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ANALYTIC OPTIMIZATION OF POROUS MEDIUM HEAT SINKS FOR ENERGY EFFICIENCY AND MINIMUM MASS

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ABSTRACT

Use of microchannel heat sinks for high heat flux applications is substantial for thermal management and it is also critical for scalable power generation. For both applications, the energy efficiency consideration of the pump power is crucial. A number of models have been created that predict the performance as a function of the geometrical parameters, taking into account, the pressure loss over the length and volume constraints. Most of the approaches either involve sophisticated calculations incorporating fluid dynamics in channels, or have an analogy with the pin-fin model, which gives simpler calculations but considers only a single laminar flow regime for optimization. Even with the simplified models available, the geometrical impact on mass and pumping power is nonlinear and not apparent for optimization. We propose an optimization of porous medium heat sinks with respect to the heat transfer rate, mass, and pumping power. These are functions of the simplest geometric parameters, i.e. porosity, pore density, and length of the porous medium.

Considering large production, mass (cost of raw material) is nearly proportional to the cost of the heat sink, we consider minimizing the mass for indirectly minimizing the overall cost. The other factor for saving energy considered here is the pumping power. This connects to both the heat transfer rate and the power consumption to drive the fluid through the porous medium.

The optimization is performed for a specific value of porosity and length of the heat sink. The model considers the effect of flow through the porous medium and the effective thermal conduction as a function of combined conductivity of the solid ligaments and the fluid in pores.

An optimum coefficient of performance (COP) is found at over 90% of porosity for minimum mass, pumping work and maximum heat transfer. This mathematical expression of the model will give a quantifiable figure-of-merit to take into account the impact of the mass and the pumping power on the performance to cost ratio.

NOMENCLATURE

A: area, m^2
B: base plate height, m
d:pore size, m
H: total height of the heat sink, m
h: heat transfer coefficient, W/m^2K
K: permeability, m^2
k:thermal conductivity, W/mK
L: length along direction of flow, m
m:mass, kg
n: number of fins
P: pressure, N/m^2
Q: total heat transfer, J
q:flow rate, Kg m/s
R: Relative Density
Re_k: Reynolds number
T: temperature, K
t:thickness, m
U: velocity, m/s
V: volume, m^3
W: work, J
x:length, m

Greek symbols

μ: dynamic viscosity Ns/m^2
ε:porosity
θ:dimensionless temperature difference
ρ:density kg/m^3
σ:specific surface area,m^2/m^3

Subscripts
b:base
cond:conduction
conv:convection
eff:effective
f:fluid
fm: foam
in:inlet
L: long length
p: overall equivalent
s:solid
sf:surface
t:total

INTRODUCTION

The current trends for exponential increase in the performance capabilities of the integrated circuits and processors require better heat sinks which not only have adequate performance but also are economical and energy friendly. Microchannel heat sinks with liquid cooling are one of the viable solutions to meet this requirement. Numerous analytical models have been developed to characterize the performance. These models are, unfortunately, complex and involve sophisticated calculations pertaining to fluid dynamics. Some models have a simplified approach with approximations in the model and that limits the applications. In order to achieve a cost/energy specific maximum performance, we need to analyze the dependence of design parameters of the heat sink on the performance. Considering the trade-offs in the impact of design parameters we will find an optimum for a specific performance. For the simplicity and generalization of the heat sink geometries, this work considers a porous heat sink as a representation of general heat sinks. The model allows a fluid flow through it, normal to the direction of heat conduction, Fig. 1.

This paper presents the analysis of the interdependence of the geometric parameters on the cooling performance measures, i.e. heat transfer coefficient, pressure drop, and permeability. These parameters are functions of design variables, i.e. porosity, pore density, and the dimensions of the heat sink. The impact of the material properties, such as density, thermal conductivities of the solid and fluid, and viscosity are also investigated. We introduce a figure-of-merit (FOM) defined to give a quantifiable measure of the cost to performance ratio. The goal is to maximize the FOM. There are two costs associated with heat sinks, the operating cost, which is a function of the work required to pump the fluid through the heat sink and the manufacturing cost which primarily depends on the mass of the material. The performance of the heat sink can be measured as the net heat transfer for a given set of operating conditions and design.

