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Macroscopic correlation for natural convection in water saturated metal foam relative to the placement within an enclosure heated from below

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ABSTRACT

Natural convection in water saturated metal foams is explored though experiments and numerical simulations. From these data a new generalized heat transfer correlation is developed for saturated metal foam with superposed water layers heated from below with arbitrary placement of the foam with respect to the bounding heated and cooled surfaces. The macroscopic correlation allows prediction of heat transfer with known or easily measurable properties of the foam. For Rayleigh numbers from 10^6 to 10^8, water saturated copper foam enhances heat transfer compared to pure water when the foam is adjacent to the heated or cooled boundary, but immersion of the foam without contact with the boundary provides no benefit. The enhancement of heat transfer is attributed primarily to conduction.

1. Introduction

Metal foams are lightweight, rigid, and offer high specific surface area and low flow tortuosity favorable for both free and forced convective heat transfer. In the present work, we focus on natural convection in water saturated metal foams and present a generalized heat transfer correlation for superposed metal foam and water layers heated from below with arbitrary placement of the foam with respect to the bounding heated and cooled surfaces. The study expands the more limited studies by Kathare et al. [1,2] of natural convection in a water filled enclosure heated from below with a horizontal layer of copper foam either filling the enclosure [1] and or partially filling the enclosure [2]. The results answer the questions: Can an enclosure partially filled with copper foam provide similar enhancement of heat transfer to a fully filled enclosure and does it matter where the foam is placed? The effects of changing the foam placement (on one, both or neither boundary), the relative height of foam in the enclosure, and the Rayleigh number on natural convection are investigated experimentally with supporting numerical analysis to provide additional cases and to visualize the underlying flow field.

Correlations for the natural convection Nusselt number are developed for the five cases illustrated in Fig. 1. These cases include (a) the Rayleigh–Bénard problem for a fluid filled cylinder heated from below, (b) water saturated foam heated from below, (c) water saturated foam with a superposed fluid layer above, (d) water saturated foam adjacent to the heated and cooled boundaries with fluid in between, and (e) water saturated foam in the center of the enclosure with superposed fluid layers above and below. The use of the dimensionless geometry factor [3], which connects the macroscopic and microscopic drag and heat flux between the solid and fluid phases allows prediction of heat transfer with known or easily measurable properties of the foam. The geometry factor,

$$\eta = \frac{A_{fs}\, d}{V_s}, \tag{1}$$

represents the distribution of the solid, which is specified solely by the shape of the solid [3]. Su et al. [3] show that the geometry factor for metal foams is accurately represented by,

$$\eta \approx \frac{0.0254 \times 6d\left[\pi d_p d + 3\sqrt{3}(d_p + d)^2/4 - \pi d_p^2/2\right]}{(1-\phi)(d+d_p)^4}, \tag{2}$$

where d_p is the characteristic dimension of the pore. The pore size, conventionally given in pores per inch, PPI, is

$$\mathrm{PPI} \approx \frac{0.0254}{d + d_p}. \tag{3}$$

2. Approach

2.1. Experimental method

Table 1 lists the operating conditions for the experiments for cases (a–d) depicted in Fig. 1. With foam on the hot boundary, natural convection heat transfer experiments were performed for 5

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Nomenclature

A	area, m^2
$\mathbf{B}$	volumetric drag force, N/m^3
c	microscopic drag coefficient constant
c_p	specific heat at constant pressure, J/kg-K
C_D	microscopic drag coefficient, Eq. (16)
C_F	Forchheimer coefficient
d	microscopic scale length scale, m
d_p	pore diameter, m
D	diameter of the enclosure, m
Da	Darcy number, K/L^2
$\mathbf{g}$	gravitational constant, kg-m/s^2
H	height of the enclosure, m
H_p	refers to the height of the porous layer, m
k	thermal conductivity, W/m-K
k'	thermal dispersion conductivity, W/m-K
k_m	effective thermal conductivity, Eq. (14), W/m-K
K	permeability, m^2
M	constant $0 < M < 1$ for Eq. (14)
n	coefficient of the macroscopic heat transfer correlation
Nu_d	microscopic Nusselt number based on microscopic scale, hd/k_f
Nu_H	transient fluid Nusselt number based on H
$\overline{\mathrm{Nu}}_H$	time averaged Nu_H
Nu_m	Nusselt number based on porous medium $(k_f/k_m)\mathrm{Nu}_H$
$\overline{\mathrm{Nu}}_m$	time averaged Nu_m
p	volume averaged pressure, $\phi\hat{p}_f$, N/m^2
PPI	pore density, pores/inch
Pr	Prandtl number, ν/α
Pr_m	porous media Prandtl number for the composite system, $\nu_f c_{p,f}/k_m$
Pr_p	the dimensionless grouped Prandtl number, $\mathrm{Pr}_m/C_F\sqrt{\mathrm{Da}}$
r, z	cylindrical coordinate, m
Ra_d	microscopic Rayleigh number, $\frac{g\beta\Delta T d^3}{\nu_f\alpha_f}$
Ra_H	macroscopic Rayleigh number, $\frac{g\beta\Delta T H^3}{\nu_f\alpha_f}$
Ra_m	porous medium Rayleigh number, $\mathrm{Ra}_H \mathrm{Da}\, k_f/k_m$
Re_d	microscopic Reynolds number, $|\hat{v}_f|d/\nu_f$
t	time, s
t_0	period for time average, s
T	temperature, K
v	microscopic velocity, m/s
$\mathbf{v}$	Darcy velocity, $\phi\hat{\mathbf{v}}_f$, m/s
V	elementary volume in REV, m^3

