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Computer cooling using a two phase minichannel thermosyphon loop heated from horizontal and vertical sides and cooled from vertical side

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Abstract In the present paper it is proposed to consider the computer cooling capacity using the thermosyphon loop. A closed thermosyphon loop consists of combined two heaters and a cooler connected to each other by tubes. The first heater may be a CPU processor located on the motherboard of the personal computer. The second heater may be a chip of a graphic card placed perpendicular to the motherboard of personal computer. The cooler can be placed above the heaters on the computer chassis. The thermosyphon cooling system on the use of computer can be modeled using the rectangular thermosyphon loop with minichannels heated at the bottom horizontal side and the bottom vertical side and cooled at the upper vertical side. The riser and a downcomer connect these parts. A one-dimensional model of two-phase flow and heat transfer in a closed thermosyphon loop is based on mass, momentum, and energy balances in the evaporators, rising tube, condenser and the falling tube. The separate two-phase flow model is used in calculations. A numerical investigation for the analysis of the mass flux rate and heat transfer coefficient in the steady state has been accomplished.

Keywords: Thermosyphon loop; Two phase flow

Nomenclature

A – cross-section area of the channel, m
B – breadth, m

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D	–	internal diameter of the tube, m
$\dot{G}$	–	mass flux, kg/(m^{2}s)
g	–	acceleration, m/s^2
H	–	height, m
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
$\dot{q}$	–	heat flux density, W/(m^{2}K)
s	–	axial co-ordinate around the closed-loop, m
x	–	quality of vapour

Greek symbols

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

Subscripts

L	–	liquid
V	–	vapour
TP	–	two-phase

1 Introduction

This paper is a continuation and extension of our previous works published in Archives of Thermodynamics [1,2,4]. Our last article provides a discussion of the conventional tubes of the thermosyphon loop. In case of a mini-channels thermosyphon it is necessary to apply some new correlations for void fraction and the local two-phase friction coefficient in two-phase region, and local heat transfer coefficient in flow boiling and condensation.

In the present paper it is proposed to consider the computer cooling capacity using the thermosyphon loop. Our closed thermosyphon loop is combined with two heaters and one cooler which are connected by tubes. The first heater may be a CPU processor located on the motherboard of personal computer. The second heater may be a chip of a graphic card placed perpendicular to the motherboard of personal computer. The cooler can be placed above the heaters on the computer chassis. Heat exchangers are connected by tubes in which the liquid refrigerant circulates.

The thermosyphon cooling system can be modeled using rectangular thermosyphon loop with minichannels heated at the bottom horizontal side and the bottom vertical side and cooled at the upper vertical side. The riser and a downcomer connect these parts. 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.

Thermosyphons are less expensive than other cooling schemes because they are pumpless. 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 [8] and thermal diodes [4]. 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" [7]. 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 conduction are not able to cool such a high heat flux. Thermosyphon cooling is an alternative cooling technology of dissipating high local heat fluxes.

2 A model of the two-phase thermosyphon loop heated from horizontal and vertical sides and cooled from vertical side

A schematic diagram of a one-dimensional model of the two-phase thermosyphon loop heated from horizontal and vertical sides and cooled from vertical side is shown in Fig. 1.

The thermosyphon loop is heated from horizontal side ($s_0 \leq s \leq s_1$) by a constant heat flux: $\dot{q}_{H1}$ and vertical side ($s_3 \leq s \leq s_4$) by a constant heat flux: $\dot{q}_{H2}$, and cooled in the vertical section ($s_7 \leq s \leq s_8$) by a constant heat flux: $\dot{q}_C$. The constant heat fluxes $\dot{q}_{H1}$, $\dot{q}_{H2}$ and $\dot{q}_C$ are applied in the cross-section area per heated and cooled length: L_{H1}, $L_{H2}$and L_C. The heated and cooled parts of the thermosyphon loop are connected by perfectly insulated channels ($s_1 \leq s \leq s_3$, $s_4 \leq s \leq s_7$, $s_8 \leq s \leq s_{10}$). The space coordinate s circulates around the closed loop as shown in Fig. 1. 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.

