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Application of a two-phase thermosyphon loop with minichannels and a minipump in computer cooling

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Abstract This paper focuses on the computer cooling capacity using the thermosyphon loop with minichannels and minipump. The one-dimensional separate model of two-phase flow and heat transfer in a closed thermosyphon loop with minichannels and minipump has been used in calculations. The latest correlations for minichannels available in literature have been applied. This model is based on mass, momentum, and energy balances in the evaporator, rising tube, condenser and the falling tube. A numerical analysis of the mass flux and heat transfer coefficient in the steady state has been presented.

Keywords: Thermosyphon loop; Two phase flow; Computer cooling

Nomenclature

A	–	cross-section area of the channel, m^2
B	–	breadth, m
D	–	internal diameter of the tube, m
$\dot{G}$	–	mass flux (mass velocity), kg/(m^{2}s)
g	–	gravitational acceleration, m/s^2
H	–	height, m

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h	–	heat transfer coefficient, W/(m^{2}K)
L	–	total length of the closed loop, m
L_H	–	length of heated section, m
L_C	–	length of cooled section, m
$\dot{m}$	–	mass flow rate, kg/s
p	–	pressure, Pa
p_{RED}	–	reduced pressure, –
$\dot{Q}$	–	heat rate, W
$\dot{q}$	–	heat flux, W/(m^2)
s	–	axial coordinate around the loop, m
U	–	wetted perimeter, m
$\dot{V}$	–	volumetric flow rate, m^3/s
x	–	quality of vapour, –

Greek symbols

α	–	void fraction, –
ρ	–	mass density, kg/m^3
τ_w	–	wall shear stress, N/m^2

Subscripts

C	–	cooler
H	–	heater
L	–	liquid
V	–	vapour
$TP, 2p$	–	two-phase
$L0$	–	liquid phase only
TPB	–	two-phase boiling
TPC	–	two-phase cooling
REF	–	reference
PB	–	pool boiling

1 Introduction

This paper is an extension of our earlier contributions published in *Archives of Thermodynamics* [4–5]. These articles provide a discussion of conventional tubes of the thermosyphon loop. In case of minichannels constituting the thermosyphon, it is necessary to use new correlations for void fraction and the local two-phase friction coefficient in two-phase region, and local heat transfer coefficient for flow boiling and condensation.

Fluid flow in a thermosyphon loop is created by the buoyancy forces that evolve from density gradients induced by temperature differences in the heating and cooling sections of the loop. The minipump can be used if the mass flux is not high enough to transport heat from evaporator to condenser. Therefore, the minipump promotes natural circulation. A closed thermosyphon loop consists of a heater, cooler and minipump, connected with tubes. The lower heater can be a CPU processor located on the motherboard of personal computer. The cooler can be placed above the heater on the computer chassis. Heat exchangers are connected by tubes in which the liquid refrigerant is in operation.

The single- and two-phase thermosyphon loops find many industrial applications, such as for example: steam generators, thermosyphon reboilers, emergency cooling systems in nuclear reactor cores and reflux boiling systems in light water reactor cores, solar heating and cooling systems, geothermal energy generation [17] and thermal diodes [2,3]. The thermal diode is a device, which allows the heat to be transferred in one direction, and blocks the heat flow in the opposite direction. Thermosyphons can be designed as a closed loop. The closed-loop thermosyphon is also known as a 'liquid fin' [16]. The increasing integration of electronic systems requires improved cooling technologies. Because of increased power levels, miniaturization of the electronic devices and typical cooling techniques, the heat removal due to conduction is not able to transfer such a high heat flux. Thermosyphon cooling is an alternative cooling technology of dissipating high local heat fluxes.

The thermosyphon effect for cooling electronic devices can be applied in innovative miniature loop heat pipe (mLHP) with the evaporator located on the loop below the condenser (Fig. 1). The prototype of mLHP consists of a flat minievaporator and capillary pump. The porous wick material is placed inside the capillary pump. The complete condensation of the medium takes place in a minicondenser. Both the mini-evaporator and minicooler are connected to the separate liquid and vapour channels. The high friction losses in mLHP can be reduced because the liquid/vapour phase flow in the separated channels and the porous wick is only present in minievaporator. The flat-shaped mini-evaporator is integrated with a capillary pump. The wicks generate the capillary pumping pressure, which is required to transport a working fluid along mLHP.

