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Influence of system pressure on the boiling heat transfer coefficient in a closed two-phase thermosyphon loop ☆

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Abstract

In recent years heat generation from electronic components has been rapidly increasing. Natural and forced convection air-cooled heat sinks are not well suited for cooling the components generating the highest heat fluxes. Other methods are therefore sought. Recently the use of thermosyphons has attracted attention as a simple and effective cooling system for high heat flux components. In a thermosyphon system the heat is absorbed during boiling and transferred as heat of vaporization from the hot part to the cold part of the system, with relatively small temperature differences.

The setup used in this study consists of a thermosyphon loop, including evaporator, condenser, downcomer and riser. The loop has three evaporators, connected in parallel, made from small blocks of copper (10 × 20 × 15) mm in which five vertical channels with diameters 1.5 mm and length 15 mm were drilled. Experimental results in terms of heat transfer coefficients at different system pressures and heat inputs are presented and compared to predictions of correlations from the literature. In all tests Isobutane is used as working fluid. © 2002 Éditions scientifiques et médicales Elsevier SAS. All rights reserved.

Keywords: Thermosyphon; Influence of pressure; Heat transfer; Two-phase; Electronic cooling

1. Introduction

Due to the rapid advancements within the electronic industry, the thermal management for electronic components has become an important and serious issue. As an example, the minimum feature size in microprocessors has been reduced from 0.35 µm in 1990 to 0.18 µm in 2000 [1]. The decrease in size and increase in performance, results in a rapid increase in heat fluxes. Natural and forced convection air cooling is limited to dissipate low or moderate heat fluxes and may not be sufficient in future electronics.

Among several methods for thermal management of high heat fluxes, thermosyphon cooling is one of the most effective and reliable heat removal techniques. A closed two-phase thermosyphon is a device for heat transmission from a hot part of the thermosyphon (evaporator) to a cold part (condenser). The thermosyphon may be designed as a thermosyphon pipe consisting of a straight tube in vertical or an inclined position, which is filled with a certain amount of refrigerant. Heat is transferred as heat of vaporization from the evaporator to the condenser inducing counter current two-phase flow in between the two. A thermosyphon loop consists of an evaporator and a condenser connected by two tubes, a riser and a downcomer. With the aim of obtaining a small temperature difference across the thermosyphon, the evaporator and condenser parts of the loop may be designed with channel geometries enhancing heat transfer in boiling and condensation. Such a device is here called an advanced thermosyphon loop. The advantages compared to the thermosyphon pipe, are the possibility of using narrower channels in connecting tubes, higher heat transfer coefficients and flexible design of the evaporator for close contact between the component and refrigerant channels. In either case, no mechanical devices are necessary for the circulation. The heat transfer performance of the system depends on the design of the thermosyphon system, and the choice of working fluid in the system (Palm and Khodabandeh [2]). The temperature difference between the evaporator wall and the refrigerant in the evaporator is the

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Nomenclature

A	area of heat transfer surface	m^2
C_p	specific heat	$J{\cdot}kg^{-1}{\cdot}K^{-1}$
d	channel inside diameter	m
E	enhancement factor	
G	mass flux	$kg{\cdot}m^{-2}{\cdot}s^{-1}$
h	heat transfer coefficient	$W{\cdot}m^{-2}{\cdot}K^{-1}$
h_{fg}	latent heat of vaporization	$kJ{\cdot}kg^{-1}$
I_{tot}	total current	A
k	thermal conductivity	$W{\cdot}m^{-1}{\cdot}K^{-1}$
L_e	heated length of channel	m
M	molecular weight	$kg{\cdot}kmol^{-1}$
$\boldsymbol{m}$	mass flow rate	$kg{\cdot}s^{-1}$
P_L	heat losses	W
Pr	Prandtl number	
p_r	reduced pressure (p/p_{cr})	
P_s	supplied heat	W
P_t	heat dissipated by thermosyphon	W
q	heat flux	$W{\cdot}m^{-2}$
Re	Reynolds number	
s	suppression factor	
t_s	saturation temperature	°C
t_w	wall temperature	°C
U	voltage	V
x	vapor quality	

Greek symbols

μ	dynamic viscosity	$N{\cdot}s{\cdot}m^{-2}$
ρ	density	$kg{\cdot}m^{-3}$

Subscript

Exp	experimental
g	vapor
L	liquid
Pool	pool boiling
tp	two-phase

most critical criteria for the thermal performance of the thermosyphon system.