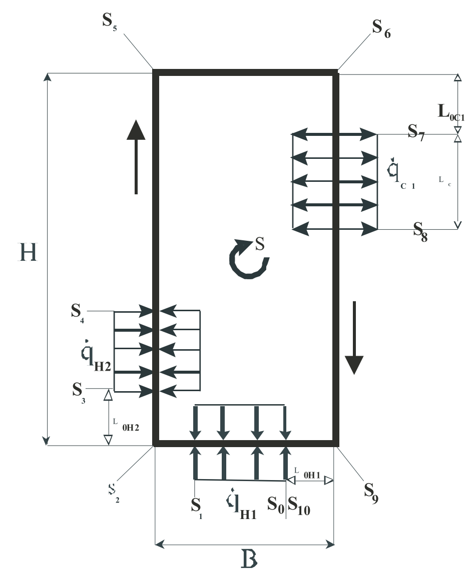

Figure 1. Model of the two-phase thermosyphon loop heated from horizontal and vertical sides and cooled from vertical side.

Superheating and subcooling are neglected and a linearly varying quality $x(s)$ are assumed according to [9].

3 Governing equations

The one-dimensional, steady-state equations governing the vertical loop flow can be written as [6]:

$$\begin{cases} \dfrac{d\dot{m}}{ds} = 0 \;, \\[2ex] -\dfrac{dp}{ds} = \underbrace{\dfrac{U}{A}\tau_w}_{\text{FRICTION TERM}} + \underbrace{\dfrac{\dot{m}}{A}\dfrac{d}{ds}\left[\dfrac{(1-x)^2}{(1-\alpha)\,\rho_L} + \dfrac{x^2}{\alpha\rho_V}\right]}_{\text{ACCELERATION TERM}} + \underbrace{\beta g\left[(1-\alpha)\,\rho_L + \alpha\rho_V\right]}_{\text{GRAVITATION TERM}} \;, \\[4ex] \dfrac{d}{ds}\left(\dfrac{\dot{m}}{A}h\right) = \begin{cases} 0 & \text{for insulated regions,} \\ \frac{U}{A}\dot{q} & \text{for heated and cooled regions,} \end{cases} \end{cases} \tag{1}$$

where: $\beta = 0$, for $\vec{e}\perp\vec{g}$; $\beta = (+1)$ for $\vec{e}\uparrow \wedge \vec{g}\downarrow$; $\beta = (-1)$ for $\vec{e}\downarrow \wedge \vec{g}\downarrow$.

In order to eliminate the pressure gradient, the momentum term in Eq. (1) is integrated around the loop:

$$\oint\left(\frac{dp}{ds}\right)ds = 0\ ,: \quad p(0) = p(L)\ . \tag{2}$$

In the case of a loop with constant cross-section area, integrating the acceleration term around the loop gives zero and the balance takes place only between the friction and gravity forces:

$$\oint\left(\frac{\Delta p}{\Delta s}\right)_{2f,T} ds\ +\ \oint\left(\frac{\Delta p}{\Delta s}\right)_{2f,H} ds = 0\ . \tag{3}$$

3.1 Gravitational term in momentum equation

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

$$\begin{aligned}&\oint\{\beta g\rho\}\,ds = g\left(\rho_V-\rho_L\right)\big\{(s_3-s_2)\,\bar{\alpha}_{\langle s_1;s_3\rangle}+(s_4-s_3)\,\bar{\alpha}_{\langle s_3;s_4\rangle}+\\&+\left[(s_5-s_4)-(s_7-s_6)\right]\bar{\alpha}_{\langle s_4;s_7\rangle}-(s_8-s_7)\,\bar{\alpha}_{\langle s_7;s_8\rangle}\big\}=0\ ,\end{aligned} \tag{4}$$

where

$$\bar{\alpha}_{\langle s_P;s_K\rangle} = \frac{1}{(s_K-s_P)}\int_{s_P}^{s_K}\alpha_{\langle s_P;s_K\rangle}(s)\,ds\ . \tag{5}$$

