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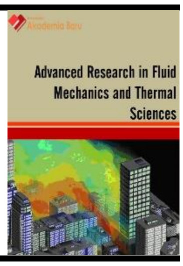

Experimental Correlation for Flow-boiling Heat Transfer in a Micro-gap Evaporator with Internal Micro-fins

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ABSTRACT

Micro-gap evaporators are prospective candidates for cooling of micro-electronic devices. Heat transfer with phase change in a micro-gap heat sink is efficient for high heat flux applications. Micro-fins induce turbulence escalating two-phase heat transfer rate in the micro-gap evaporator. However, no correlation has been developed so far for fin-induced pseudo-turbulence and heat transfer rate in micro-gap heat sink. The scope of this paper is to develop a correlation for dimensionless heat transfer rate in a micro-finned micro-gap heat sink for refrigerant 134a. The correlation signifies the effect of evaporation rate on thermal resistance of the heat sink. For this purpose, a micro-gap evaporator was fabricated with a gap height adjusted to accommodate 48 rectangular shaped micro-fins. A heater at the bottom of the heat sink provides variable heat flux. The evaporator is installed in a test section along with a heat pump in the outer circuit. The frequency of the compressor is varied using a variable frequency drive. A pre-heater is used to vary inlet temperature of refrigerant. In the developed correlation, dimensionless heat flux is function of calculated vapor Reynolds number and Biot number. Results calculated from the correlation show maximum 2.7% error in comparison to actual results.

1. Introduction

In modern world, miniature electronic devices have immense popularity because of their light weight and carrying facility. Beside civil applications, integrated devices are also used for military and research purposes. These devices are densely packed integrated circuits (ICs), which produce high heat flux during operation. Heat flux dissipation from ICs has already attained 1000 Wcm^{-2} in the year 2008 and still it is in progress [1]. Thermal management of these components is a major challenge for researchers. Moreover, various micro-opto-electronic devices such as micro-solar cells, image

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sensors, laser diodes etc. are widely used in engineering, which necessitates extensive cooling facility. Two-phase micro-scale cooling is a great choice for this purpose.

Many well-known contributions have been put in the timeline of over thirty years research on micro-scale fluid flow and heat transfer. Many numerical and experimental investigations were accomplished by researchers to design an effective cooling system using microchannel heat sink for tiny electronic and electromechanical devices, invented by Tuckerman and Peace [2]. However, two-phase flow in microchannels also exhibits some disadvantages such as higher pressure drop, flow boiling instabilities, flow reversal, exceeding critical heat flux etc. Recently, micro-gap heat sinks have been found potential to reduce prevailing two-phase problems [3]. In micro-gaps, walls of microchannels, which act as fins, are eliminated and coolant flows through the gap. Alam *et al.*, [4] showed that micro-gap heat sinks reduce flow boiling instabilities and generate more uniform surface temperature than micro-channels.

Ahmed *et al.*, [5] introduced the concept of micro-finned micro-gap evaporator to enhance heat transfer rate by both extending surface area and inducing pseudo-turbulence in the flow field. In another publication, Ahmed *et al.*, [6] showed that rectangular fins are superior over triangular fins for heat transfer enhancement. Numerical simulation for parametric study on micro-fins and turbulence generation are featured in other papers of these authors [7,8]. Most recently, they presented experimental results on micro-finned micro-gap evaporators for both transient and steady-state conditions [9].

Many empirical correlations have been proposed to model heat transfer in microchannels [10-16]. However, none of these correlations are suitable for micro-finned micro-gap heat sinks. Moreover, estimated correlations are independent of rate of evaporation, which is a major drawback since it has significant effect on two-phase heat transfer rate. Hence, an adaptive correlation based on evaporation related parameters should be predicted from experimental results.

In this paper, a correlation for dimensionless heat transfer rate in a micro-finned micro-gap heat sink for refrigerant 134a has been developed. Heat transfer rate is a function of Reynolds number calculated from vapour velocity at the outlet of the evaporator and Biot number. The Vapour Reynolds number signifies the effect of evaporation rate on heat transfer. The proposed model has been tested for validation purpose.

