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Technical Note

Experimental investigation of forced convection heat transfer augmentation with metallic fibrous materials

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1. Introduction

Efficient heat spreaders and dissipaters are continually on demand for various industrial applications. One such heat transfer mechanism is to make use of metallic fibrous media. These have many desirable characteristics. They weigh less, and are inexpensive. The fibrous media deliver high heat transfer rates and keep the temperature of the dissipating surface to a low value. The fibers together have a large surface area and are efficient in distributing the convective heat transfer. In addition, the random nature of fibers further enhances heat transfer through a mechanism called thermal dispersion [1]. Fibrous heat spreaders are promising for various cooling applications.

Several theoretical and some experimental investigations have been reported in the literature, which address particular fundamental aspects of convective heat transfer in porous media. Koh and coworkers [2,3] have shown analytically and experimentally that porous inserts in a convective cooling system significantly enhance heat transfer. More recent studies that considered heat transfer enhancement using porous media include [4,5]. Angirasa and Peterson [1] presented a more detailed discussion of relevant theoretical and numerical studies.

The aim of the present experimental investigation is to demonstrate the heat transfer enhancement with metallic fibrous heat dissipaters. For this purpose, an aluminum square plate of 2.54 cm side is air-cooled by a fibrous block brazed to the plate. Experimental measurements for forced convection heat transfer with air are presented.

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2. Fibrous heat dissipater

The experimental study was designed to investigate the forced convective air-cooling with two aluminum fibrous blocks differing in porosity and fiber structure. Each aluminum fibrous block measured 2.54 cm $\times$2.54 cm base, and 1.27 cm thickness. The fibrous block has an open cell matrix configuration (see Fig. 1). Porosity (ϕ) here is defined as the volume fraction of the void in the fibrous block. The porosities of the samples studied were 0.97, and 0.93. Photographs of representative fibrous heat sinks are shown in Fig. 1. Numerical calculations [1] have indicated that the preferred fibrous medium for heat transfer enhancement will have larger porosities (indicated as above) and thinner fibers. Typical values of the permeability (K) of the fibrous media are 10^{-8}–10^{-7} m^2.

An aluminum fibrous block is manufactured by the directional solidification of super-heated liquid metal. This process takes place in an inert gas/vacuum environment. The open structure of the rigid foam is made up of diodecahedronal-shaped cells connected by continuous solid metal ligaments. The ligaments are not porous, and they approximate single strand wire. The test piece comprised of the fibrous block brazed to an aluminum plate of 2.54 cm $\times$ 2.54 cm base, and 0.32 cm thick.

3. Experimental setup

The tests were carried out in a wind tunnel made of plexiglass having a square cross-section and internal side of 5.43 cm (Fig. 2). Air-flow at room temperature was effected by connecting a Central Blower Company variable speed fan to one side of the wind tunnel which

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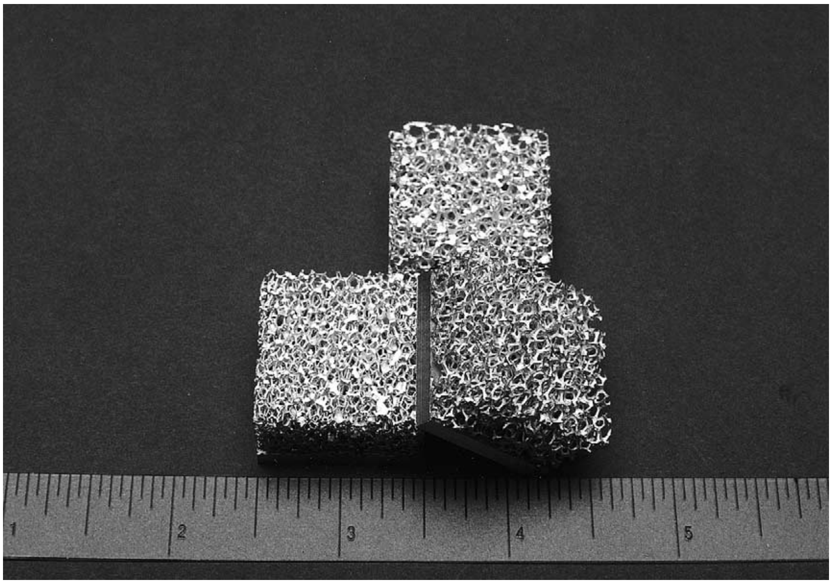

Fig. 1. Photograph of some aluminum fibrous heat blocks.

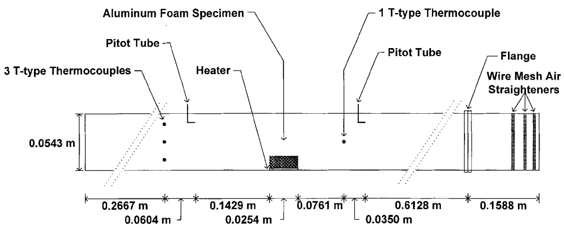

Fig. 2. Sketch of the experimental setup.

acts as a suction fan. The entrance region of the wind tunnel contained three copper wire-mesh air-straighteners with approximately 80% porosity. These were mounted about 5 cm apart.

