient</th>
<th>Effective conductivity (W/m K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5</td>
<td>0.899</td>
<td>2.28</td>
<td>0.075</td>
<td>7.32</td>
</tr>
<tr>
<td>2</td>
<td>5</td>
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<td>2.40</td>
<td>0.084</td>
<td>5.33</td>
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<tr>
<td>3</td>
<td>10</td>
<td>0.9085</td>
<td>1.62</td>
<td>0.078</td>
<td>6.71</td>
</tr>
<tr>
<td>4</td>
<td>10</td>
<td>0.9386</td>
<td>1.54</td>
<td>0.085</td>
<td>4.78</td>
</tr>
<tr>
<td>5</td>
<td>20</td>
<td>0.92</td>
<td>1.11</td>
<td>0.081</td>
<td>5.97</td>
</tr>
<tr>
<td>6</td>
<td>20</td>
<td>0.9353</td>
<td>1.14</td>
<td>0.085</td>
<td>4.99</td>
</tr>
<tr>
<td>7</td>
<td>40</td>
<td>0.9091</td>
<td>0.51</td>
<td>0.078</td>
<td>6.67</td>
</tr>
<tr>
<td>8</td>
<td>40</td>
<td>0.9586</td>
<td>0.54</td>
<td>0.086</td>
<td>3.48</td>
</tr>
</tbody>
</table>

*Could not be tested in the vertical orientation.
5 Results

5.1 Vertical Orientation. The first set of experiments was conducted with the metal foam samples held in the vertical orientation and heated from the side. The results are shown in Fig. 3 in the form of the nondimensional parameters, defined as follows:

\[
\frac{\text{Nu}}{\text{Ke}} = \frac{hL}{k_f}, \quad \text{Ra}_L = \frac{g\beta(T_b - T_{\text{amb}})L^3}{\alpha_f \nu_f}, (3)
\]

where \( L \) is the length of the sample in the vertical direction, which is equal to 6.25 cm for our samples. The following conclusions can be drawn from the plots. For a given porosity, the resistance to flow decreases with an increase in pore size due to an increase in permeability. The result is enhanced flow and heat transfer rate. For a given PPI (pore density), the performance is enhanced with a decrease in porosity or an increase in metal content. This indicates that conduction is still the dominant mode of heat transfer. This result is consistent with our previous results on forced convection in metal foams [23].

The parameters on which \( \text{Nu}_L \) is expected to depend are the Rayleigh number, \( \text{Ra}_L \), and Darcy number (Da). We propose a correlation of the form

\[
\text{Nu}_L = C \text{Ra}_L^{0.25} \text{Da}^{0.8} \quad (4)
\]

where \( \text{Da} = K/L^2 \). The choice of the index for \( \text{Ra} \) was based on the existing correlations for buoyancy-induced convection on a vertical flat plate. Note that the porosity dependence is captured in Eq. (4) through the Darcy number, which is a function of the permeability \( K \) and, hence, porosity of the medium. Using a least-squares fit, the best correlation obtained was

\[
\text{Nu}_L = 38.94 \text{Ra}_L^{0.25} \text{Da}^{0.23} \quad (5)
\]

with a maximum error of ±8%. The correlation predictions are plotted along with experimental data in Fig. 3.

The correlation given by Eq. (5), however, suffers from a drawback. It is applicable only for aluminum foams since the Rayleigh and Darcy numbers are defined in terms of fluid properties. It has a practical merit in terms of comparison with existing correlations for other heat sinks, which are generally defined in terms of fluid properties. However, for a more general result, it is appropriate to formulate a correlation in terms of nondimensional parameters that are based on effective properties of the medium.