The main objective of a presently ongoing study at the Royal Inst. of Technology, Stockholm, has been to investigate how to design an advanced thermosyphon loop for high cooling capacity while maintaining low temperature differences.

One method to achieve small temperature differences is the use of enhanced surfaces. The effect of using threaded surfaces in the evaporator section was studied by Khodabandeh and Palm [3]. The pressure is known to have a significant effect on the heat transfer coefficient and temperature difference. Many researchers have investigated this effect. From the beginning of 1930s studies have been made to determine size, shape and frequency of single bubbles at different pressures. From 1950s efforts were spent to develop semi-empirical correlations to predict heat transfer coefficients as a function of pressure.

The static bubble departure theories predict a slow decrease of the departure radius at increasing pressure with given cavity size. Slooten [4] showed that with higher pressure, smaller cavities are activated which give rise to additional, smaller bubbles.

In the last decades many researchers have studied the effect of pressure on the boiling heat transfer coefficient. Cooper [5], Gorenflo [6], Klimenko [7], Liu and Winterton [8], Steiner and Taborek [9], Bao et al. [10], Ross and Radermacher [11], and Tran et al. [12], as well as other researchers, all reported that the boiling heat transfer coefficient increases with increasing pressure.

2. Experimental setup

The experimental set up, illustrated in Fig. 1, consists of a roll-bond condenser, air cooled by free convection. The evaporator section was made of three small blocks of copper (10 × 20 × 15 mm), connected in parallel, in each of which five vertical circular channels with diameter 1.5 mm,

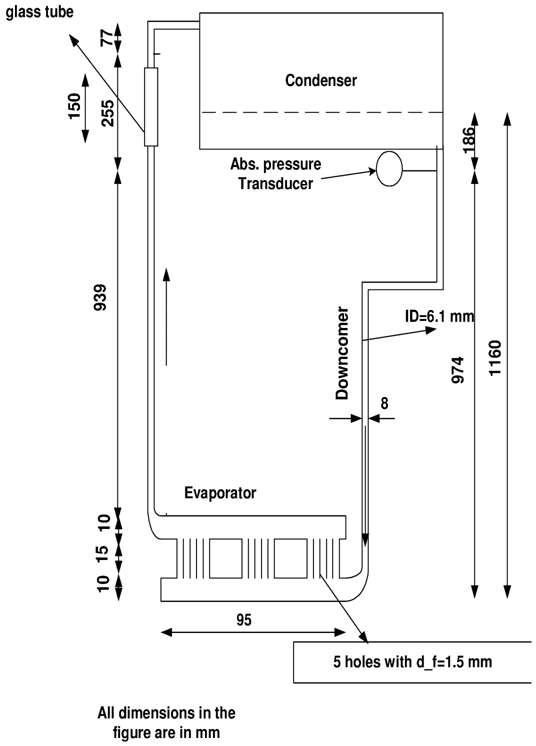

Fig. 1. Schematic drawing of the thermosyphon system.

