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Procedia Engineering 56 (2013) 337 – 343
Procedia Engineering
www.elsevier.com/locate/procedia

5th BSME International Conference on Thermal Engineering

Cooling of microchannel heat sinks with gaseous coolants

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Abstract

In this paper, the optimization of the cooling performance of a rectangular microchannel heat sink is investigated with four different gaseous coolants; air, ammonia gas, dichlorodifluoromethane (R-12) and chlorofluoromethane (R-22). A systematic robust thermal resistance model together with a methodical pumping power calculation is used to formulate the objective functions, the thermal resistance and pumping power. The non-dominated sorting genetic algorithm (NSGA-II), a multi-objective algorithm, is applied in the optimization procedure. The optimized thermal resistances obtained are 0.178, 0.14, 0.08 and 0.133^{o}K/W for the pumping powers of 6.4, 4, 22.4 and 16.5 W for air, ammonia gas, R-12 and R-22, respectively. These results show that among all the gaseous coolants investigated in the current study, ammonia gas exhibited balanced thermal and hydrodynamic performances. Due to the Montreal Protocol, the coolant R-12 is no longer produced while R-22 will eventually be phased out. The results from ammonia provide a strong motivation to conduct more investigations on the potential usage of this gaseous coolant in the electronic cooling industry.

Keywords: Microcahnnel; gaseous coolants; thermal resistance; optimization.

Nomenclature

A_{eff}	effective area available for heat transfer (m^2)
A_{hs}	heat sink cross sectional area (m^2)
A_t	induction tubes cross sectional area (m^2)
Cp	specific heat (J/kg.K)
D_h	hydraulic diameter (m)
f	friction factor
H	heat sink height (m)
H_c	channel height (m)
G	volumetric flow rate (l/s)
h_{av}	average heat transfer coefficient (W/m^2.K)
k	thermal conductivity (W/m.K)
L	Heat sink length (m)
Nu	Nusselt number
n	number of microchannels
Δp	pressure drop (mbar)
Δp_{tu}	tube pressure drop (mbar)
q	heat flux (W/m^2)

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Selection and peer review under responsibility of the Bangladesh Society of Mechanical Engineers
doi:10.1016/j.proeng.2013.03.128

Re	Reynolds number
R	thermal resistance (K/W)
W	heat sink width (m)
w_c	channel width (m)
w_w	wall (fin) width (m)
V_{mf}	velocity inside the channels (m/s)
V_{mt}	velocity inside the tubes (m/s)
P_p	pumping power (W)
Greek symbols	
α	channel aspect ratio
β	fin spacing ratio
ρ	density (kg/m^3)
μ	dynamic viscosity (kg/m.s)
□	fin efficiency
Subscripts	
hs	Heat sink
tu,t	tube
tot	total
f	fluid (coolant)
w	wall

1. Introduction

With the enormous development in the capabilities of the microelectronic mechanical systems (MEMS), the use of the microchannel heat sink has acquired great importance because it provides a high heat dissipation rate, compatibility with the small allowable space and ultimately a low manufacturing cost. Liquid coolants have been extensively used with the microchannel heat sinks for their high capabilities of absorbing heat. However, the associated issues of the high pumping power demands [1,2], leakage [3], and passage clogging [4] in the case of the nanofluids usage, have not been fully addressed. Air was used as an alternative coolant in many microchannel heat sink systems [5-7]. However, its poor heat removal capabilities [8] have limited its applications. In this study, the overall performance of the microchannel heat sink using several gaseous coolants is investigated. The coolants are air, ammonia gas, R-12 and R-22. The search for an alternative coolant to overcome the difficulties of the liquid coolants was the motivation to consider these gaseous coolants. An optimization scheme which incorporates the thermal resistance model as an objective functions formulator and the non-dominated sorting genetic algorithm (NSGA-II) as an optimization performer is employed to investigate the overall performance of the considered system under these different gaseous coolants.

2. Mathematical model

Figure 1 illustrates the schematic drawing of the rectangular microchannel heat sink under investigation in the current study.