A survey of the literature on natural convection in porous media reveals that the most commonly used and appropriate definitions of Nusselt and Rayleigh numbers are as follows:

\[
\text{Nu} = \frac{hL}{k_e}, \quad \text{Ra}_K = \frac{g\beta(T_b - T_{\text{amb}})KL}{\alpha_f \nu_f} \quad (6)
\]

where \( k_e \) is the effective thermal conductivity of the porous sample, \( K \) is its permeability, and \( \alpha_f \) is its effective thermal diffusivity. For our samples, the values of \( k_e \) and \( K \) (listed in Table 1) were calculated using the models proposed by Boomsma and Poulikakos [27] and Bhattacharya et al. [28], respectively. For the effective thermal diffusivity, the following relation was used:

\[
\alpha_f = \frac{k_e}{\rho_e C_p e} = \frac{k_e}{[\epsilon \rho_f + (1 - \epsilon) \rho_e] C_{pf} + (1 - \epsilon) C_{pe}} \quad (7)
\]

Evans and Plumb [29] had shown that, for regular Darcy flow over a vertical flat surface covered with porous media, the Nusselt number varies as the square root of the Rayleigh number. A theoretical derivation for the same can also be found in Nield and Bejan [6]. Hsu and Cheng [30] and Kim and Vafai [31] considered the effects of boundary friction and used the Brinkmann model to conclude that

\[
\text{Nu} \approx \text{Ra}_K^{1/2}, \quad \text{when} \quad \text{Ra}_K^{1/2} \gg \text{Da}^{1/2} \quad (8a)
\]

\[
\text{Nu} \approx \text{Ra}_K^{1/4} \text{Da}_K^{-1/4}, \quad \text{when} \quad \text{Ra}_K^{1/2} \approx \text{Da}^{1/2} \quad (8b)
\]

For our set of runs, the first condition holds well. However, none of these studies incorporate the inertial effects, which arise due to the non-Darcy flow terms. In our set of experiments, these effects cannot be neglected. Plumb and Huenefeld [32] introduced the Forchheimer term into the boundary layer equations, and defined a Forchheimer parameter, \( \chi = f K^{0.5} \). They showed that the effect of the quadratic drag is to slow down the flow and, hence, retard the overall heat transfer. In an alternative analysis, Bejan and Poulikakos [33] showed that, for very high values of the Forchheimer parameter, the Nusselt number varies as the fourth root of the Rayleigh number, i.e., \( \text{Nu} \approx \text{Ra}_K^{1/4} \). Thus, for an intermediate range of Forchheimer parameters, the dependence of the Nusselt number may be expected to lie in between these two parameters. However, we did not find any such systematic variation in our experiments.
extremes. A regression analysis to our experimental data shows that, for our range of parameters,

\[
\overline{\text{Nu}} = 0.002 \, \text{Ra}^{0.35} \, \text{Da}^{-0.44}
\]  

(9)

Figure 4 shows a comparison of the above correlation prediction with the experimental data. As seen from the plot, the agreement is reasonable with an error band of ±20%. However, it may be mentioned here that Eq. (9) has been derived—based on experimental data on aluminum foams in air only—in absence of data on other materials and fluids. If such data are available in the literature in the future, the coefficient and indices in the equation may be changed keeping the same form of the equation and definition of the nondimensional parameters.

Finally, a comparison of the experimental data was also done with the theory of Kaviany and Mittal [34] for natural convection from a vertical plate to high permeability porous media. While the trends of the results look similar, the theory underpredicts the experimental Nusselt numbers by factors of 2 to 4.

5.2 Horizontal Orientation. The next set of experiments was conducted with the metal foam samples held in the horizontal orientation, and heated from the bottom as opposed to the side in the vertical orientation. The results are shown in Fig. 5. The trends are found to be similar as in the vertical orientation. The heat transfer coefficient increases with an increase in pore size due to higher permeability of the porous medium and, consequently, higher entrainment of air. On the other hand, for a given pore density, the heat transfer rate increases with the solid fraction—emphasizing the role of conduction, heat transfer through the solid.

As before, two correlations; one based on the fluid properties and other on effective properties, were derived. They respectively are

\[
\overline{\text{Nu}}_f = 11.24 \, \text{Ra}_f^{0.25} \, \text{Da}^{0.16}
\]  

(10)

\[
\overline{\text{Nu}} = 0.02 \, \text{Ra}^{0.2} \, \text{Da}^{-0.2}
\]  

(11)

The error bands in these correlations are ±7% and ±20%, respectively. Here, the Nusselt and Rayleigh numbers are defined as \(\overline{\text{Nu}}_f = hL^*/k_f\) and \(\text{Ra}_f = g\beta(T_b−T_{\text{amb}})L^*^3/\alpha_f\nu_f\), in accordance with Goldstein et al. [35], where \(L^* = L/4\). It may also be noted that the
length scale for the Darcy number in these cases was taken to be the height $H = 5$ cm of the metal foam sample, i.e., $Da = K/H^2$. This is in contrast to the vertical orientation, where the length scale was taken as $L$.