and length 15 mm were drilled. A riser and a downcomer connected the evaporator to the condenser. A glass tube was provided in the riser near the condenser for visualization of the flow regimes as well as for monitoring any fluctuation of the flow in the thermosyphon system. The hot components were simulated by pieces of copper with a front area of 9.53 × 9.53 mm^2 heated by electric heaters. The heating power was distributed equally to three resistive heaters, one on each evaporator, and was varied in steps from 10–110 W to each evaporator. The heat load was determined by measuring the voltage and current by using a Fluke 45 Dual Multimeter. Tests were done at the reduced pressures 0.02–0.05–0.1–0.2–0.3. Two sets of evaporators were used, one with threaded and one with smooth channels. The threaded channels had an inner minimum diameter of 1.12 mm, an average diameter of 1.27 mm and a pitch of 0.35 mm (Fig. 2). Once the system was stable, the power, the pressure and temperatures were recorded. Isobutane was used as working fluid. The inside wall temperatures were measured by three thermocouples one for each copper block inserted in holes 1 mm from the inside vertical channels on the side facing the heater. For the measurement of the system pressure, an electronic pressure transducer (Druck, PDCR, 960, 10 bar) was used. The corresponding saturation temperature was calculated by the software program Refprop6 [13]. The temperature and saturation pressure were registered by a data logger (Campbell, 21×). The experimental heat transfer coefficient was defined by the ratio of heat flux, P_t/A, and temperature difference, $t_w - t_s$, between the evaporator wall and the saturation temperature of the fluid. The measured temperature difference is adjusted with the temperature difference due to the heat conduction between the location of the thermocouple in the wall and inner tube surface. The heat transfer coefficient is affected by errors in the temperature difference between the wall and the fluid and in the heat flux measurement. For calibration purpose, this difference was measured at zero power input at thermally stable conditions. It was found that the difference in temperature determined from the thermocouple reading and the pressure reading was always less than 0.2 K. This value is therefore considered to be the error in the temperature difference during the tests. All supplied heat, P_s, is not dissipated through the heat sink. Some of the heat, P_L, is conducted through the insulation (PTFE) at the top and bottom sides. The heat dissipated in the evaporator, P_t, is given by $P_t = P_s - P_L$ (Table 1). The supplied heat is calculated by $P_s = U \cdot I_{tot}/3$. According to the manufacturer, the relative maximum error in the voltage and current measurements is about 0.025% and 0.2% of reading respectively. The relative losses, P_L/P_s, for reduced pressure 0.1 are estimated to be between 16–5% for 10 and 110 W heat loads respectively. As can be seen, due to the fact that the error caused by heat losses in some cases is high, P_t, has been used for estimation of heat transfer coefficient. The maximum relative uncertainty in the estimation of P_L has been estimated to 10%.

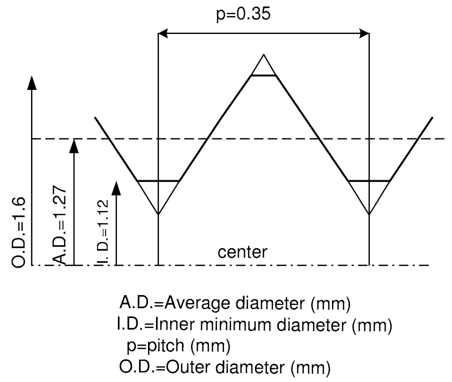

Fig. 2. Schematic drawing of the threaded surface.

Table 1
Heat dissipated in the evaporator

p_r	P_t (W)	P_t (W)	P_t (W)	P_t (W)	P_t (W)	P_t (W)
0.02	7.6	25.1	43.6	62.1	81.1	99.5
0.05	8.3	26.4	45.2	64.7	83.7	102.3
0.1	8.4	27.1	46.0	65.8	85.0	104.2
0.2	8.7	27.7	46.8	66.5	85.7	105.6
0.3	8.9	28.5	47.8	67.1	87.2	106.4

3. Results and discussion

3.1. The effect of pressure on the heat transfer coefficient

According to the above-cited references it is generally found that the heat transfer coefficient increases with the reduced pressure to the power of 0.2 to 0.35, except near the critical point where the exponent is larger.