3.2 Frictional term in momentum equation

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 [6]:

$$\frac{U}{A}\tau_w = \left(\frac{-dp}{ds}\right)_{2p} = R\left(\frac{-dp}{ds}\right)_{L0}\ , \tag{6}$$

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

$$\begin{aligned}&\oint\left(\frac{U}{A}\tau_w\right)ds = \left(\frac{dp}{ds}\right)_{L0}\big\{(s_1-s_0)\,\bar{R}_{\langle s_0;s_1\rangle}+(s_3-s_1)\,\bar{R}_{\langle s_1;s_3\rangle}+\\&(s_4-s_3)\,\bar{R}_{\langle s_3;s_4\rangle}+(s_7-s_4)\,\bar{R}_{\langle s_4;s_7\rangle}+(s_8-s_7)\,\bar{R}_{\langle s_7;s_8\rangle}+(s_{10}-s_8)\big\}\ ,\end{aligned} \tag{7}$$

where

$$\bar{R}_{\langle s_P;s_K\rangle} = \frac{1}{(s_K - s_P)} \int\limits_{s_P}^{s_K} R(s)\, ds \; . \tag{8}$$

Substituting Eqs. (4) and (7) into the momentum equation (1) gives

$$\begin{aligned} &\left(\frac{dp}{ds}\right)_{L0} \left\{ (s_1 - s_0)\, \bar{R}_{\langle s_0;s_1\rangle} + (s_3 - s_1)\, \bar{R}_{\langle s_1;s_3\rangle} + (s_4 - s_3)\, \bar{R}_{\langle s_3;s_4\rangle} + \right. \\ &\left. + (s_7 - s_4)\, \bar{R}_{\langle s_4;s_7\rangle} + (s_8 - s_7)\, \bar{R}_{\langle s_7;s_8\rangle} + (s_{10} - s_8) \right\} + \\ &+ g\,(\rho_V - \rho_L) \left\{ (s_3 - s_2)\, \bar{\alpha}_{\langle s_1;s_3\rangle} + (s_4 - s_3)\, \bar{\alpha}_{\langle s_3;s_4\rangle} + \right. \\ &\left. + \left[(s_5 - s_4) - (s_7 - s_6)\right] \bar{\alpha}_{\langle s_4;s_7\rangle} - (s_8 - s_7)\, \bar{\alpha}_{\langle s_7;s_8\rangle} \right\} = 0 \; . \end{aligned} \tag{9}$$

The El-Hajal's empirical correlation for the void fraction at low pressures for separate model of two phase flow in minichannels is applied [6]. The local two-phase friction coefficient in two-phase adiabatic region was calculated using the Zhang-Webb formula [6]. In two-phase diabatic regions the local two-phase friction coefficient was calculated using the Tran formula [6].

4 Minichannels. Distributions of the mass flux rate and heat transfer coefficient in flow boiling and condensation

The mass flux rate distribution $\dot{G}$ versus heat flux $\dot{q}_H$ was obtained numerically for the steady-state conditions for separate model, as is shown in Fig. 2. R11 was used as a working fluid ($t = 20$ °C , $p = 0.895\times10^5$ Pa) [5].

The heat transfer coefficient in flow boiling for minichannels α_{TPB} distributions versus heat flux $\dot{q}_{h1}$ in evaporator ($s_0 \leq s \leq s_1$) were obtained numerically with Mikielewicz [10] and Saitoh [12] correlations (Tab. 1). The results are presented for minichannels in Fig. 3.

Table 1.