The thermosyphon effect can also be applied in concept of micro-CHP-cogeneration where evaporator is located below condenser (Fig. 2). The use of capillary forces for pumping of the working fluid in the Clausius-Rankine cycle is a new idea that allows the reduction or even the elimination of the pumping device in such cycles. It is also possible to exploit the gravity force to support the operation of a circulation pump by placing the evaporator below the condenser and creating the thermosyphon loop.

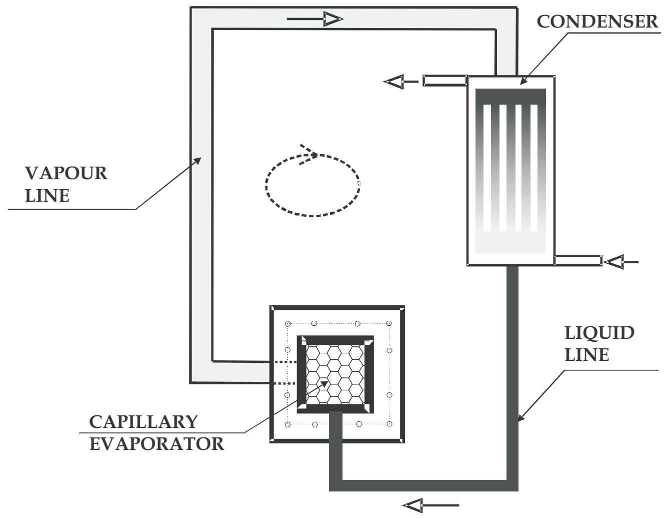

Figure 1: Idea scheme of miniature loop heat pipe (mLHP) with thermosyphon loop.

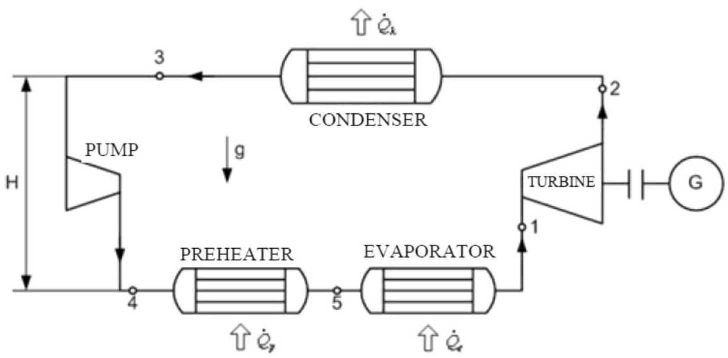

Figure 2: Structural scheme of CHP with LHP and thermosyphon loop [20].

The specific aim of this paper is to report on a new cooling method of the computer using the thermosyphon loop with minichannels and minipump. The thermosyphon cooling system used in computer can be modeled by the rectangular thermosyphon loop with minichannels and minipump heated at the bottom vertical side and cooled at the upper vertical side. The case presented here is a new aspect of computer cooling. The trend toward miniaturization of electronic components provides the adoption of advanced cooling solutions.

2 The model of the two-phase thermosyphon loop with minichannels and minipump

A schematic diagram of a one-dimensional model of the two-phase thermosyphon loop with minichannels and minipump heated from lower vertical section and cooled from upper vertical section is shown in Fig. 3. Fluid flow in a thermosyphon loop is created by the buoyancy forces that evolve from the density gradients induced by temperature differences in the heating and cooling sections of the loop. The minipump can be used if the mass flux is not high enough to transport heat from evaporator to condenser. Therefore, the minipump enhances natural circulation. The thermosyphon loop is heated from lower vertical section ($s_0 \leq s \leq s_1$) by a constant heat flux: $\dot{q}_H$ and cooled in the upper vertical section ($s_4 \leq s \leq s_5$) by a constant heat flux: $\dot{q}_C$. The constant heat fluxes $\dot{q}_H$ and $\dot{q}_C$ are applied in the cross-section area per heated and cooled length: L_H and L_C. The heated and cooled parts of the thermosyphon loop are connected by perfectly insulated channels ($s_1 \leq s \leq s_4,\ \ s_5 \leq s \leq s_8$).