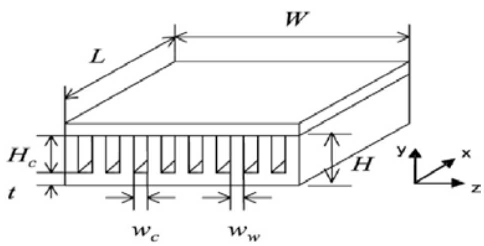

Fig. 1. Schematic drawing of the microchannel heat sink.

The microchannel heat sink comprised of n number of microchannels attached to each other with an adiabatic covering plate bonded on top to close the microchannels. Induction tubes are used to transport the coolant to and from the microchannels to avoid the bypass flow and to provide a sufficient length for the flow to reach the fully develop status. The above mentioned microchannel system performance is evaluated using a systematic thermal resistance model and a methodical pumping power calculation. The approach offered by Wen and Choo [9] and Kleiner et al. [10] to evaluate the total thermal resistance and pumping power are followed and modified where needed. Kleiner et al. [10] model was used because it was experimentally verified and it showed superior thermal and hydrodynamic performance compared to the previous conventional air-cooled microchannel heat sink systems.

2.1. Thermal performance model

The thermal performance of a microchannel heat sink is evaluated through its total thermal resistance. The total thermal resistance of any heat sink is described as the ratio of the temperature difference between the maximum temperature of the substrate and the coolant inlet temperature, to the heat flux. The maximum temperature is normally located at the end of the microchannels and the heat flux is assumed to be uniformly applied to the back side of the microchannel heat sink. The total thermal resistance of the heat sink is given by:

$$R_{tot} = \frac{T_{surf,\max} - T_{in}}{q} \tag{1}$$

where $T_{surf,max}$ and T_{in} are the highest and the inlet temperatures of the substrate and the coolant, respectively. Eqn. (1) can be expressed in terms of the dominated components of the total thermal resistance,

$$R_{tot} = R_{conv} + R_{capa} \tag{2}$$

where R_{conv} and R_{capa} are the convective and capacitive thermal resistances, respectively. The first term in Eqn. (2) can be expressed as,

$$R_{conv} = \frac{1}{h_{av} A_{eff}} \tag{3}$$

The effective area for heat transfer in Eqn. (3) can be defined as,

$$A_{eff} = nL(w_c + 2\eta H_c) \tag{4}$$

The number of the microchannels and the fin efficiency are calculated according to the following equations,

$$n = \frac{W}{w_w + w_c} \tag{5}$$

$$\eta = \frac{\tanh(mH_c)}{mH_c} \tag{6}$$

where m is the fin parameter given by,

$$m = \sqrt{\frac{2h_{av}}{k_w w_w}} \tag{7}$$

where k_w is the thermal conductivity of the heat sink material which is assumed to be made of aluminum. The second term in Eqn. (2) is the capacitive thermal resistance which can be expressed as,

$$R_{capa} = \frac{1}{\rho_f cp_f G} \tag{8}$$

$$G = nH_c w_c V_{mf} \tag{9}$$

The convective and capacitive thermal resistance expressions are further simplified using the channel aspect ratio (α) and the fin spacing ratio (β) along with several other auxiliary equations as follows,

$$\alpha = \frac{H_c}{w_c} \tag{10}$$

$$\beta = \frac{w_w}{w_c} \tag{11}$$

$$D_h = \frac{2}{1+\alpha} H_c \tag{12}$$

$$\mathrm{Re} = \frac{2\rho_f G}{\mu_f n H_c} \frac{\alpha}{1+\alpha} \tag{13}$$

$$Nu = \frac{h_{av} 2 H_c}{k_f (\alpha + 1)} \tag{14}$$

Substituting Eqns. (10-14) into Eqns. (3, 8) results in the final expression of the total thermal résistance as,

$$R_{tot} = \frac{L}{cp_f \mu_f} \frac{2}{\mathrm{Re}} \frac{1+\beta}{1+\alpha} + \frac{1}{h_{av}} \frac{1+\beta}{1+2\alpha\eta} \tag{15}$$