MIKIELEWICZ (2007)	SAITOH (2007)
$\frac{\alpha_{TPB}^{JM}}{\alpha_{REF}} = \sqrt{(R_{M-S})^n + \frac{1}{1+P}\left(\frac{\alpha_{PB}}{\alpha_{REF}}\right)^2}$	$\alpha^{SAITOH} = F \cdot \alpha_L + S \cdot \alpha_{POOL}$

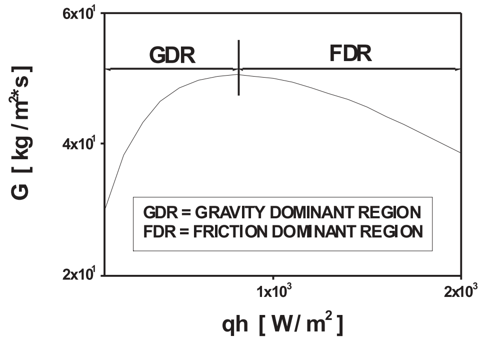

Figure 2. Mass flux rate $\dot{G}$ as a function of $\dot{q}_H$ ($L = 0.2$ m, $B = 0.04$ m, $H = 0.06$ m, $D = 0.002$ m).

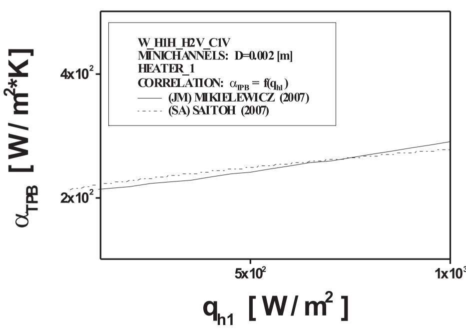

Figure 3. Heat transfer coefficient α_{TPB} as a function of $\dot{q}_{h1}$.

The heat transfer coefficient in condensation for minichannels α_{TPC} distributions versus heat flux $\dot{q}_C$ in condenser ($s_6 \leq s \leq s_7$) was obtained numerically with Mikielewicz [11], Shah [6] correlations (Tab. 2). The results are presented for minichannels in Fig. 4.

Table 2.

MIKIELEWICZ (2007)	SHAH (1979)
$\frac{\alpha_{TPC}^{JM}}{\alpha_{REF}} = \sqrt{(R_{M-S})^n}$	$\alpha_{TPC}^{SHAH} = \alpha_{L0} \times \left[(1-x)^{0.8} + \frac{3.8x^{0.76} \cdot (1-x)^{0.04}}{(P_{red})^{0.38}}\right]$

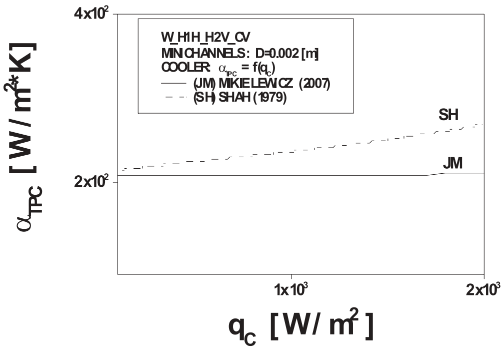

Figure 4. Heat transfer coefficient α_{TPC} as a function of $\dot{q}_C$.

5 Conclusions

The results show that the one-dimensional two-phase separate model can be used to describe heat transfer and fluid flow in the thermosyphon loop heated from horizontal and vertical sides and cooled from horizontal side for minichannels. 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 the thermosyphon loop with minichannels the following correlations have been used: the El-Hajal [6] correlation for void fraction, the Zhang-Webb [6] correlation for the friction pressure drop of two-phase flow in adiabatic region, the Tran [6] correlation for the friction pressure drop of two-phase flow in diabatic region and the Mikielewicz [10] and Saitoh (2007) [12] correlations for the heat transfer coefficient in evaporator, Mikielewicz [11] and Shah [6] correlations for the heat transfer coefficient in condenser. R11 was used in calculations as the working fluid.

The distribution of mass flux rates against heat flux approach a maximum and then decrease in case of minichannels. Two flow regimes can be clearly identified: GDR – gravity dominant regime and FDR – friction dominant regime, as it is shown in Fig. 2. The heat transfer coefficient in flow boiling increases with an increasing heat flux for minichannels (Fig. 3). The heat transfer coefficient in condensation slowly increases with an increasing heat flux for minichannels (Fig. 4).

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