Because of the law of corresponding states the variation of thermodynamic and transport properties with reduced pressure is similar for different fluids.

As can be seen in Fig. 3 the temperature difference increases with increasing heat flux, but with different slopes, depending on the saturation pressure in the system. As the heat transfer coefficient is the heat flux divided by the temperature difference, this indicates higher heat transfer coefficients with increasing pressure.

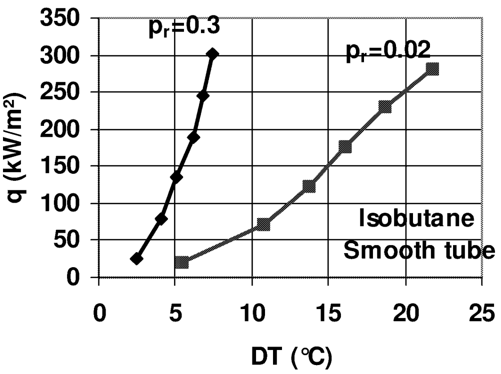

Fig. 3. Heat flux vs. temperature difference, Isobutane, smooth tube.

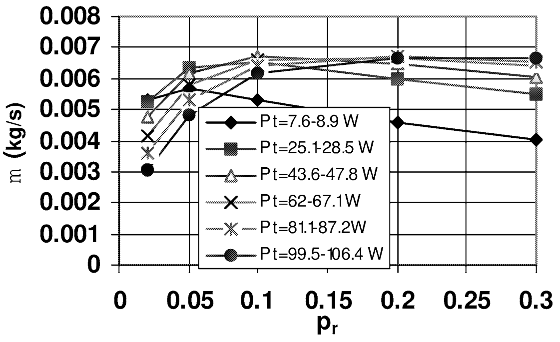

Fig. 4. Mass flow vs. reduced pressure at different heat fluxes from 20.9–301 kW·m^{-2}, Isobutane. Result of simulation.

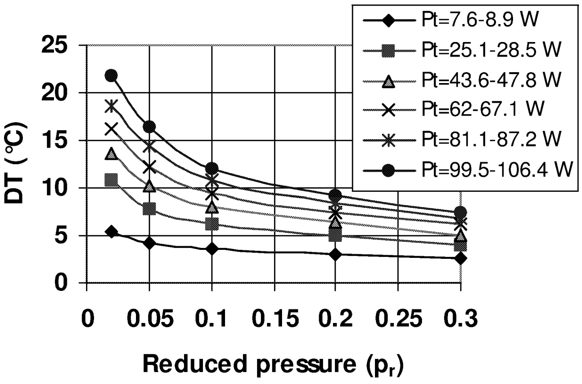

Fig. 5. Temperature difference vs. reduced pressure for smooth tube, Isobutane.

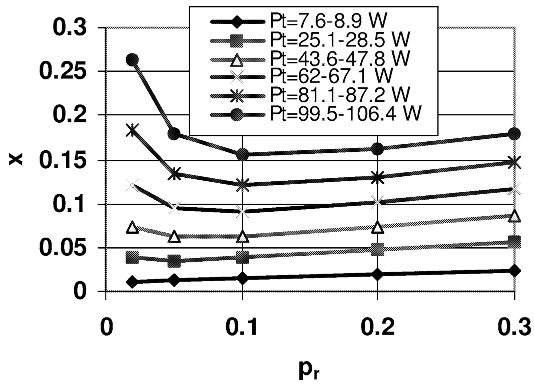

Fig. 6. Mass fraction vs. reduced pressure for smooth tube, Isobutane. Result of simulation.