FOM is evaluated as a function of the porosity while all the other parameters are assumed constant including the pore density. It is observed that the relationships for permeability and heat transfer coefficient itself change with the value of pore density and hence for simplicity it is assumed constant. For the comparison with conventional heat sinks, the coefficient-of-performance (COP) for the porous heat sink is also evaluated. The COP is also a function of porosity. We analytically optimize the design parameters for the COP and above FOM as a function of porosity and other geometric variables.

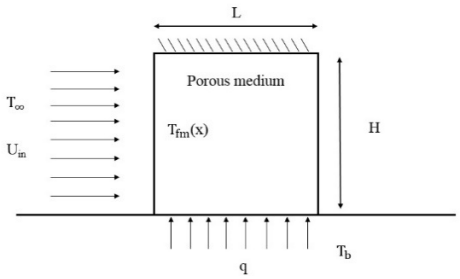

Fig. 1 Schematic representation of the heat sink

MODEL

The parameters associated to determine the FOM and COP are: permeability, Reynolds number, heat transfer coefficient, an equivalent temperature difference, pumping work and the mass. We analyze these parameters individually first. Then, we will combine them into the COP and FOM analysis. Porosity is defined as the ratio of the void volume to the total volume of the sample.

$$\varepsilon = \frac{V_f}{V_t} = 1 - R \qquad (1)$$

Where 'R' is the relative density. Porosity is a dimensionless quantity.

1) Permeability

The permeability is a measure of the ability of the geometry to allow flow of fluid though the medium. It is a function of porosity, pore density, interconnectivity, orientation and tortuosity [1].A computational and experimental analysis with respect to the geometric dimensions for a unit cell of a homogenized model has been carried out in [1]. Numerical Analysis for permeability was presented in [2] and [3] with respect to porosity, pore density and the net surface area. These analyses discussed have the porosity as the primary variable of permeability.

The permeability in case of experimentation is determined using homogenized version of Darcy's law [4], [5-6]. The experimental data for a variety of porous material including metal as well as organic foams [1], [7-11] suggests that the permeability is an exponential function of the porosity. .

With more careful consideration, however, this exponential relationship holds true only for high (> 0.5) porosity values and the permeability varies linearly with the porosity at low values of porosity (<0.5) [9]. Additionally, experimental data [7] suggests that the dependence on the porosity decreases as the pore density, which is number of pores per inch (PPI), increases and finally reaches to a point where the permeability becomes completely independent of the porosity.For this analysis, a 10 PPI of foam structure with a body-cubic-center (bcc) lattice structure is considered. The bcc structure is a homogenous distribution of spherical voids with a body centered cubic unit cell. The expression for permeability is obtained by fitting a curve through the experimental data in [7]. Both linear and exponential relationships are evaluated and two different trends for COP curve are analyzed.

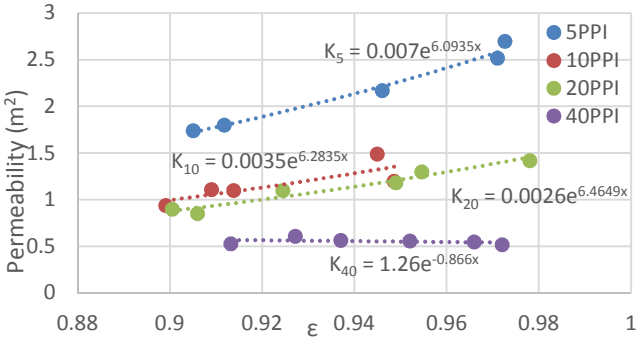

Fig.2 Experimental Values for Permeability for different pore densities reproduced from [7]

The exponential expression is given by

$$K = ae^{b\varepsilon} \qquad (2)$$

Where, variables 'a' and 'b' vary according to the pore density. Fig. 3 shows the permeability as a function of porosity.