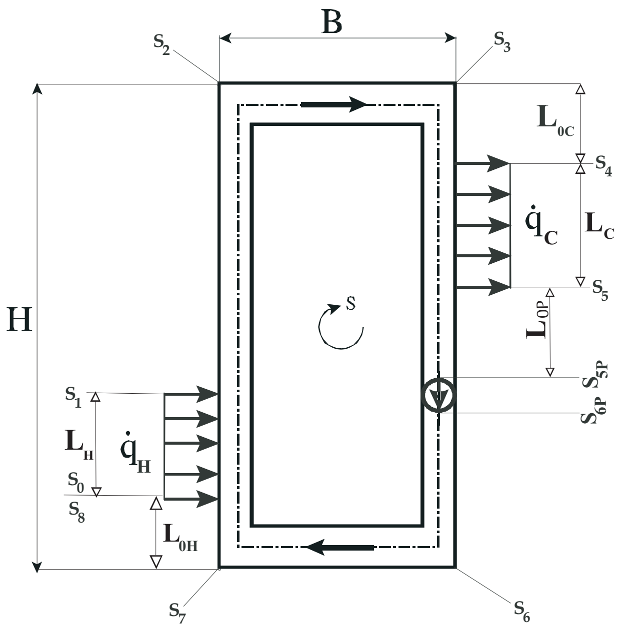

Figure 3: Model of the two-phase thermosyphon loop with minichannels and minipump heated from lower vertical section and cooled from upper vertical section.

As shown in Fig. 3 the space coordinate s circulates around the closed loop. The total length of the loop is denoted by L, the cross-section area of the channel by A and the wetted perimeter by U. Both superheating and subcooling are neglected and a linearly varying quality function $x(s)$ are assumed according to [3–10].

3 Governing equations

The one-dimensional, steady-state governing equations for thermosyphon loop with minichannels and minipump can be written as [3–10,18]:

$$\begin{cases} \dfrac{d\dot{m}}{ds} = 0, \\[2ex] -\dfrac{dp}{ds} = \underbrace{\dfrac{U}{A}\tau_w}_{friction\,term} + \underbrace{\dfrac{\dot{m}}{A}\dfrac{d}{ds}\left[\dfrac{(1-x)^2}{(1-\alpha)\,\rho_L} + \dfrac{x^2}{\alpha\,\rho_V}\right]}_{acceleration\ term} + \underbrace{\beta\, g\,\left[(1-\alpha)\,\rho_L + \alpha\,\rho_V\right]}_{gravitation\ term} \\[6ex] \dfrac{d}{ds}\left(\dfrac{\dot{m}}{A}\,h\right) = \begin{cases} 0 & \text{for insulated regions} \\ \frac{U}{A}\,\dot{q} & \text{for heated and cooled regions} \end{cases} \end{cases} \tag{1}$$

where $\beta = 0$ for $\vec{e}\perp\vec{g}$; $\beta = (+1)$ for $\vec{e}\uparrow \wedge \vec{g}\downarrow$; $\beta = (-1)$ for $\vec{e}\downarrow \wedge \vec{g}\downarrow$ (where $\vec{g}$ – vector acceleration, $\vec{e}$ – versor, $\uparrow$ – sense of a vector).

In the equation of motion of the thermosyphon loop with natural circulation, the pressure term of integration around the loop is equal to zero

$$\oint\left(\frac{dp}{ds}\right) ds = 0.$$

For the thermosyphon loop with minipump the pressure term is

$$\oint\left(\frac{dp}{ds}\right) ds = \Delta p_{PUMP} = \rho_L\, g\, H_{PUMP};\, H_{PUMP} = H_{MAX}\left[1 - \left(\frac{\dot{V}}{\dot{V}_{MAX}}\right)^2\right],$$

with H_{MAX} , $\dot{V}_{MAX}$ from minipump curve [11].