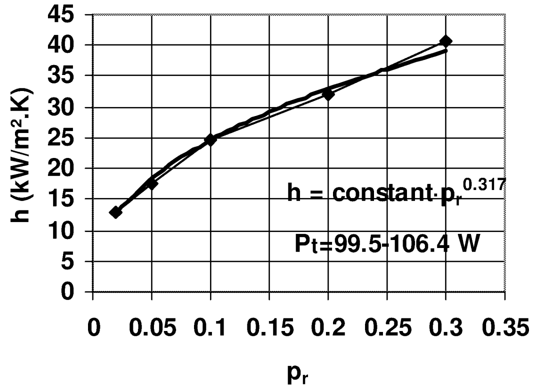

Fig. 7. Heat transfer coefficient vs. reduced pressure for smooth tube, Isobutane.

In order to find the effect of pressure on the mass flow in the system, a simulation program was used. With all other parameters unchanged, the simulation shows that the mass flow in the channel is a function of both the reduced pressure and the heat flux. As seen in Fig. 4, it decreases with increasing heat flux at low reduced pressures but increases with increasing heat flux at higher reduced pressures. For a given heat flux it first increases with increasing reduced pressure, reaches its maximum and finally decreases slightly. It should be noted that for a range of reduced pressures and heat fluxes the mass flow is fairly constant. Fig. 5 shows the steady increase in the temperature difference with increasing heat input. It also clearly shows that the temperature difference increases as the reduced pressure decreases. As shown in Fig. 6, the simulation of vapor mass fraction indicates almost constant values at reduced pressures in the range 0.1–0.3, but significantly higher vapor fractions at lower reduced pressures, especially at higher heat fluxes. Experimental results are often expressed in the form of $h \propto \text{constant} \cdot p_{\mathrm{r}}^{\mathrm{m}}$, in which m is generally between 0.2–0.35. In the present case, $m = 0.317$ correlates the experimental data well for the smooth tube with Isobutane as refrigerant (Fig. 7).

3.2. Effect of threaded surface at different reduced pressures

The effect of threaded surfaces on the heat transfer coefficient was investigated by Khodabandeh and Palm [3], using R134a and Isobutane as refrigerants at constant pressure. That relatively low temperature differences can be achieved at high-reduced pressure is shown in Fig. 8. The same figure shows, that for most heat fluxes the temperature difference is reduced to less than a third by increasing the reduced pressure from 0.02 to 0.3.

3.3. Comparison between calculated and experimental heat transfer coefficients

The experimental results are compared to the Cooper pool boiling correlation [5] and to Liu and Winterton's [8] correlation for flow boiling. Liu and Winterton argued that the bulk boiling region can be dominated by either the nucleate or convective boiling. The contribution of the nucleate boiling should be suppressed when the convection is larger, and vice versa. Cooper's correlation was used to predict h_{pool}.

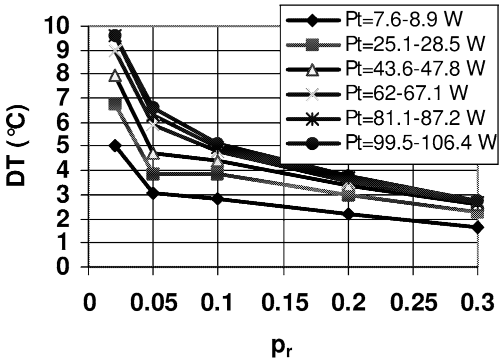

Fig. 8. Temperature difference vs. reduced pressure for threaded surface, Isobutane.

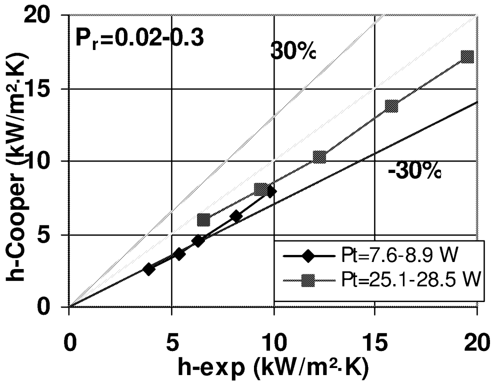

Fig. 9. Comparison between the calculated and measured heat transfer coefficient for smooth tube surface with Isobutane.