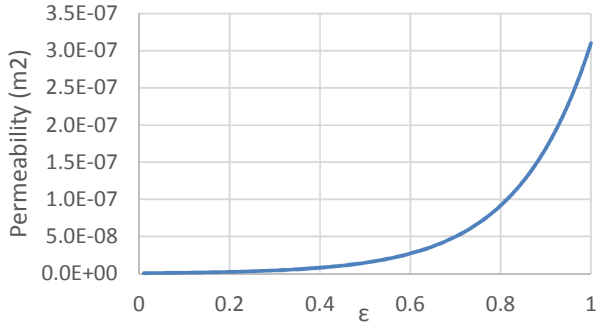

Fig. 3 Permeability v/s Porosity (10PPI foam)

2) Reynolds Number

The Reynolds number is found from the expression of friction factor and knowing the pressure gradient in the Darcy's equation. The friction factor is equal to the sum of inverse of the Reynolds number and the Ergun's Coefficient. This expression is empirical for the Reynolds number 'Re_k', specifically analyzed for porous mediums and is shown to hold true by Paek et al [3]. For lower values of porosities the Ergun's coefficient is relatively negligible [3]. Hence,

$$f = \frac{1}{Re_k} + C_E \approx \frac{1}{Re_k} = \frac{(-\frac{\partial P}{\partial x})\sqrt{K}}{\rho U_{mod}^2} \qquad (3)$$

This expression is widely used in numerous literatures and agrees well with the experimental work for metal foams, [3], [6], [12], and [13] as well as numerical [2]. Reynolds number is considered as a function of permeability, porosity and the inlet velocity, while the viscosity and the density of fluid are assumed to be constant.

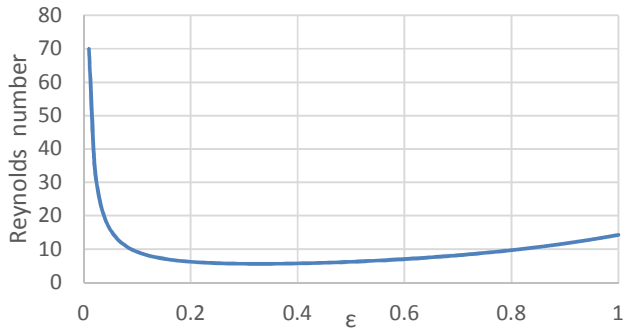

Fig. 4 Reynolds number v/s Porosity

Also a flow velocity that satisfies the low Reynolds number condition is assumed. The velocity lies in the normal operating conditions for a heat sink. (1 m/s, Air)

$$R_{ek} = \frac{\rho U_{in}\sqrt{K}}{\mu\varepsilon} \qquad (4)$$

Note that not only is Reynolds number a function of the porosity, but so is permeability. Fig. 4 shows the relation. The Reynolds is very high (tends to infinity at 0) at low porosity values as the fluid velocity will be considerably large. It drops exponentially and the gradually increases at high porosity values as the permeability term starts to dominate.

3) Heat Transfer Coefficient for porous media

There are multiple methods to determine the heat transfer coefficient. Kim et al [13] use a space averaging technique wherein the heat transfer coefficient is evaluated experimentally by knowing the operating temperatures, the inlet velocities, and the material properties. In case of experimentation the heat transfer coefficient is found out by energy balance when the temperatures at all the interfaces are known [6],[14]. Multiple fin model has also been used before to approximate the porous media as multiple fins [15]. The number of fins in approximation depends on the porosity while the dimensions of the fins depend on the

overall dimension of the medium. The overall heat transfer is evaluated for the array of fins and correlated to the heat transfer for the porous medium. . Bodla et al determined the heat transfer coefficient analytically for constant heat flux boundary condition in the fluid regime using the governing equation defined by the energy balance condition. Similar kinds of approaches exist in other literatures which differ in some assumptions and techniques for volume averaging [7], [16]