2. Experimental Setup and Instrumentation

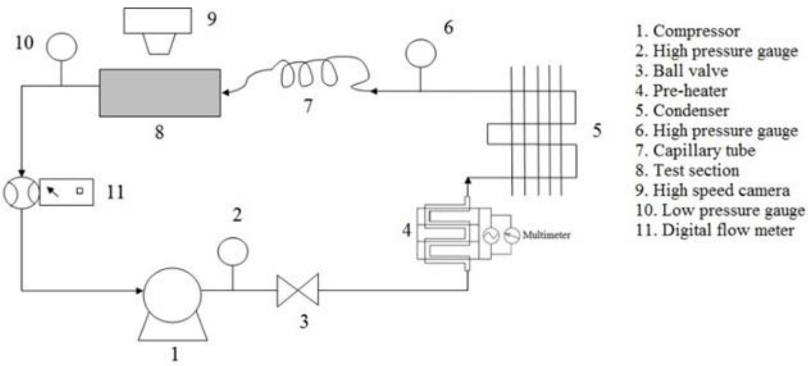
The test setup consists of a heat pump and a test section. R-134a has been selected as working fluid.

2.1 Heat Pump

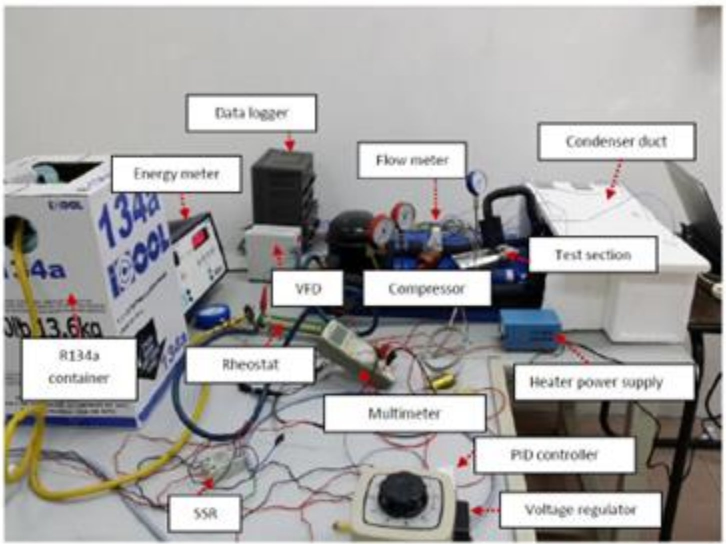
The major components of the heat pump are hermetically sealed low back pressure compressor, condenser with a cooling fan, evaporator and capillary tube. The test section is an acrylic structure with inlet and outlet manifolds, which accommodates the heater and the evaporator. All other components of the heat pump are installed at the outer circuit of the test section. Proper controlling systems are provided to control various input parameters. A coil heater with PID controller controls refrigerant temperature at the outlet of the compressor. Temperatures are recorded using a data logger. Flow rate can be regulated either by controlling compressor frequency using a variable frequency drive (VFD) or operating a ball valve installed at the outlet of the compressor. Volumetric flow rate of refrigerant vapour at the outlet of the test section is acquired using a separate data acquisition system and plotted directly in computer by TracerDAQ software. Schematic diagram of the test section with main components and photograph of the test rig are shown in Figure 1(a) and 1(b), respectively.

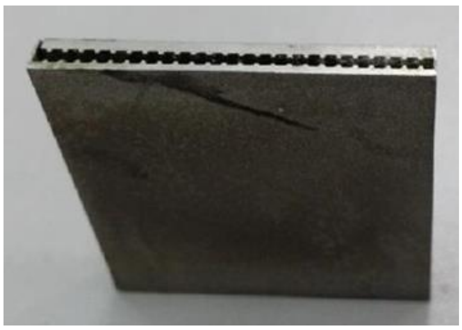
2.2 Test Section

The heat sink and a peltier heater are installed inside of the test section. A micro-gap with 48 internal micro-fins of rectangular shape has been fabricated by wire-cut EDM on aluminium substrate. Fabricated evaporator has a footprint area of 30 mm × 30 mm. Lower and upper base thicknesses are 2 mm and 1 mm respectively and the gap height is 1 mm. Each micro-fin has height and width of 0.3 mm. Fabricated micro-finned micro-gap has been shown in Figure 1(c). A 12 V DC power supply is provided to the heater, which is controlled by a rheostat. Heater power is measured using a multimeter.