$$h_{\mathrm{tp}} = \left[(E \cdot h_{\mathrm{L}})^2 + (s \cdot h_{\mathrm{pool}})^2\right]^{0.5}$$

$$h_{\mathrm{pool}} = 55 \cdot p_{\mathrm{r}}^{0.12} \cdot \left(-\log 10(p_{\mathrm{r}})\right)^{(-0.55)} \cdot M^{(-0.5)} \cdot q^{0.67}$$

$$E = \left[1 + (x) \cdot Pr_{\mathrm{L}} \cdot \left(\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{g}}} - 1\right)\right]^{0.35}$$

$$s = \left[1 + 0.055 \cdot E^{0.1} \cdot (Re_{\mathrm{L}})^{0.16}\right]^{(-1)}$$

$$h_{\mathrm{L}} = 0.023 \cdot \left(\frac{k_{\mathrm{L}}}{d}\right) \cdot (Re_{\mathrm{L}})^{0.8} \cdot (Pr_{\mathrm{L}})^{0.4}$$

$$Re_{\mathrm{L}} = \frac{G \cdot (1 - x) \cdot d}{\mu_{\mathrm{L}}}$$

$$Pr_{\mathrm{L}} = \frac{C_{p_{\mathrm{L}}} \cdot \mu_{\mathrm{L}}}{k_{\mathrm{L}}}$$

$$x = \frac{4 \cdot q \cdot L_{\mathrm{e}}}{h_{\mathrm{fg}} \cdot G \cdot d}$$

In the calculation x is taken as the thermodynamic equilibrium quality at the end of the heated channel. As can be seen from Figs. 9–11 the heat transfer coefficient calculated by Cooper's correlation is in good agreement with the experimental results. Cooper's correlation is developed for nucleate boiling without considering convective boiling. At lowest heat input and lowest reduced pressure, there seems to be a certain contribution of convective evaporation, which may be the explanation to the higher experimental heat transfer coefficients compared to Cooper's correlation.

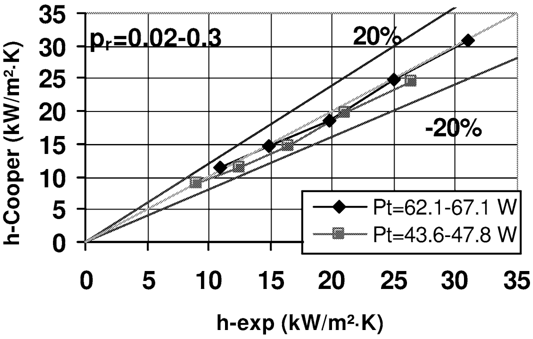

Fig. 10. Comparison between the calculated and measured heat transfer coefficient for smooth tube surface with Isobutane.

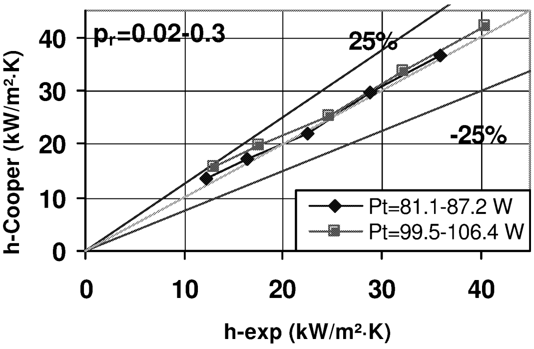

Fig. 11. Comparison between the calculated and measured heat transfer coefficient for smooth tube surface with Isobutane.