Greek symbols

α	thermal diffusivity, m^2/s
ϕ	porosity
ΔT	temperature scale, K
η	dimensionless geometry factor, $A_{fs}d/V_s$
μ	dynamic viscosity of the fluid, N-s/m^2
ν	kinematic viscosity, m^2/s
ρ	density, kg/m^3

Subscripts

f	fluid
h	heat transfer coefficient
m	porous medium
s	solid

Superscripts

—	time average
$*$	dimensionless variable
^	macroscopic variable
∞	far field

and 10 PPI foam, Rayleigh numbers based on the total height of the enclosure and the water properties from 2×10^6 to 5×10^8, height to diameter aspect ratios from H/D = 0.1–0.8, and ratios of the total foam layer height to the total height of the apparatus, H_p/H equal to 0.25, 0.5, and 0.75. With foam on both boundaries, experiments were performed with 10 PPI foam, $1 \times 10^7 \leqslant \mathrm{Ra}_H \leqslant 3 \times 10^8$, $0.3 \leqslant H/D \leqslant 0.8$, and H_p/H equal to 0.5 and 0.75. Natural convection heat transfer experiments were also performed for water alone for $9 \times 10^5 \leqslant \mathrm{Ra}_H \leqslant 3 \times 10^8$ and $0.2 \leqslant H/D \leqslant 0.8$. The effective thermal conductivity of the foam was measured at Rayleigh numbers below the onset of convection.

The cylindrical copper foam disks are 127 mm DIA, with porosity ranging from 0.88 to 0.92, and pore densities of 5 and 10 PPI. The geometry factor η = 3.928. The permeability, K, and drag coefficient, C_f, of the foam were determined by measuring the pressure drop across foam samples in specialized wind tunnel for this purpose.

The test chamber, previously described in detail [1,2], was a well-insulated vertical acrylic cylinder of 12.7 cm ID with a heated lower boundary and a vertically adjustable constant temperature upper boundary. The working fluid was degassed water. Heating from the bottom was provided by a resistance element located beneath a 9.53 mm thick copper bottom plate. A similar guard heater and a separator plate assembly at the bottom minimized heat loss to 3% of the applied flux. The top plate was held at 292 ± 0.5 K with a spiral wound brazed copper tube through which cooling water was circulated.

Temperatures were measured with 30 gauge type-T thermocouples individually calibrated with reference to a Class A RTD. The largest thermocouple bias error was 0.25 K. An ice bath served as the reference temperature during experiments. For the top and bottom copper plates, six thermocouple junctions were located 0.53 mm beneath the surface in contact with the metal foam–water medium. Similarly positioned thermocouples were located on the bottom plate beneath the surface and in the copper plate below the acrylic separator plate. Temperatures of the bounding surfaces were recorded at steady state ~2 h after the initiation of heating.

In the reduction of data, all thermophysical properties were evaluated at the mean of the lower and upper surface temperatures. The overall heat transfer coefficient was determined by the difference between the average temperatures of the upper and lower surfaces and the net heat flux through the foam-water layer. Maximum experimental uncertainty in the Darcy, Rayleigh, and Nusselt numbers is 8%, 11% and 12% respectively.

2.2. Numerical method

The numerical data yields the temperature and velocity fields for the experimental cases (a–d) and case (e) which was not included in the experiments. The local thermal equilibrium (LTE) model was applied to solve for the temperature and velocity distributions within the porous and fluid-only regions. The approach follows that presented by Su et al. [3]. The dimensionless governing equations based on a macroscopic length scale equal to the height of the enclosure H, a temperature scale $\Delta T = T_{hot} - T_{cold}$, and the velocity scale $\sqrt{g\beta H\Delta T}$, are

$$\hat{\nabla}^* \cdot v^* = 0, \tag{4}$$