The gravitational term in the momentum equation (1) can be expressed as

$$\oint \{\,\beta\, g\, \rho\,\}\; ds = g\;(\rho_V - \rho_L)\;\Big\{\;(s_1 - s_0)\;\bar{\alpha}_{\langle s_0;s_1\rangle} - (s_5 - s_4)\;\bar{\alpha}_{\langle s_4;s_5\rangle} + \\ + \left[(s_2 - s_1) - (s_4 - s_3)\right]\;\bar{\alpha}_{\langle s_1;s_4\rangle}\;\Big\} = 0\,, \tag{2}$$

where

$$\bar{\alpha}_{\langle s_i;s_j\rangle} = \frac{1}{(s_j - s_i)} \int_{s_i}^{s_j} \alpha_{\langle s_i;s_j\rangle}\,(s)\;ds\;\;. \tag{3}$$

The frictional component of the pressure gradient in two-phase regions was calculated using the two-phase separate model. Due to friction of fluid, the pressure gradient in two-phase regions can be written as follows [14,18]:

$$\frac{U}{A}\,\tau_w = \left(\frac{-dp}{ds}\right)_{2p} = R\;\left(\frac{-dp}{ds}\right)_{L0}\;\;, \tag{4}$$

where R is the local two-phase flow multiplier. $\left(\frac{dp}{ds}\right)_{L0}$ is the liquid only frictional pressure gradient calculated for the liquid total mass flow rate. After integrating the friction term around the loop, we obtain

$$\oint \left(\tfrac{U}{A}\,\tau_w\right)\;ds \;\;= \left(\tfrac{dp}{ds}\right)_{L0}\;\Big\{\;(s_1 - s_0)\;\bar{R}_{\langle s_0;s_1\rangle} + (s_4 - s_1)\;\bar{R}_{\langle s_1;s_4\rangle} \\ +(s_5 - s_4)\;\bar{R}_{\langle s_4;s_5\rangle} + \;(s_8 - s_5)\;\Big\}\;, \tag{5}$$

where

$$\bar{R}_{\langle s_i;s_j\rangle} = \frac{1}{(s_j - s_i)} \int_{s_i}^{s_j} R\,(s)\;ds\;\;. \tag{6}$$

Substituting Eqs. (2) and (5) into the momentum Eq. (1) gives

$$\left(\tfrac{dp}{ds}\right)_{L0}\;\Big\{\,(s_1 - s_0)\,\bar{R}_{\langle s_0;s_1\rangle} + (s_4 - s_1)\,\bar{R}_{\langle s_1;s_4\rangle} + (s_5 - s_4)\,\bar{R}_{\langle s_4;s_5\rangle} + \\ + (s_8 - s_5)\,\Big\} + g\;(\rho_V - \rho_L)\;\Big\{\;(s_1 - s_0)\;\bar{\alpha}_{\langle s_0;s_1\rangle} - (s_5 - s_4)\;\bar{\alpha}_{\langle s_4;s_5\rangle} \\ +\Big[\,(s_2 - s_1) - (s_4 - s_3)\,\Big]\,\bar{\alpha}_{\langle s_1;s_4\rangle}\;\Big\}\; + \,\rho_L\, g\, H_{PUMP} = 0\,. \tag{7}$$

The El-Hajal *et al.* [13] empirical correlation for the void fraction was applied. The local two-phase friction coefficient in two-phase adiabatic region was calculated using the Zhang and Webb [25] formula. In two-phase heating and cooling sections the local two-phase friction coefficient was calculated using the Tran *et al.* [23] and Cavallini *et al.* [12] formula, respectively (Tab. 1). The working fluid was distilled water. A miniature pump curve from (Blanchard *et. al.* [11]) was included in calculations.

Table 1: Minichannels. Correlation for the void fraction and the friction pressure drop of two-phase flow.