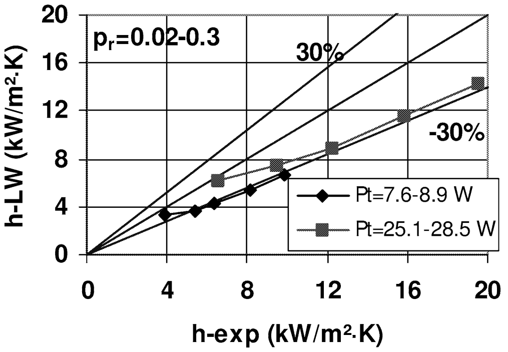

Fig. 12. Comparison between the calculated and measured heat transfer coefficient for smooth tube surface with Isobutane.

Figs. 12–14 show a comparison of the heat transfer coefficient calculated by Liu and Winterton's correlation vs. experimental results. The correlation predicts the data well; the mean deviation is about 17 percent. However, it seems clear that nucleate boiling is the dominant mechanism in all tests. In this regime, the size, shape and bubble departure frequency have a significant influence on the heat transfer coefficient. Presently, there seems to be no generally accepted explanation to the observed enhancement of heat transfer at increasing pressure. Niro [14] suggested, by combining the Clausius–Clapeyron equation with the Laplace equation for the over-pressure inside a bubble, that the superheat necessary for bubble nucleation would decrease with increasing pressure. According to Slooten [4]

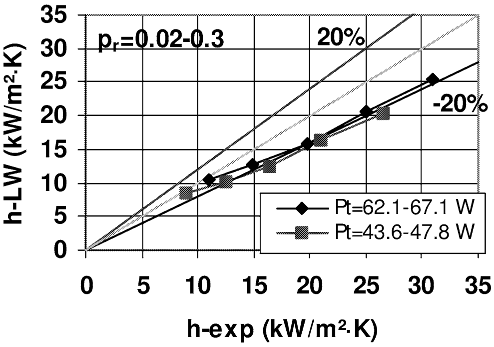

Fig. 13. Comparison between the calculated and measured heat transfer coefficient for smooth tube surface with Isobutane.

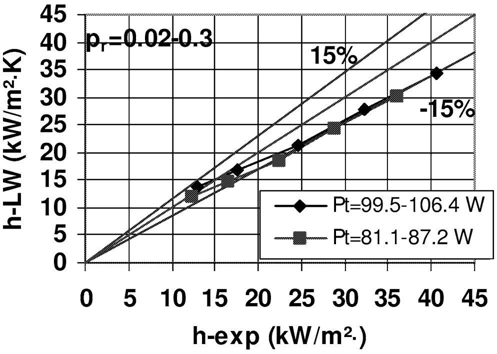

Fig. 14. Comparison between the calculated and measured heat transfer coefficient for smooth tube surface with Isobutane.

for static departure (high pressure), the average departure diameter decreases with increasing pressure. This reduction is due to the additional activation of smaller cavities at higher pressures, and smaller cavities give smaller bubbles. But for a given cavity, Slooten observed only a small reduction of the departure diameter at increasing pressure. Activation of the additional new small cavities due to the high pressure, implies higher heat transfer coefficient with the same reasoning as for rough surfaces. According to Collier [15] the product of bubble departure frequency and diameter is constant. However under this condition, with decreasing bubble departure diameter at increased pressure, bubble frequency increases, which contributes to increasing heat transfer coefficient.

4. Conclusion

The pressure has a significant effect on the boiling heat transfer coefficient in the narrow channels of the thermosyphon evaporator. It was found that the heat transfer coefficient at most points at a given heat flux is more than three times larger at the reduced pressure $p_r = 0.3$ than $p_r = 0.02$. The experimental heat transfer coefficients for smooth tube surfaces are in relatively good agreement with Liu–Winterton and Cooper's pool boiling correlations. This indicates a dominant contribution of nucleate boiling to heat transfer. The improvement of the heat transfer coefficient with increasing pressure was thought to be primary due to the fact that at higher pressure a larger number of cavities are active. Comparing smooth and threaded channels, using Isobutane as refrigerant, the heat transfer coefficient at all pressures was considerably improved by the threads. The largest improvements were found at the highest heat fluxes.

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