6 Buoyancy-Induced Convection in Finned Metal Foam Heat Sinks

The second part of our study involves buoyancy-induced convection in finned metal foam heat sinks. Referring to Fig. 6, a finned metal foam heat sink is essentially a longitudinal finned heat sink in which the space in the channel gaps between two adjacent fins is filled with metal foam. Calmidi [36] first proposed this heat sink and claimed that it combines the advantages of extended surfaces of fins and thermal dispersion in metal foams. Bhattacharya and Mahajan [37] conducted forced convection studies on these heat sinks using 5 and 20 PPI foam samples of 90% porosity. The number of fins was varied from one to four for 20 PPI. The results showed heat transfer enhancement up to 100% compared to normal metal foams with minimal attendant rise in pressure drop. In the present study, we evaluate the heat transfer performance of the heat sinks used in [37] in buoyancy-induced convection.

Fig. 6 Picture of a finned metal foam sample (20 PPI, four fins)

7 Experiments on Finned Metal Foam Heat Sinks

The finned heat sinks were fabricated by replacing the air gap in the channels of a longitudinal finned heat sink by metal foam. The metal foam samples were first cut to precise geometry using a bandsaw, and then bonded to the base plate and fin surfaces using the thermal epoxy Thermaxx® (Ablestick Inc.) having a reported thermal conductivity of 20–25 W/m K. These heat sinks were then tested under natural convection conditions, in vertical and horizontal orientations, in an experimental setup identical to that of the metal foam heat sinks. As before, the heat transfer coefficient for a typical run was calculated using Eq. (1).

The heat transfer coefficients for the three 20 PPI finned metal foam heat sinks are plotted in Fig. 7 as a function of $\Delta T$. It is seen that the heat transfer coefficient increases with an increase in the number of fins. However, the relative enhancement is progressively lower with the addition of each fin. This behavior, as in forced convection, can be attributed to the interaction of the boundary layers formed on the adjacent finned surfaces. For the 5 PPI samples having the same configuration, the heat transfer coefficients were found to be higher by 5 to 7% for the same temperature difference. Similar to regular metal foams, this can be attributed to the higher permeability of the porous medium resulting in lesser flow resistance. The maximum heat transfer coefficient attained for the four-finned heat sinks was found to be 57 W/m$^2$ K and 62 W/m$^2$ K at $\Delta T$ of about 50°C, respectively, for the 20 PPI and 5 PPI samples.

The next set of experiments was conducted with the finned metal foam heat sinks in the vertical orientation. The results are shown in Fig. 8. The trends are found to be very similar to those for the horizontal orientation. The heat transfer coefficient increases with an increase in the number of fins. However, the difference in the performances of the two-finned and four-finned heat sinks is found to be very small for both 5 and 20 PPI. As before, this can be attributed to the existence of an optimum number of fins.

8 Comparisons With Other Studies

For comparison with commercially available heat sinks, we chose one longitudinal finned product of AAVID Inc. [38]. The specifications of the heat sink are: Part No. 65530, base area: 8.937 in. $\times$ 3 in., number of fins: 14, and fin height: 3.072 in. For this heat sink, the manufacturer’s data sheet lists a thermal resistance of 1.4°C/W at a temperature difference of 14°C. This cor-
and 48 W/m² K, respectively. This implies that enhancements of optimized longitudinal finned heat sink in the entire sinks, and the 20 PPI heat sink with four fins, are superior to the experimental data in Figs. 7 and 8. It is seen that all the 5 PPI heat sinks, the improvements are 9.5% and 13%, respectively. However, it may be noted that our heat sinks have lower height (2.5 in.) compared to this commercial product (3.072 in.). Our study on buoyancy-induced convection in metal foams has shown that conduction plays a dominant role in enhancing heat transfer in this porous medium. Hence, the heat transfer coefficient is expected to increase with an increase in the height of the sample. Also, in the manufacturer’s data sheet, it is not mentioned whether this value is for the horizontal or vertical orientation of the heat sink. Applying the correlations of Bar-Cohen and Rohsenow [39], and Van de Pol and Tierney [40] for natural convection in finned heat sinks under vertical orientation, it is found that the reported value agrees with the correlation predictions. This indicates that the manufacturer’s data sheet is most likely for vertical orientation.