The bottom side of the plexiglass wind tunnel contained a 2.54 cm × 2.54 cm hole where the fibrous heat block was mounted. The distance from the entrance of the channel to the test piece is fifteen times larger than the hydraulic diameter of the wind tunnel. Any gaps between the plexiglass and the aluminum plate on the fibrous heat block were sealed with high temperature RTV silicon. The test piece was held flush with the outer edge of the wind tunnel wall with the aid of a ceramic insulation strip which also helps in reducing the heat losses to the ambient. A fiberglass insulation was placed below the ceramic insulator strip as a further mechanism for reducing heat losses.

The base plate of the heat sink was heated by a 2.54 cm × 2.54 cm flexible silicone rubber heater with a rated wattage of 10 W. A thin layer of high temperature RTV silicon was used for attaching the strip heater to the aluminum plate. The power supplied with the heater was calculated by the formula, $W = P_R \times (V_A^2/V_R^2)$, where W is the actual wattage, P_R is the rated wattage, V_A is the actual voltage, and V_R is the rated voltage. The rated voltage of the heater was 115 V AC and V_A was measured with a multimeter. Variable voltage was supplied from a wall AC outlet through a variable autotransformer. The experimental uncertainty of the instrument in the measurement of voltage was ±0.5% of the measured value.

Temperatures of the air were measured using T-type copper-constantan thermocouples, one upstream and three downstream of heat sink. For measuring the temperature of the aluminum base plate of the heat sink four small holes were drilled in the plate (1.18 mm in diameter). Copper-constantan thermocouples were inserted with thermal grease into these holes, and then were sealed in place with an epoxy. The thermocouple measurements were taken with Omega Thermocouple Thermometer. The uncertainty in the measurement of temperature was ±0.5 °C.

Air velocity at two locations was calculated from pressure measurements with pitot static tubes and Dwyer micromanometers. The uncertainty of the instrument was ±0.15 Pa. The second measurement, however, was not necessary for the results reported here, but was used to verify the trends of the velocity measurements, and the consistency of the pressure measuring devices.

4. Experimental procedures

The tests were carried out at different values of air velocity and power for each of the two samples. The velocity varied between 2 and 9 m/s and power, 3.7 and 9.2 W. Although the highest possible operational temperature of the heater was 120 °C, the heat dissipater was effective in keeping the maximum measured temperature of the base plate to 50 °C.

Data were taken in steady-state condition, which was assumed to have reached if the base plate temperature did not vary by ±0.1 °C during an interval of 10 min. Steady-state conditions usually prevailed within 30 min. The upstream temperature almost always coincided with the room temperature. The downstream temperature was obtained by averaging the three thermocouple readings. No fluctuations were observed in the temperatures of the downstream flow. The temperature of the base plate was obtained by averaging the values obtained from the four thermocouples. The four values were nearly identical in most cases.

Heat losses due to conduction, natural convection, and radiation were estimated as follows. Small amounts of heat were supplied to the heat sink with the air blower off. This heat input is essentially the same as the heat losses under forced convection under similar temperature potential between the plate and the ambient, provided the forced convection velocities are substantially larger than the natural convection velocities. At these higher velocities the interaction between natural and forced convection may be considered to be simply additive. The heat loss under natural convection conditions was plotted as a function of temperature-difference, and a least-square fit was obtained. The heat losses calculated from this fit were deducted from the forced convection heat input measurements.

The overall thermal resistance (R_t) was calculated by dividing temperature difference ($\Delta T = T_w - T_a$) with the heat rejected. The Nusselt number was obtained as, $Nu = D_h/(R_t A k)$, where D_h is the hydraulic diameter of the wind tunnel, R_t is the overall thermal resistance, A is the area exposed to air flow, and k is the thermal conductivity of the air at film temperature. For the sake of comparison with heat transfer from a flat plate this area was taken to be the surface area of the base plate, i.e., 2.54 cm × 2.54 cm. The Reynolds number is given by, $Re = VD_h/\nu$, where V is the air velocity, and ν is the kinematic viscosity of air at the film temperature. The Reynolds number range for the experiments was 17,000–29,000.

5. Results and discussion

To test for the repeatability of the experimental measurements the temperature difference between the plate surface and the ambient was determined for various velocities. The porosity of the test piece for this experiment was 0.93 and the power input was approximately 6.6 W. Two sets of readings were taken on different days and the results were plotted (not shown here for brevity). The data sets fell on the same line well within the limits of experimental uncertainties.

The base to ambient temperature difference is plotted for different Reynolds number and power input in Figs. 3 and 4 for porosities of 0.97, and 0.93, respectively. The temperature difference is fairly invariant over a range of Reynolds number. The fibers are efficient in distributing the flow uniformly over the media with the resultant effective thermal dispersion. Comparing the temperature differences in Figs. 3 and 4 at the same power input, it is seen that the heat sink with lower porosity keeps the surface to a lower temperature because of the larger surface area exposed to convective heat transfer.