In this analysis, instead of approximating geometries or volume averaging the medium, we approximated the heat transfer coefficient from the Reynolds number. There is a correlation observed between the Reynolds number and heat transfer coefficient in the experimental data [12]. The experimental work was done for a temperature range 350-400K. We used the regression curves and yielded a logarithmic relationship between the Reynolds number and heat transfer coefficient as,

$$h = \alpha Re_K^{\varepsilon\beta} \quad (5)$$

Where, α = 9 and β= 0.85 for a 10 PPI foam.
This correlation holds true when the Reynolds number is small. The principle being that the heat transfer is dominated by conduction at low Reynolds number whereas convection dominates at high Reynolds number. This heat transfer coefficient is an average for the entire heat sink.

The relation of the heat transfer coefficient to the porosity observes a similar trend to that of the Reynolds number in Fig.4. The value of heat transfer coefficient lies between 100 W/m^{2}K and 200 W/m^{2}K.

4) Specific surface area

Specific surface area σ [m^2/m^3] is the surface area of fluid-solid contact per unit volume for the fluid flowing through the medium. Surface area depends on the pore density and the porosity. The surface area here actually refers to the area of contact between the fluid and the solid assuming that there are continuous flow paths and continuous solid bridges for any porosity. Zero porosity will have zero surface area and as the porosity increases, the contact surface area goes on increasing until reaching a point that the solid volume no longer maintain the bridges so that surface are immediately disappear at near 100% porosity. We keep the pore density constant (10PPI) and find the analytical relationship between the porosity and surface area. Liu [17] provides a general mathematical relationship between the porosity and the specific surface area. The constants in the expression are determined experimentally for aluminum.

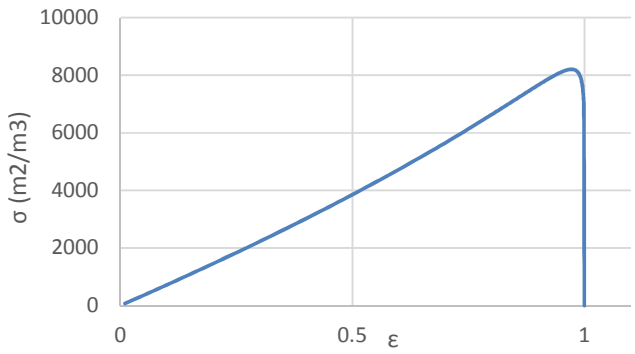

Fig. 5 Specific Surface Area v/s Porosity

In this work, aluminum porous heat sink is considered. The relationship is given by

$$\sigma = \frac{k_s}{d}[(1-\varepsilon)^{\frac{1}{2}} - (1-\varepsilon)](1-\varepsilon)^n] \quad (6)$$

Where, k_s= 281.8 and n=-0.4 for aluminum. Similar expressions are also observed in [18]. The expression and constants change depending on the material kind and the pore density. It is observed that initially as the porosity will go on increasing, the surface area will increase. This will continue up to a point after which there will be drastic drop in the surface area as the solid fraction reaches very small values (high values of porosity).

5) Dimensionless temperature difference

We have assumed an average heat transfer coefficient. However, the local heat transfer depends on the localized temperature difference between the inlet fluid and at the solid surface. In order to obtain the temperature distribution across the heat sink, a pin fin model is used. The governing differential equation is determined using the energy balance. The conduction is in the direction of height (Z direction). Assuming a porous medium as a solid-like pin-fin, the heat conduction through both the metal part and fluid part are considered. Hence the effective thermal conductivity is determined by,

$$k_{eff} = k_s(1-\varepsilon) + k_f\varepsilon \quad (7)$$

And,

$$-k_s A_{conds}\frac{dT_{fm}}{dx}\Big|_x - k_f A_{condf}\frac{dT_{fm}}{dx}\Big|_x$$

$$= -k_s A_{conds}\frac{dT_{fm}}{dx}\Big|_{x+dx} - k_f A_{condf}\frac{dT_{fm}}{dx}\Big|_{x+dx} + h_{fm}A_{conv}(T_{fm} - T_\infty) \quad (8)$$