We next compare our finned metal foam heat sinks with longitudinal finned heat sinks of identical geometries in vertical orientation. Bar-Cohen and Rohsenow [39] formulated the following correlation for the Nusselt number that combines the heat transfer relations for buoyancy-induced convection adjacent to an isothermal flat plate and in a channel:

\[
\text{Nu} = \frac{hb}{k_f} = \left( \frac{576}{\text{Ra}^{1/3}} + \frac{2.873}{\sqrt{\text{Ra}}} \right)^{-0.5}
\]

They also showed that the optimum fin spacing for maximum heat transfer is given by

\[
h_{opt} = \frac{2.714}{\left( \frac{\text{Ra}^{1/3}}{b^4} \right)^{1/4}}
\]

when the thickness of the fins is neglected. Using these relations, the heat transfer coefficients for the optimized longitudinal heat sinks in vertical alignment were computed and compared to our experimental data in Figs. 7 and 8. It is seen that all the 5 PPI heat sinks, and the 20 PPI heat sink with four fins, are superior to the optimized longitudinal finned heat sink in the entire \(\Delta T\) range. For the 5 PPI heat sink with four fins in vertical orientation, the enhancement is about 65% at the lower values of \(\Delta T\) (~10°C), and decreases to about 24% at higher values (~50°C). In the horizontal orientation, the enhancements are 52% at lower values of \(\Delta T\), and 19% at higher values. It may be noted that, to the best of our knowledge, there are no comprehensive studies in the literature reporting the optimized configuration for longitudinal finned heat sinks in the horizontal orientation. As a result, we compared our horizontal cases also with the formulation of Bar-Cohen and Rohsenow [39].

Note that the optimum number of fins \(N_{opt}\) is a function of Rayleigh number, or equivalently \(\Delta T\). For example, in Figs. 7 and 8, \(N_{opt}\) at \(\Delta T\sim10^\circ C\) is 8; while at 50°C, it is 11. This is because at higher values of \(\Delta T\), or equivalently \(\text{Ra}\), the thermal boundary layer formed on the fin surfaces is thinner, which allows for more fins to be accommodated for a fixed heat sink width without experiencing the degradation in heat transfer due to overlapping thermal boundary layers. As a result, the value of \(N_{opt}\) increases with \(\Delta T\). On the other hand, the number of fins for our finned metal foam heat sinks remains constant. Hence, the relative enhancement in heat transfer goes down with \(\Delta T\).

It may be noted that our finned metal foam heat sinks, which are found to be superior to the longitudinal finned heat sinks, are not optimized. The thermal performance can be further enhanced by optimizing the number of fins and fin thickness. We also note that, in all our comparisons with the longitudinal finned heat sinks, we assumed fins to be isothermal at the base plate (i.e., fin \(\eta=100\%\)). In reality, the fin \(\eta\) and, hence, heat transfer coefficients for the longitudinal finned heat sinks will be lower than those given by Eq. (12). Or, in other words, the gains reported above are conservative.

As a last comparison, we compare our results with those reported in Sahraoui and Refai-Ahmed [41] on flat plate fin heat sinks in natural convection in the vertical orientation. In their study, they had optimized a finned heat sink having a base area of 300 mm \(\times\) 300 mm (12 in. \(\times\) 12 in.) and height of 50 mm (2 in.). Setting a target temperature difference of 20°C, they found the optimum number of fins to be 23. The minimum value of thermal resistance obtained at this optimum point was about 0.4°C/W. This corresponds to a heat transfer coefficient of 27.78 W/m² K. For our heat sinks, the values of \(h\) at \(\Delta T=20^\circ C\) for the 20 PPI samples in the vertical orientation are 35 W/m² K,
42 W/m² K, and 43 W/m² K for the one-, two-, and four-finned metal foam heat sinks, respectively. For 5 PPI, the values are 44 W/m² K, 48 W/m² K, and 50 W/m² K.

Finally, it may be recalled that in our heat sinks, the metal foam was bonded to the base and fins using thermal epoxy. This can result in potentially high contact resistance, thereby decreasing the heat transfer rate. A better method of attachment, such as brazing, can then lead to even higher values of "h".