Researcher	Correlation
El-Hajal *et al.* [13]	$\alpha_{HAJAL} = \frac{\alpha_{HOM}-\alpha_{STEINER}}{\ln\left(\frac{\alpha_{HOM}}{\alpha_{STEINER}}\right)}$; $\alpha_{HOM} = \frac{1}{1+\frac{1-x}{x}\left(\frac{\rho_V}{\rho_L}\right)}$; $\alpha_{STEINER} = \left(\frac{x}{\rho_V}\right) \times$ $\times \left\{[1+0.12\,(1-x)]\left[\frac{x}{\rho_V}+\frac{1-x}{\rho_L}\right] + \right.$ $\left. + \frac{1.18\cdot(1-x)\,[g\cdot\sigma\,(\rho_L-\rho_V)]^{0.25}}{\dot{G}\,(\rho_L)^{0.5}}\right\}^{(-1)}$ (8)
Zhang and Webb [25]	$\left(\frac{dp}{dl}\right)_{2p}^{Z-W} = \Phi_{L0}^2\left(\frac{dp}{dz}\right)_{L0}$; $\Phi_{L0}^2 = (1-x)^2 + 2.87\,(x)^2\left(\frac{P}{P_{CRIT}}\right)^{(-1)} +$ $+1.68\,(1-x)^{0.25}\left(\frac{P}{P_{CRIT}}\right)^{(-1.64)}$ (9)
Tran *et al.* [23]	$\left(\frac{dp}{dl}\right)_{2p}^{TRAN} = \Phi_{L0}^2\left(\frac{dp}{dz}\right)_{L0}$; $\Phi_{L0}^2 = 1+\left(4.3\,Y^2-1\right)\left[N_{CONF}\,(x)^{0.875}\,(1-x)^{0.875}+(x)^{1.75}\right]$; $N_{CONF} = \frac{\left[\frac{\sigma}{g\cdot(\rho_L-\rho_V)}\right]^{0.5}}{D}$; $Y = \sqrt{\left(\frac{dp}{dz}\right)_{V0} \Big/ \left(\frac{dp}{dz}\right)_{L0}}$ (10)
Cavallini *et al.* [12]	$\left(\frac{dp}{dl}\right) = \Phi_{L0}^2\left(\frac{dp}{dz}\right)_{L0}$; $\Phi_{L0}^2 = E + \frac{1.262\cdot F\,H}{(\mathrm{We})^{0.1458}}$; $(\mathrm{We}) = \frac{\dot{G}^2\,D}{\sigma\,\rho_V}$; $E = (1-x)^2 + x^2\,\frac{\rho_L\,f_{V0}}{\rho_V\,f_{L0}}$; $F = x^{0.6978}$; $H = \left(\frac{\rho_L}{\rho_V}\right)^{0.3278}\left(\frac{\mu_V}{\mu_L}\right)^{(-1.181)}\left(1-\frac{\mu_V}{\mu_L}\right)^{3.477}$ (11)

4 Results

The mass flux distribution, $\dot{G}$, versus heat flux, $\dot{q}_H$, for loop with the minipump and gravity loop without the minipump were obtained numerically under the steady-state condition using separate model, as is shown in Fig. 4. For thermosyphon loop without the minipump the two-flow regimes can be clearly identified (Fig. 4): GDR – gravity dominant regime and FDR – friction dominant regime. The presence of GDR and FDR dominant regime was also demonstrated experimentally by Agostini and Ferreira [1]. The explanation of existing regimes were given by Vijayan *et. al.* [24].

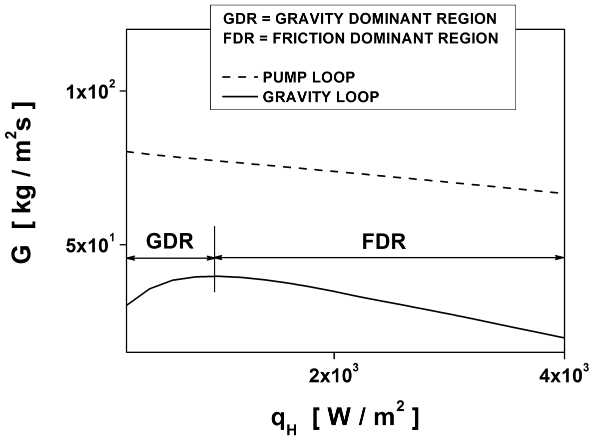

Figure 4: Mass flux $\dot{G}$ as a function of $\dot{q}_H$ (L = 0.2 m, B = 0.02 m, H = 0.08 m, D = 0.0025 m).