(a)





(b) (c)

Fig. 1. (a) Schematic diagram of the test rig (main components) (b) Experimental setup (main components and accessories) (c) Fabricated micro finned micro gap

3. Data Reduction

After performing the tests, heat transfer and fluid flow parameters are calculated from experimental data using relevant formulas. Power supply to the heater is calculated from following equation

$$Q = I \times V \tag{1}$$

Here Q is the heater power, I is the current and V is the voltage applied across the heater terminals. Hence, heat flux applied at the bottom of the heat sink

$$q_{in} = \frac{Q}{A} \tag{2}$$

Here A = footprint area of the heat sink = 30 mm × 30 mm. Wall heat flux

$$q_{eff} = q - q_{loss} \tag{3}$$

Dimensionless numbers are calculated from the following formulas

Biot number

$$Bi = \frac{hD_h}{k_s} \tag{4}$$

Vapour Reynolds number

$$Re_v = \frac{\rho_v \dot{V}_v L}{A_{cs}\mu_v} \tag{5}$$

where L is the characteristics length of the channel, k_s is the thermal conductivity of channel material and $\dot{V}_v$ is the volumetric flow rate of vapour.

4. Results and Discussion

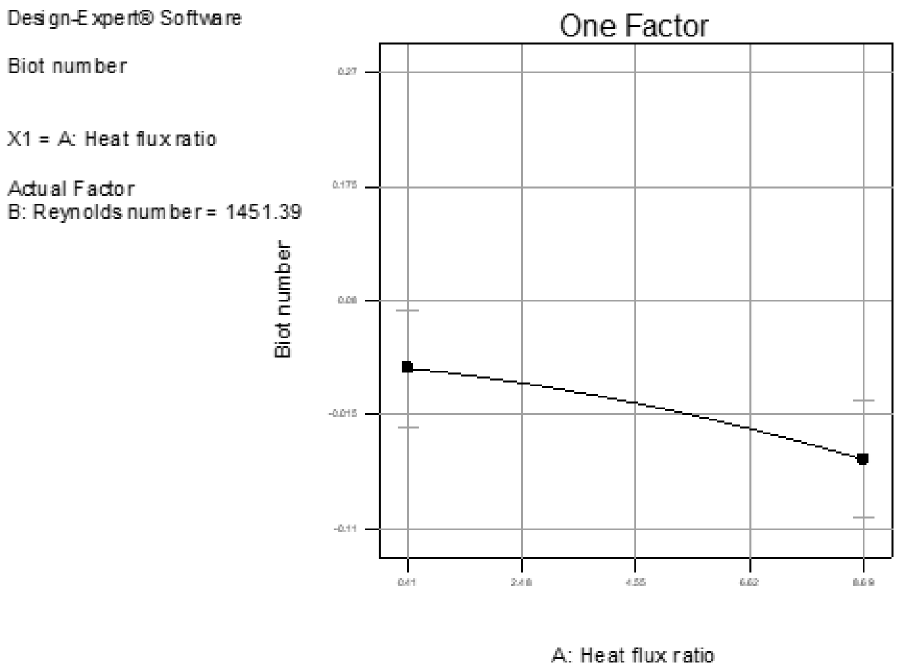
4.1 Empirical Modeling

An empirical model for evaporative heat transfer in micro-finned micro-gap heat sink has been developed, which correlates dimensionless heat flux $\left(\bar{q} = q^{"}/q_{in}\right)$ with Biot number $(Bi = hD_h/k_s)$ and Reynolds number calculated from vapour velocity at the outlet of the heat sink $\left(Re_v = \rho_v \dot{V}_v L/A_{cs}\mu_v\right)$. The correlation has been developed from experimental data using Response surface methodology (RSM). Design expert software has been used for this purpose.