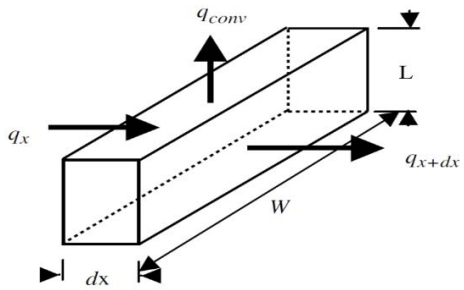

Fig. 6 Control Volume

Boomsma et al [6] use the same relationship for analyzing the heat sink performance. In this case analysis is carried out using the same model as [19]. The temperature for a single layer is assumed to be constant throughout the layer in the analysis. This assumption is supported by a detailed study and experimentation in [19] itself. The areas occupied by the solid and fluid phase are evaluated as a function of the porosity at any cross-section. After applying the boundary conditions and solving the differential equations, the following relationship is obtained.

$$\theta(X) = \frac{\cosh m_{fm}(1-X)}{\cosh m_{fm}} \tag{9}$$

Where, θ and X are the dimensionless temperature and the dimensionless length defined as

$$\theta = \frac{T_{fm}-T_\infty}{T_b-T_\infty} \quad \text{And} \quad X = \frac{x}{L} \tag{10}$$

Where m_{fm} is

$$m_{fm}^2 = \frac{h\sigma H^2}{k_s(1-\varepsilon)+k_f\varepsilon} \tag{10}$$

To obtain the weighted average temperature difference for the entire medium, the function is integrated over the entire length.

$$\theta_p = \int_0^1 \frac{\cosh m_{fm}(1-X)}{\cosh m_{fm}} dx = \frac{\tanh m_{fm}}{m_{fm}} \tag{12}$$

Multiplying this term with the temperature difference (between base and fluid) gives us the characteristic temperature difference of the entire pin fin. Variation of θ and θ_p with respect to the porosity is shown below. The temperature difference goes on reducing with increasing porosity. This implies that the average weighted temperature of the medium approaches that of the fluid, confirming dominance of convection heat transfer over conductions. Even if the temperature gradient in case of conduction is maximum as the porosity tends to 1, the cross-section area is considerably small causing the net heat flow to tend to zero.

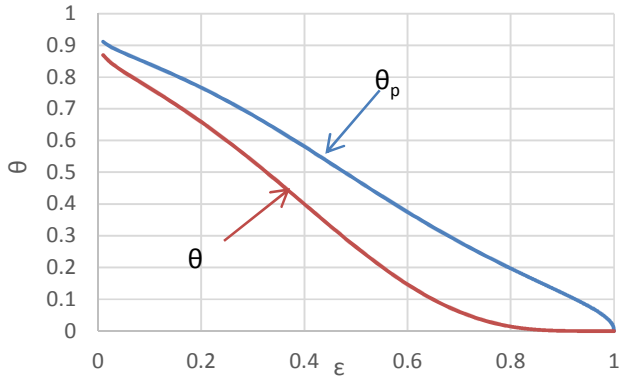

Fig. 7 θ and θ_p v/s porosity (θ at X=0.9)

6) Pumping work

The pumping work is determined from the pressure loss and the flow rate. The pressure loss can be found out by integrating the pressure gradient

$$\Delta P = \int -\frac{\partial P}{\partial x} dx \tag{13}$$

From Darcy's law as reported in [8],

$$-\frac{\partial P}{\partial x} = \frac{\mu}{K} U_{mod} \tag{14}$$

The pumping work then is found by

$$W_{pp} = \Delta P \times \dot{q} = \frac{\mu}{K} {U_{mod}}^2 \times L \times A_L \tag{15}$$

For the pressure gradient we have assumed that the inertia coefficient is small and doesn't dominate which holds true for small Reynolds number as discussed previously.