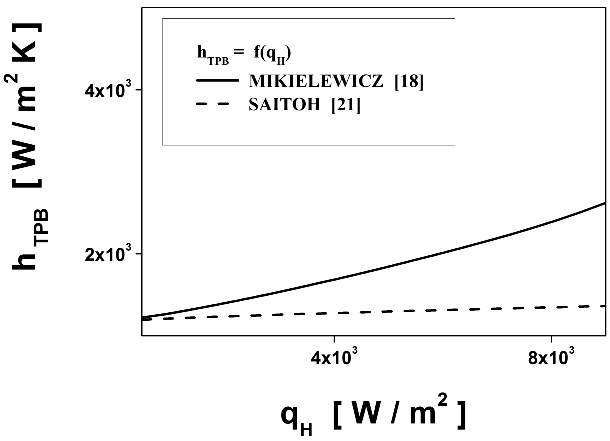

Figure 5: Heat transfer coefficient h_{TPB} as a function of $\dot{q}_H$ for heater.

Table 2: Minichannels. Correlation for the heat transfer coefficient for flow boiling.

Researcher	Correlation	
Mikielewicz *et al.*, [19]	$\frac{h_{TPB}^{JM}}{h_{REF}} = \sqrt{\left(R_{M-S}\right)^{n} + \frac{1}{1+P}\left(\frac{h_{PB}}{h_{REF}}\right)^{2}}\ ;$ $R_{M-S} = \left[1 + 2\left(\frac{1}{f_1} - 1\right) x \left(N_{CONF}\right)^{(-1)}\right] (1-x)^{\frac{1}{3}} + x^{3}\frac{1}{f_{1z}}\ ;$ $LAM \Rightarrow n = 2\ ;\ \alpha_{REF}^{LAM} = 4.36\left(\frac{\lambda_L}{D}\right);\ f_1^{LAM} = \left(\frac{\mu_L}{\mu_V}\right)\left(\frac{\rho_V}{\rho_L}\right);$ $f_{1z}^{LAM} = \left(\frac{\lambda_L}{\lambda_V}\right);$ $TUR \Rightarrow\ n = 0.76\ \ ;\ h_{REF} = 0.023\frac{\lambda_L}{D}\left(\mathrm{Re}_{L0}\right)^{0.8}\left(\mathrm{Pr}_L\right)^{\frac{1}{3}}\ ;$ $f_1^{TUR} = \left(\frac{\mu_L}{\mu_V}\right)^{0.25}\left(\frac{\rho_V}{\rho_L}\right);\ f_{1z}^{TUR} = \left(\frac{\mu_V}{\mu_L}\right)^{\frac{7}{15}}\left(\frac{c_{pL}}{c_{pV}}\right)^{\frac{1}{3}}\left(\frac{\lambda_L}{\lambda_V}\right)^{\frac{3}{2}}\ ;$ $P = 2.53\times 10^{(-3)}\left(\mathrm{Re}_{L0}\right)^{1.17}\left(\mathrm{Bo}\right)^{0.6}\left(R_{M-S}-1\right)^{(-0.65)}\ ;$ $(\mathrm{Bo}) = \frac{\dot{q}}{\dot{G}r};\ \left(\mathrm{Re}_{L0}\right) = \frac{\dot{G}d}{\mu_L};$ $h_{PB} = 55\dot{q}^{0.67} M^{(-0.5)}\left(\frac{P_n}{P_{CRIT}}\right)^{0.12}\left[-\log_{10}\left(\frac{P_n}{P_{CRIT}}\right)\right]^{(-0.55)};$	(12)
Saitoh *et al.*, [21]	$h_{TPB}^{SAITOH} = Eh_{REF} + S\,h_{POOL};$ $h_{POOL} = 207\left(\frac{\lambda_L}{d_b}\right)\left(\frac{q\,d_b}{\lambda_L T_{SAT}}\right)^{0.745}\left(\frac{\rho_G}{\rho_L}\right)^{0.581}\left(\mathrm{Pr}_L\right)^{0.533}\ ;$ $h_{REF} = \begin{cases} (\mathrm{Nu})_{LAM}\left(\frac{\lambda_L}{D}\right) & ; \quad LAM \\ 0.023\left(\mathrm{Re}_L\right)^{\frac{4}{5}}\left(\mathrm{Pr}_L\right)^{0.4}\left(\frac{\lambda_L}{D}\right) & ; \quad TUR \end{cases}$ $E = 1 + \frac{\left(\frac{1}{X}\right)^{1.05}}{1+\left(\mathrm{We}_G\right)^{(-0.4)}}\ ;\ \left(\mathrm{Re}_G\right) = \frac{\dot{G}_G D}{\mu_G}\ ;\ \left(\mathrm{Re}_L\right) = \frac{\dot{G}_L D}{\mu_L}\ ;$ $S = \frac{1}{1+0.4\cdot\left[\left(10^{(-4)}\right)\cdot\left(Re_{TP}\right)\right]^{1.4}};\ d_b = 0.51\cdot\left(\frac{2\cdot\sigma}{g\cdot(\rho_L-\rho_G)}\right) 0.5;$ $\left(\mathrm{We}_G\right) = \frac{\dot{G}_G D}{\sigma\rho_G};\ \left(\mathrm{Re}_{TP}\right) = \left(\mathrm{Re}_L\right)(F)^{1.25}\ ;\ \dot{G}_G = \dot{G}x;\ \dot{G}_L = \dot{G}(1-x)\ ;$ $X = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_G}{\rho_L}\right)^{0.5}\left(\frac{\mu_L}{\mu_G}\right)^{0.1} \quad \text{for} \quad \begin{cases} \left(\mathrm{Re}_L\right) > 1000 \\ \left(\mathrm{Re}_G\right) > 1000 \end{cases}\ ;$ $C_G = 0.046\ \ ;\quad C_L = 16\ \ ;$ $X = \left(\frac{C_L}{C_G}\right)^{0.5}\times\left(\mathrm{Re}_G\right)^{(-0.4)}\times\left(\frac{G_L}{G_G}\right)^{0.5}\left(\frac{\rho_G}{\rho_L}\right)^{0.5}\left(\frac{\mu_L}{\mu_G}\right)^{0.5}$ $\text{for} \quad \begin{cases} \left(\mathrm{Re}_L\right) < 1000 \\ \left(\mathrm{Re}_G\right) > 1000 \end{cases}.$	(13)

The heat transfer coefficient for flow boiling in evaporator with minichannels h_{TPB} distributions versus heat flux $\dot{q}_H$ in evaporator $(s_0 \leq s \leq s_1)$ was numerically obtained using Mikielewicz *et al.* [19] and Saitoh *et al.* [21] correlations (Tab. 2). The results obtained for minichannels are presented in Fig. 5. The results show that the distributions of heat transfer coefficient for flow boiling in the heater with minichannels calculated using the

Mikielewicz *et al.* [19] formula, the modified Saitoh *et al.* formula [21] have the same trend. According to the results of this study, mass flow increases with increasing heat flux. The similar effect is reported by Khodabandeh [15]. He suggested that the observed trend indicates on the dominance of nucleate boiling.

The heat transfer coefficient in flow condensation in cooler with minichannels h_{TPC} distributions versus heat flux $\dot{q}_C$ in condenser ($s_4 \leq s \leq s_5$) was calculated numerically using Mikielewicz *et al.* [19] and Shah [22] correlations (Tab. 3).

Table 3: Minichannels. Correlation for the condensation heat transfer coefficient.