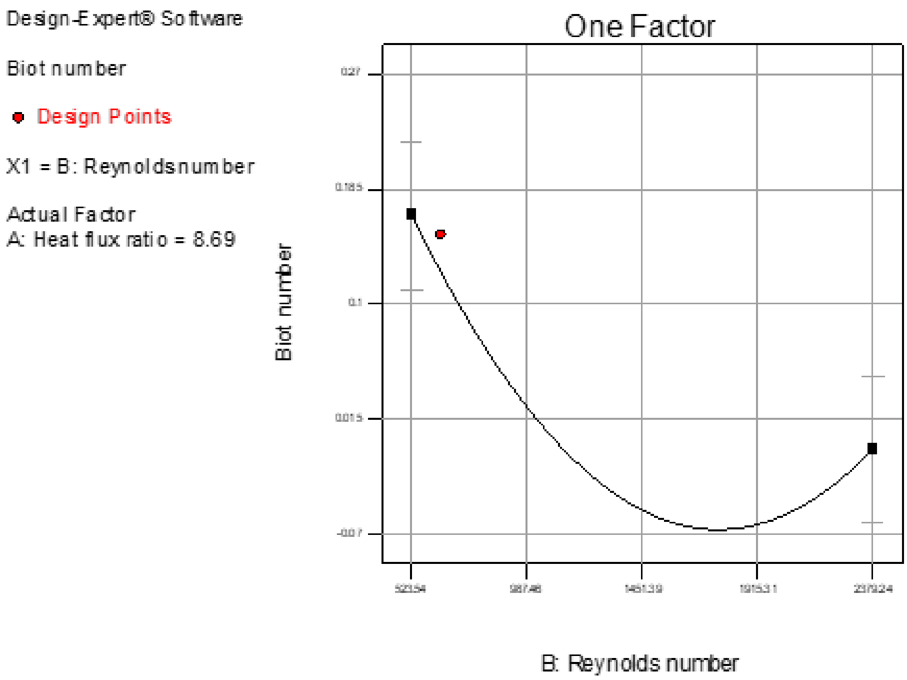
In the developed correlation, Re_v illustrates the significance of pumping power and evaporation rate on convective thermal resistance of the heat sink, as conductive thermal resistance is fixed. The proposed correlation has been developed for the range of Re_v = [523.5, 2379.24]. Hence, it can be predicted that the vapour flow is from laminar to transition flow region. Effect of thermal resistance can be estimated from Biot number (Bi).

One factor plots, provided in Figure 2, illustrate the relationship of vapour Reynolds number and dimensionless heat flux with Biot number. It shows that Biot number decreases with increasing heat flux. On the other hand, for $\bar{q} = 8.69$, Biot number decreases for the range of Re_v = 523.5 - 1500 and then elevates for further vapour Reynolds number increment.

Design summary of the experiment is provided in Table 1. Model summary statistics, provided in Table 2, showed that quadratic and cubic models are suggested for the given problem. A quadratic model has been selected and a simplified model is proposed by reducing some terms.



(a)



(b)

Fig. 2. One factor plots – (a) Biot number (*Bi*) vs. dimensionless heat flux ($\overline{q}$) (Re_v = 1451.39) and (b) Biot number vs. vapour Reynolds number (Re_v) ($\overline{q}$ = 8.69).

Table 1
Design summary for empirical modelling from experimental data (Study type: Response surface, Initial design: Central composite, Design model : Quadratic)

Factor	Name	Units	Type	Low actual	High actual	Mean	Std. Dev.
A	Heat flux ratio		Numeric	0.41	8.69	4.29	2.712
B	Reynolds number		Numeric	523.54	2379.24	1176.81	539.981

Response	Name	Units	Obs	Analysis	Min	Max	Mean	Std. Dev.
Y1	Biot number		13	Polynomial	0.012	0.261	0.075	0.079

Table 2
Model summary statistics for empirical modelling from experimental data.

Source	Std. Dev.	R^2	Adjusted R^2	Predicted R^2	PRESS	
Linear	0.061	0.5486	0.4583	0.0571	0.077	
2FI	0.063	0.5637	0.4182	-1.6168	0.21	
Quadratic	0.024	0.9509	0.9158	0.1490	0.070	suggested
Cubic	5.683×10^{-3}	0.9992	0.9968	-2.9340	0.32	suggested

Model coefficients have been estimated from least-square method, which are shown in Table 3. The coefficients are chosen from 95% confidence intervals.

Table 3
Coefficient estimation for empirical modelling from experimental data.