The average heat transfer coefficient appearing in Eqn. (15) is evaluated using the Nusselt number correlation given by Kim and Kim [11] for a laminar fully developed flow,

$$Nu = 2.253 + 8.164 (\frac{\alpha}{1+\alpha})^{1.5} \tag{16}$$

2.2. Hydrodynamic performance model

In the current study, the hydrodynamic performance of the microchannel heat sink is assessed using a pressure drop calculation and the associated required pumping power. The methodology offered by Kleiner et al. [10] where induction tubes are employed to transfer the coolant is used and modified. The total pressure drop is given by,

$$\Delta p_{tot} = \Delta p_{hs} + \Delta p_{tu} \tag{17}$$

where Δp_{tot} , Δp_{hs} and Δp_t are the total, heat sink and tubes pressure drops, respectively. The pressure drop [10] inside the microchannel is modified and the final expression for the total pressure drop is

$$\Delta p_{tot} = f_{hs} \frac{L}{D_h} \rho_f \frac{V_{mf}^2}{2} \text{(20)} + (1.79 - 2.32(\frac{1}{1+\beta}) + 0.53(\frac{1}{1+\beta})^2) \rho_f \frac{V_{mf}^2}{2} + f_{t1} \frac{L_{t1}}{D_{tu}} \rho_f \frac{V_{mt}^2}{2} + f_{t2} \frac{L_{t2}}{D_{tu}} \rho_f \frac{V_{mt}^2}{2} + 0.42 \rho_f \frac{V_{mt}^2}{2} + (1 - \frac{A_t^2}{A_{hs}^2}) \rho_f \frac{V_{mt}^2}{2} + 0.42 (1 - \frac{A_t^2}{A_{hs}^2}) \rho_f \frac{V_{mt}^2}{2} + \rho_f \frac{V_{mt}^2}{2} \tag{18}$$

Finally the pumping power can be evaluated through the following equation,

$$Pp = \Delta p_{tot} \times G \tag{19}$$

The friction factor that appears in the total pressure drop equation is evaluated using the correlation provided by Copeland [12],

$$f\,\mathrm{Re} = [(3.2(\frac{\mathrm{Re} D_h}{L})^{0.52})^2 + (4.7 + 19.64 B)^2]^{0.5} \tag{20}$$

where B is a geometrical parameter given by ,

$$B = \frac{(\frac{1}{\alpha})^2 + 1}{(\frac{1}{\alpha} + 1)^2} \tag{21}$$

3. Optimization procedure

In this paper, two design variables are selected; the channel aspect ratio (α) and the fin spacing ratio (β). The limits of these design variables are taken from Kleiner et al. [10] with $23.742 < \alpha < 59.808$ and $0.254 < \beta < 0.5$. The system is treated as a multi-objective function with the thermal resistance (Eqn. 15) and the pumping power (Eqn. 19) being the first and the second objectives. These objective functions are optimized using the NSGA-II.

3.1. The applied algorithm

The objective functions considered in the current study posses a competing nature in which the increase in one results in a decrease in the other. The NSGA-II known for its strong capability in optimizing multi-objective functions [13], is used to perform the optimization process. The methodology applied to perform the optimization and hence to generate the Pareto optimal front is fully described in Ahmed et al. [14].

4. Results and discussion

In this section, the overall performance of the considered system is investigated for four different coolants, air, ammonia gas, R-12 and R-22. For a constant volumetric flow rate (G = 5.3 l/s) and under the same operating conditions (Table 1), the performance of the system with air as a coolant is used as a benchmark for comparison with other coolant performances as it can be seen in Figs. 2 and 3 for ammonia gas, and R-12 and R-22, respectively.