$$U_{in} = \frac{A_f}{A_t} U_{mod} \tag{16}$$

$$\frac{A_f}{A_t} = \frac{\frac{V_f}{L}}{\frac{V_t}{L}} = \frac{V_f}{V_t} = \varepsilon \tag{17}$$

Substituting from Eq. (16) and Eq. (17) in Eq. (15), we observe the following relation.

$$W_{pp} = \frac{\mu U_{in}^2 LA}{\varepsilon^2 K} \tag{18}$$

The work tends to infinity as the porosity tends to zero (maximum work to pump fluid through the medium). The work drops exponentially as the porosity increases. The overall magnitude is small in this case as the dimensions of the sample in question are small and the fluid considered is air.

7) Mass

The mass 'm' is found from the porosity, the envelop volume and the density of the metal (aluminum).

$$m = \rho_s(V_t - V_f) = \rho_s V_t(1-\varepsilon) \tag{19}$$

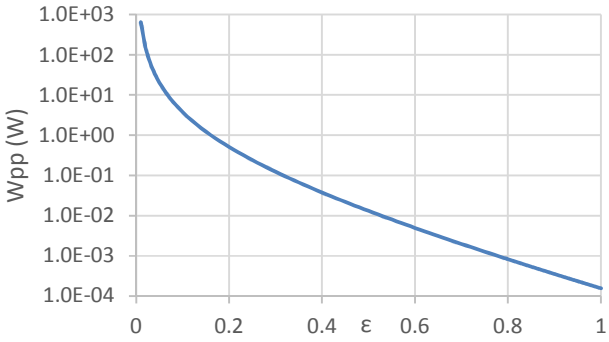

Fig. 8 Pumping work against v/s Porosity; heat sink dimensions 3cmx5cmx5cm, Fluid: air

8) Figure-of-Merit (FOM)

FOM is evaluated as

$$\mathrm{FOM} = \frac{Q}{W_{pp}} \times \frac{1}{(1-\varepsilon)} \tag{20}$$

Where,

$$Q = h \times A_s \times \theta_p \times \Delta T \tag{21}$$

ΔT here corresponds to the temperature difference between the base and the fluid flowing through the medium,

$$A_{sf} = \sigma \times V_t \tag{22}$$

The numerator of Eq. (20) takes into account the performance, whereas the denominator accounts for the cost. The objective is to maximize the FOM in order to achieve the maximum performance for the cost. Alternatively, other FOMs may be defined solely on the basis of cost

$$\mathrm{FOM}_{cost} = \frac{Q}{W_{pp}} \times \frac{1}{(1-\varepsilon)} \times \frac{1}{G\rho_s V_t} \tag{23}$$

Or with respect to weight,

$$\mathrm{FOM}_{weight} = \frac{Q}{W_{pp}} \times \frac{1}{(1-\varepsilon)} \times \frac{\rho_f}{\rho_s} \tag{24}$$

For comparison with the conventional heat sink, the COP is also computed.

$$\mathrm{COP} = \frac{Q}{W_{pp}} \tag{25}$$

To compare the equivalent FOM for conventional heat sinks, we can define

$$(1-\varepsilon') = \frac{n(H-B)Lt}{(H-B)LW} \tag{26}$$

RESULTS & DISCUSSION

The FOM and COP are plotted as a function of porosity in Figs. 9and 10. The FOM increases exponentially with porosity. Hence the porosity should be as large as possible in order to achieve the maximum FOM. The maximum value of porosity must be limited due to the manufacturing capability and the required mechanical strength. The exponential trend suggests that even a small increase in porosity will result in substantial increase in FOM. Note that the value of FOM is dominated by 1/(1-ε) and values of Q and W_{pp} have less impact. The above term causes the FOM to rise exponentially even if Q drops (as A_s and θ_p decrease).