The experimental measurements of losses due to natural convection, radiation, and conduction, were obtained under no-flow condition. The losses for the two blocks are closer because of the close proximity of the values of their porosities. Nevertheless, the larger

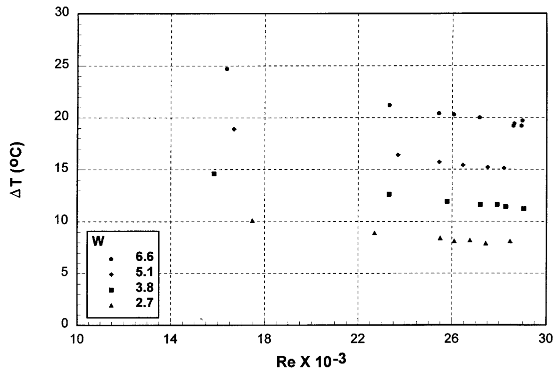

Fig. 3. Plate to air temperature difference for $\phi = 0.97$.

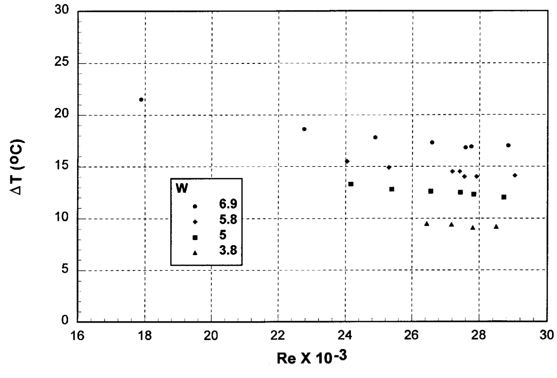

Fig. 4. Plate to air temperature difference for $\phi = 0.93$.

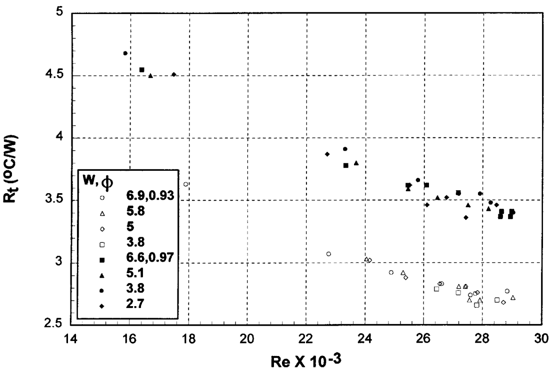

Fig. 5. Overall thermal resistance.

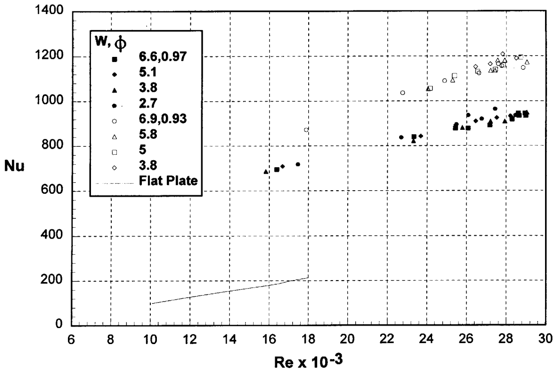

Fig. 6. Average Nusselt number.

porosity sink yielded slightly higher heat losses because of better natural circulation of air in the porous block. Separate line fits were obtained by least-squares method for these two data sets to yield the relations $q_{\mathrm{loss}} = 0.0489(\Delta T) - 0.0348(\phi = 0.97)$, and $0.0485\ (\Delta T) - 0.0598(\phi = 0.93)$. The heat losses for forced convection experiments were calculated from the above equations and deducted from the heat input.

The overall thermal resistance for the two heat sinks is plotted as a function of Reynolds number in Fig. 5. With increasing Reynolds number, the thermal resistance expectedly decreases for both sinks. The thermal resistance for the sink with higher porosity is larger for a given Reynolds number because of smaller surface area exposed to the flow. However, this should not lead to the conclusion that fibrous media with lower porosity will lead to lower resistance. There is a trade off between the larger surface area for heat transfer and better distribution of flow around fibers that leads to higher heat transfer rates. At sufficiently smaller porosity, the latter mechanism substantially diminishes thereby increasing the thermal resistance.

Fig. 6 shows the average Nusselt number as a function of Reynolds number. For the reasons given with reference to thermal resistances in Fig. 5, the sink with lower porosity delivers higher heat transfer rates. As can be seen in Fig. 6, a fibrous block of even 0.5 in. thick can achieve substantial heat transfer enhancement compared to a plain flat plate under the same forced convection conditions.

6. Conclusions

An experimental study to test the effectiveness of metallic fibrous heat dissipaters was presented. Within a high porosity range (above 90%), porous blocks with lower porosity typically have lower thermal resistance and higher heat transfer rates. Under natural convection conditions these trends are reversed although the differences are rather insignificant. Metallic porous heat dissipaters can achieve substantial heat transfer augmentation when compared to plain flat plate.

Acknowledgements

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