Researcher	Correlation
Mikielewicz *et al.* [19]	$\frac{h_{TPB}^{JM}}{h_{REF}} = \sqrt{\left(R_{M-S}\right)^n}$ (14)
Shah [22]	$h_{TPC}^{SHAH} = h_{L0} \times \left[(1-x)^{0.8} + \frac{3.8\,(x)^{0.76}(1-x)^{0.04}}{(p_{RED})^{0.38}}\right]$ (15)

The results obtained for minichannels are presented in Fig. 6. The heat transfer coefficient for flow condensation in minichannels slowly increases with increasing heat flux in the cooler $\langle s_4\,;\ s_5\rangle$ for the Mikielewicz *et al.* [19] formula, the modified Shah formula [22] as it shown in Fig. 6. Comparison of these two distributions show that the results have similar tendency.

5 Conclusions

The results of the study shows that the one-dimensional two-phase separate model can be used to describe heat transfer and fluid flow in the thermosyphon loop with minichannels and minipump heated from lower vertical section and cooled from upper vertical section. The quality of vapour in the two-phase regions is assumed to be a linear function of the coordinate around the loop.

In order to evaluate performance of the thermosyphon loop with minichannels the following correlation have been used: El-Hajal *et al.* [13] correlation for void fraction, Zhang and Webb [25] correlation for the friction pressure drop of two-phase flow in adiabatic region, Tran *et al.* [23] and Cavallini *et al.* [12] correlation for the friction pressure drop of two-phase flow in heat-

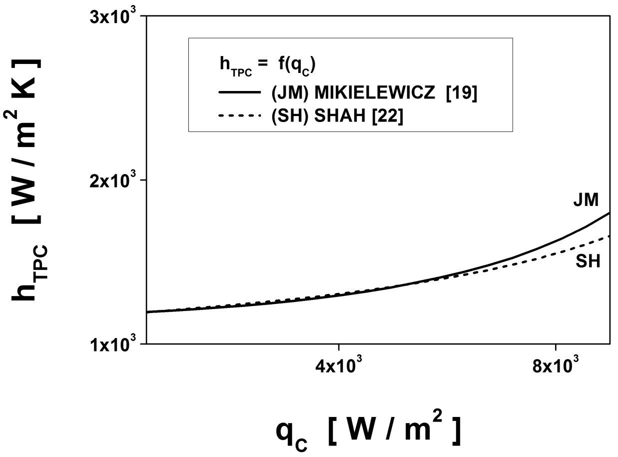

Figure 6: Cooler. Heat transfer coefficient h_{TPC} as a function of $\dot{q}_C$.

ing and cooling section, respectively (Fig. 4), and Mikielewicz *et al.* [19] and Saitoh *et al.* [21] correlations for the heat transfer coefficient for flow boiling in evaporator with minichannels (Fig. 5), Mikielewicz *et al.* [19] and Shah [22] correlations for the heat transfer coefficient in flow condensation in cooler with minichannels (Fig. 6). Distilled water as the working fluid was used in calculations.

In case of thermosyphon loop with minichannels and without a minipump the distribution of the mass flux against the heat flux approaches a maximum and then slowly decreases and the two-flow regimes can be clearly identified as the gravity dominant regime (GDR) and the friction dominant regime (FDR) as presented in Fig. 4. If the mass flow rate is not high enough to circulate the necessary fluid to transport heat from evaporator to condenser, the minipump can be used to promotes natural circulation.

For thermosyphon loop with minichannels and minipump at steady-state condition as it is demonstrated in Fig. 4, the mass flux, $\dot{G}$, decreases with increasing heat flux, $\dot{q}_H$, for friction dominant region (FDR).

The heat transfer coefficient for flow boiling in evaporator with minichannels increases with increasing heat flux (Fig. 5). The heat transfer coefficient for flow condensation in cooler with minichannels slowly increases with increasing heat flux (Fig. 6).

The numerical analysis of the model of thermosyphon loop with minichannels and minipump heated from lower vertical section and cooled from upper vertical section indicates that the presented solution is a power tool for increasing the effectiveness of computer cooling. The offered variant has both a simple design and manufacture and can be a theoretical basis for further work on prototype testing.

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