Equivalent 1/(1-ε) values (from Eq. 26) for heat sinks made up of magnesium, aluminum and copper were calculated from the experimental data in [24] and plotted alongside the resultant FOM for porous aluminum in Fig, 9. It is observed that the conventional heat sinks have a better FOM at low porosity values, but at high porosity (>0.9) values FOM of the porous media will be much higher than that of the conventional heat sink.

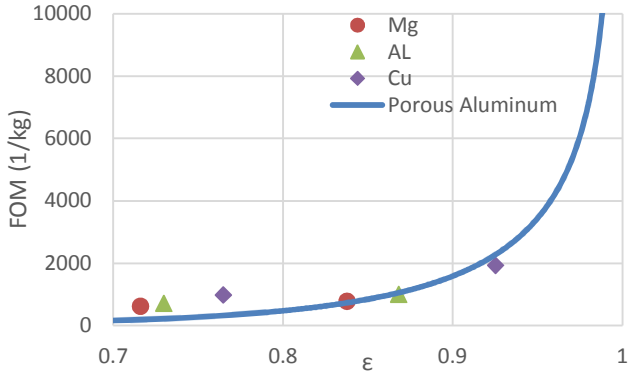

Fig. 9 FOM v/s Porosity for aluminum foam with air flow at 1m/s ,temperature difference of 20K between air and base plate; 3cmx5cmx5cm

For the COP, there exists an optimum porosity. The optimum porosity value is observed above 0.95 for this condition. In general, the optimum value will be different for different designs, it will be a function of material properties and geometric parameters. The impact of the above parameters on the COP was studied as well. It is observed from Fig.10, reducing the inlet velocities exponentially increases the maximum COP. Interestingly, the optimum value of the porosity decreases to lower values by decreasing the inlet velocity. The increasing trend in the maximum COP is due to the change in pumping work and is not substantially affected by the heat transfer coefficient. This is because at low Reynolds number heat transfer by conduction dominates over convection.

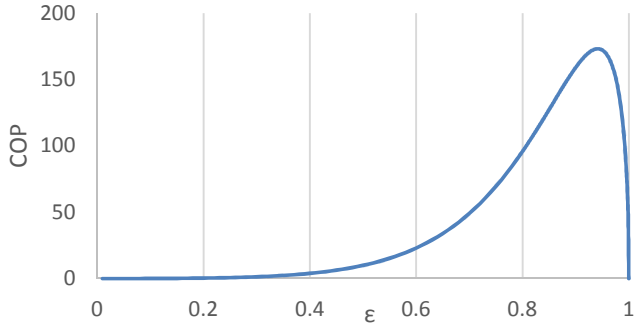

Fig. 10 COP v/s Porosity for aluminum foam and air flow at 0.5 m/s with a temperature difference of 20K between air and base plate; 3cmx5cmx5cm

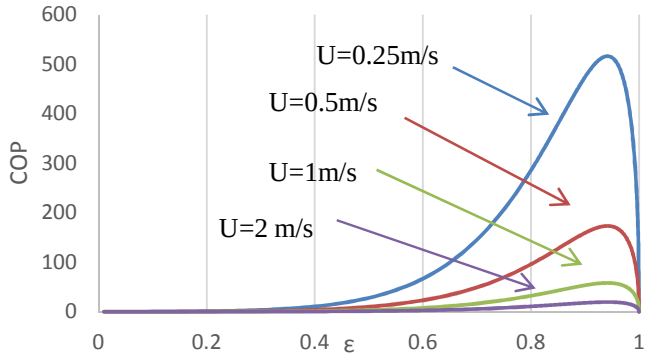

Fig. 11 Variation in inlet velocity temperature difference of 20K between air and base plate; 3cmx5cmx5cm

A similar trend observes with varying viscosity values. The COP increases by reducing viscosity values. This fact could be useful for a liquid selection in oil cooling applications. The COP is evaluated for different fluids by varying the respective fluid properties of density, viscosity and thermal conductivity. As we discussed earlier, the dependence of permeability on porosity may be exponential only for high porosity and might be linear for lower porosity. The results for COP were analyzed for these two different models of permeability, linear model and exponential one. The results for COP and FOM are shown in Fig. 13 and Fig.14, respectively.