Factor	Coefficient estimate	Standard error	95% CI Low	95% CI High	VIF
Intercept	-5.464E-003	0.017	-0.044	0.033	
A-Heat flux ratio	-0.038	0.014	-0.071	-5.552E-003	1.06
B-Reynolds number	-0.086	0.017	-0.13	-0.047	1.25
A2	-8.920E-003	0.031	-0.080	0.062	1.27
B2	0.13	0.026	0.072	0.19	1.14

CI=confidence interval
VIF=variation inflation factor

Finally, developed correlation in terms of actual factors is the following

$$Bi = 0.48248 - 4.45846 \times 10^{-3}\bar{q} - 5.36516 \times 10^{-4}Re_v - 5.20412 \times 10^{-4}\bar{q}^2 + 1.52769 \times 10^{-7}Re_v^2 \tag{6}$$

Eq. (6) is solved for dimensionless heat flux $\left(\bar{q} = q^{"}/q_{in}\right)$, which gives the following expression:

$$\bar{q} = -\frac{0.22 \mp \sqrt{0.00053Re_v^2 - 0.03747Re_v + 0.23323 - 0.38127Bi}}{0.19063} \tag{7}$$

4.2 Model Adequacy Test

Figure 3 shows a good agreement between predicted value from proposed correlation and actual experimental values.

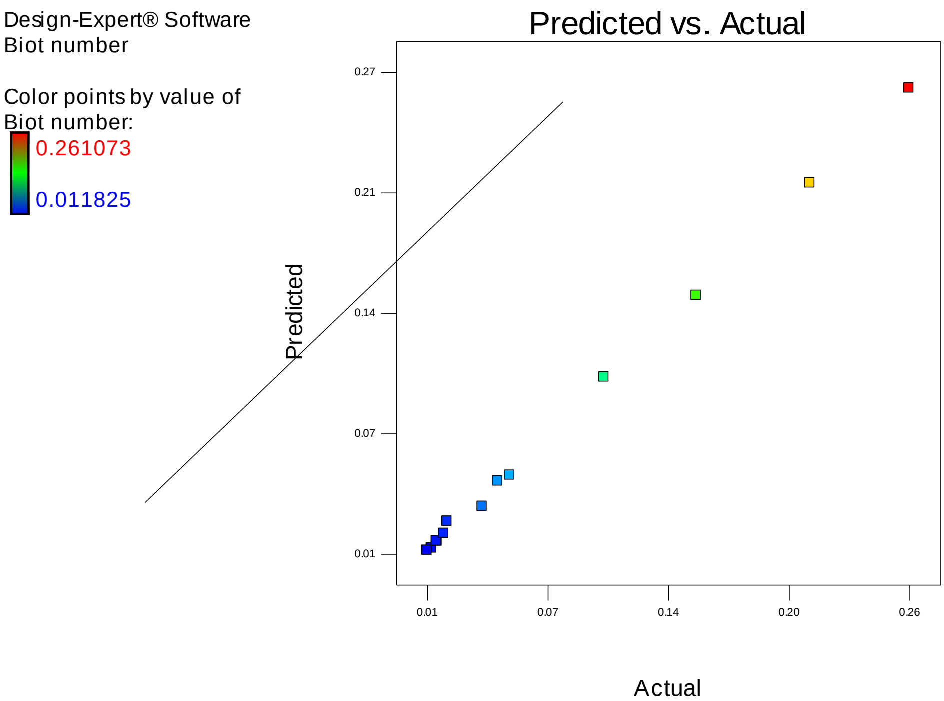


Fig.3. Plot of predicted values against actual experimental values

4.3 Model Validation

From Table 4, it is observed that the highest deviation of model predicted value from actual Biot number is 2.7% for $\bar{q} = 5.62$ and Re_v = 2273.43. Hence, it can be estimated that the proposed model is valid.

Table 4
Percentage of error for empirical model

Heat flux ratio ($\bar{q}$)	Vapour Reynolds number (Re_v)	Prediction (Biot number)	Actual (Biot number)	Prediction error (%)
3.52	945.32	0.292068	0.288	1.4
5.62	2273.43	0.27007	0.263	2.7
7.2	675.77	0.285001	0.28	1.8
6.17	2253.45	0.265936	0.261	1.9
2.69	851.89	0.320036	0.318	0.7

5. Conclusion

This paper describes the methodology of developing a correlation for heat transfer rate in rectangular finned micro-gap evaporator. The following concluding remarks can be drawn

i. Heat transfer rate in a micro-gap evaporator decreases with the increment evaporation rate as well as vapour Reynolds number.
ii. The model predicted for Biot number as a function of dimensionless heat flux and vapor Reynolds number signifies the effect of evaporation rate on thermal resistance of the heat sink.

iii. The model is validated as the highest error rate is only 2.7% between predicted and actual value of Biot number.

In future, a critical heat flux (CHF) can be achieved by using a high power heater. Effect of vapour Reynolds number on critical heat flux (CHF) condition can be studied and a mathematical model can be established.

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