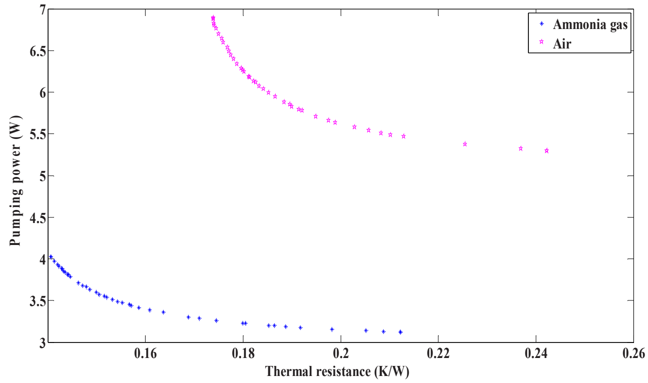

Fig. 2. The overall performance of the considered system with air vs. ammonia gas.

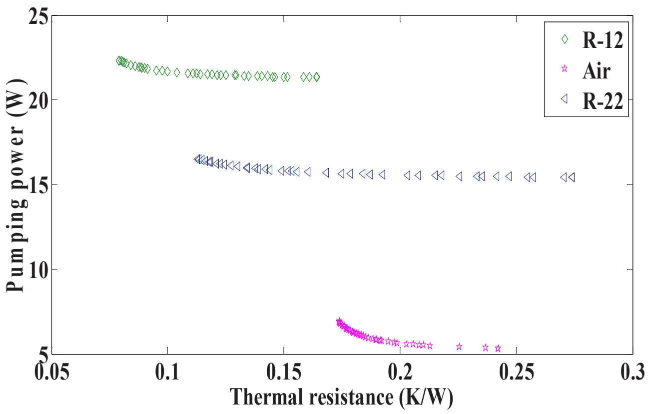

Fig. 3. The overall performance of the considered system with air vs. R-12 and R-22.

It can clearly be seen from Fig. 2 that ammonia gas behaves better than air as a cooling fluid. It provided a significant reduction in the total thermal resistance and pumping power for the same operating conditions. This behavior is attributed to the excellent thermophysical properties of ammonia. The specific heat capacity and the thermal conductivity of ammonia are better than air in the expected range of operating temperature. As for air vs. R-12 and R-22 (Fig. 3), the latter two provided much lower thermal resistance than air but at the expense of the required pumping power. The Montreal Protocol has stopped any industrial application of R-12 due to its significant contributions to the Global Warming Phenomena (GWP) and Ozone Depleting Potential (ODP).

Table 1 Assumed parameters and thermophysical properties at 27 oC.

Parameters	Values
Heat sink lateral dimensions, (W×L) (cm^2)	5×5
Channel height, H_c (cm)	2.5
Induction tubes length, L_t (m)	0.5
Induction tubes diameter, D_t (mm)	19
Thermal conductivity of aluminum, k_w(W/m.K)	238

R-22 lowers the thermal resistance significantly compared to air but with a very high pumping power requirement too. It can be seen that ammonia gas showed a reasonable performance in both aspects, thermal and hydrodynamic, compared to the other coolants considered in this research. Known for its environmental friendly behavior and not requiring sophisticated machinery to be produced, ammonia gas can be a very suitable alternative for air and water generally used in the heat sinks. The optimized results of the current study are listed in Table 2.

Table 2. Optimized results of the current study.

Parameters	Air	Ammonia gas	R-12	R-22
Thermal resistance, R (K/W)	0.178	0.14	0.08	0.113
Pumping power, Pp (W)	6.4	4	22.4	16.5
Channel aspect ratio, α	51.799	59.692	58.912	59.785
Fin spacing ratio, β	0.272	0.267	0.254	0.254

5. Conclusions

In this research, the overall performance of a rectangular microchannel heat sink is examined for four different coolants, air, ammonia gas, R-12 and R-22. Ammonia gas showed balance thermal and hydrodynamic performances under the same operating conditions compared to the other coolants investigated in this study. R-12 and R-22 provided a lower thermal resistance but they cannot be considered due to the high pumping power demands and their environmental issues. The results obtained provide motivation for further efforts to be spent on exploration of other coolant performances in the area of microchannel heat sinks industry.

Acknowledgements

The authors would like to acknowledge the supports from Universiti Teknologi Malaysia (UTM) under UTM Razak School Grant Vote No. 4B026 for providing the fund and facilities throughout the course of this research.

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