Future work is planned to study the effect of the heat sink height in enhancing heat transfer to arrive at the optimum heat sink configuration for metal foams. The finned metal foam heat sinks can also be optimized in terms of porosity, number of fins, and fin height. The resulting optimal configurations are expected to outperform the existing finned heat sinks and lead to even higher values of heat transfer coefficients. However, it should be noted that these metal foams offer considerable flow resistance to the fluid transport due to the tortuosity of flow paths. Hence, fluid entrainment spaces—on the sides as well as on the top—play an important role in heat transfer in these forms of porous media. In our experiments, adequate entrainment space was allowed from all sides. However, if, in practice, enough entrainment is not provided due to stringent space constraints, the enhancement in heat transfer may go down. Studies on this aspect also form part of our future work.

9 Conclusions

Experimental data on buoyancy-induced convection in metal foams in the horizontal and vertical orientations are presented. The results indicate that heat transfer from a heated surface is considerably enhanced when metal foam is bonded onto the heated surface. This enhancement is primarily due to conduction in the metal fibers and interfacial heat exchange between the solid and fluid phases. The enhancement depends on the porosity and pore size of the foam sample. It was observed that the heat transfer rate is enhanced for larger pore sizes and lower porosities.

The experiments on finned metal foam heat sinks show a considerable enhancement in heat transfer rates compared to those for regular metal foam heat sinks. Further, the finned metal foam heat sinks were found to be superior in thermal performance to optimized longitudinal finned heat sinks. The heat transfer coefficient is found to increase with an increase in the number of fins. The relative enhancement is, however, found to decrease with each additional fin, which indicates the existence of an optimum number of fins, beyond which the enhancement in heat transfer due to increased surface area is offset by the retarding effect of overlapping thermal boundary layers. The 5 PPI samples are found to be superior to the 20 PPI heat sinks in thermal performance due to the higher permeability of the porous medium.

Acknowledgment

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Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>area, m²</td>
</tr>
<tr>
<td>a</td>
<td>surface area, m²</td>
</tr>
<tr>
<td>b</td>
<td>channel gap, m</td>
</tr>
<tr>
<td>C</td>
<td>empirical coefficients</td>
</tr>
<tr>
<td>C_p</td>
<td>specific heat, J/kg K</td>
</tr>
<tr>
<td>Da</td>
<td>Darcy number =K/L² or K/H²</td>
</tr>
<tr>
<td>d</td>
<td>diameter, m</td>
</tr>
<tr>
<td>E</td>
<td>drag force, N/m³</td>
</tr>
<tr>
<td>f</td>
<td>inertial coefficient</td>
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<tr>
<td>g</td>
<td>acceleration due to gravity, 9.81 m/s²</td>
</tr>
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<td>H</td>
<td>height of the porous medium, m</td>
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<tr>
<td>h</td>
<td>heat transfer coefficient, W/m² K</td>
</tr>
<tr>
<td>K</td>
<td>permeability, m²</td>
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<tr>
<td>k</td>
<td>thermal conductivity, W/m K</td>
</tr>
<tr>
<td>L</td>
<td>length of the porous medium, m</td>
</tr>
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<td>L*</td>
<td>characteristic length, m=(A/P)</td>
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<tr>
<td>N</td>
<td>number of fins</td>
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<tr>
<td>n</td>
<td>correlation index</td>
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<td>Nu</td>
<td>Nusselt number</td>
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<td>perimeter of base, m</td>
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<td>Pr</td>
<td>Prandtl number = μC_p/k</td>
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<td>temperature, K</td>
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Subscripts

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<td>amb</td>
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Greek Symbols

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<td>ν</td>
<td>kinematic viscosity, m²/s</td>
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<tr>
<td>μ</td>
<td>viscosity, N/m s</td>
</tr>
<tr>
<td>α</td>
<td>thermal diffusivity, m²/s</td>
</tr>
<tr>
<td>β</td>
<td>coefficient of volume expansion, K⁻¹</td>
</tr>
<tr>
<td>ε</td>
<td>porosity</td>
</tr>
<tr>
<td>ρ</td>
<td>density, kg/m³</td>
</tr>
<tr>
<td>χ</td>
<td>Forchheimer parameter, m</td>
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</tbody>
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References