These two curves are expected to provide the upper limit and the lower limit. The two curves overlapping each other at high porosity indicates that the COP can be predicted with better accuracy by a smaller bound. The large deviation in the COP in lower porosity region indicates the need for experimental validations

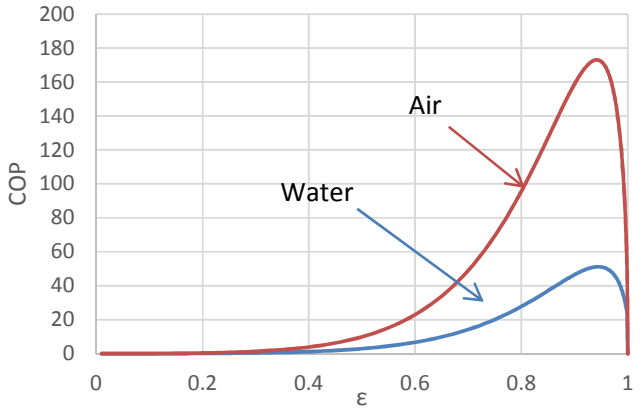

Fig. 12 COP for different fluids

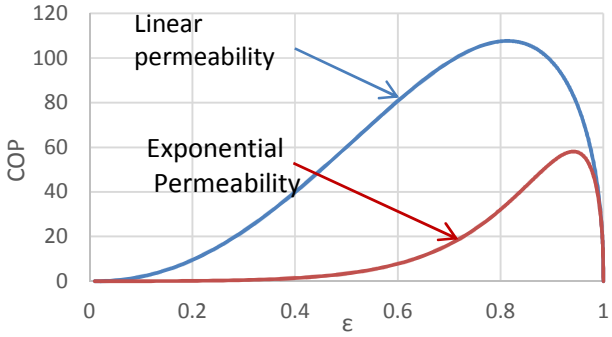

Fig. 13 Variation in COP with permeability

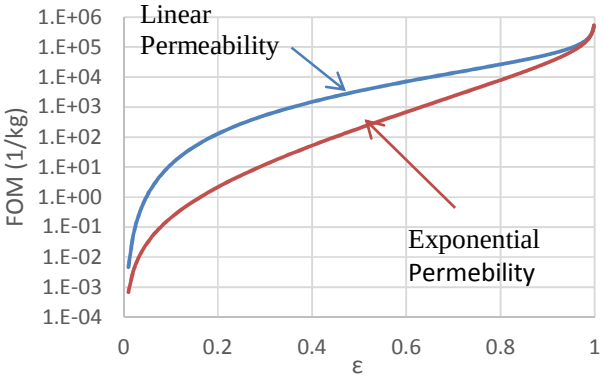

Fig. 14 Variation in FOM with permeability

CONCLUSION

The dependence of the performance and cost on the basic geometric parameters for porous medium heat sinks was studied. A figure-of-merit FOM is introduced to give a quantitative measure of the performance to cost ratio.

Porosity as large as possible is desirable to maximize FOM. An effort to even slightly, increase, the porosity may also result in a substantial increase in the FOM based on the exponential relationship. An optimum porosity is found for maximum COP. The optimum porosity depends on the pore density, envelop dimensions of the heat sink, unit cell type (bcc in this case) in addition to the material properties of foam and fluid. Additionally, conventional finned heat sinks have a better performance than low porosity foam heat sink for the same volume fraction, but the performance of the foam heat sink will be much higher than that of the conventional heat sink at high porosity.

The impact of the pore density and the inlet velocity was also analyzed. The COP increases with either reducing the inlet velocity or increasing the fluid density. The material properties, i.e. the fluid viscosity and the thermal conductivity of both the fluid and foam are also involved.

The relationship between the permeability and the porosity showed two trend lines corresponding to the linear model and the exponential model. Further experimentation may verify the practical values in between the two curves, which would provide a bound for FOM and COP vs. porosity respectively.

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