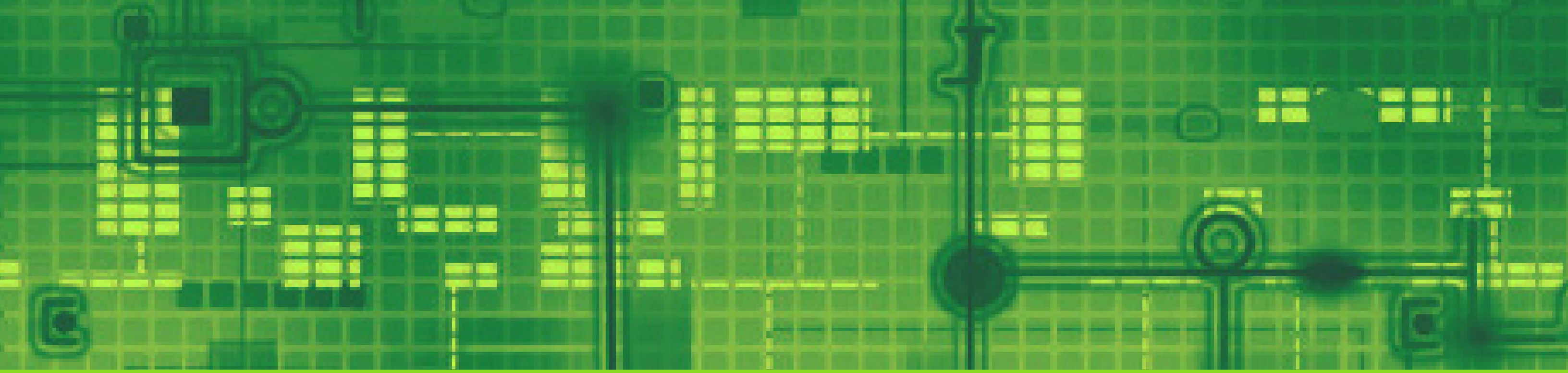

Theoretical Cooling Limit

of a Thermosyphon

Introduction

Passive, two phase liquid cooling schemes for thermal management of electronics rely on heat pipes and thermosyphons. Thermosyphons are passive cooling devices able to overcome the capillary limitation of heat pipes by using gravitationally induced buoyancy to drive the flow. There are several configurations in which thermosyphons can be deployed [1].

This article reviews an analytical approach for finding the theoretical cooling limit for a thermosyphon with a flow boiling arrangement. A pressure model was developed and used to predict the critical heat flux (CHF) of such a device [2].

Nomenclature

A_c low area in boiler section

A_S heated area of boiler surface

d_{rise} rising tube inner diameter

d_{fall} falling tube inner diameter

g gravitational acceleration

H liquid level in reservoir

H_{fg} latent heat of vaporization

L square dimension of boiling surface

L_1, L_2, L_3 lengths of tubing sections

$\dot{m}$ mass flow rate

P pressure

P_o atmospheric pressure

ΔP pressure drop

ΔP_A acceleration pressure drop

ΔP_F frictional pressure drop

ΔP_G gravitational pressure drop

L_e length of evaporator section, m

q" heat flux

q''_m critical heat flux

$r_{arena} = (A_{boilerinletcrosssection}/A_{boileroutletcrosssection})^2$ square ratio of boiler inlet cross section area to boiler outlet cross section area

T temperature

x_L fluid quality at boiler exit

α void fraction

ρ density

Thermosyphon Modeling

Several types of thermosyphons are presented in Figure 1 [1]. A two phase thermosyphon may be constructed as a single pipe or as a thermosyphon loop. In a thermosyphon loop, the evaporator and condenser sections are linked by tubes for liquid and vapor circulation.

Figure 2 presents a thermosyphon using a flow boiling arrangement in the evaporator section. In this configuration, the heat generating component is placed vertically in a relatively narrow channel to allow for a quick vapor leave and liquid replenish. This kind of system, with water as the working fluid, reached a critical heat flux (CHF) of 146 W/cm^2 [3].

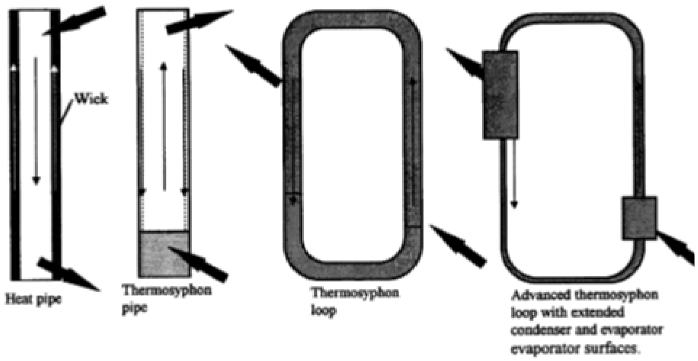

Figure 1. Heat Pipe and Different Types of Thermosyphons [1]

A model for the flow boiling type of thermosyphon was developed in order to determine the maximum cooling density achievable for a certain geometrical configuration and working fluid. Figure 3 illustrates the simplified thermosyphon loop used to construct the analytical model. Three types of pressure drops are considered: frictional, gravitational and pressure drop due to flow acceleration. The latter occurs only in the boiler section of the system. The pressure drops associated with the geometrical changes in the loop are neglected. The pressure at the free surface of the coolant is held constant around atmospheric pressure (P_o). It is assumed that no heat transfer is taking place from the connecting tubes. Also, it is assumed that the liquid in the falling tube is saturated.

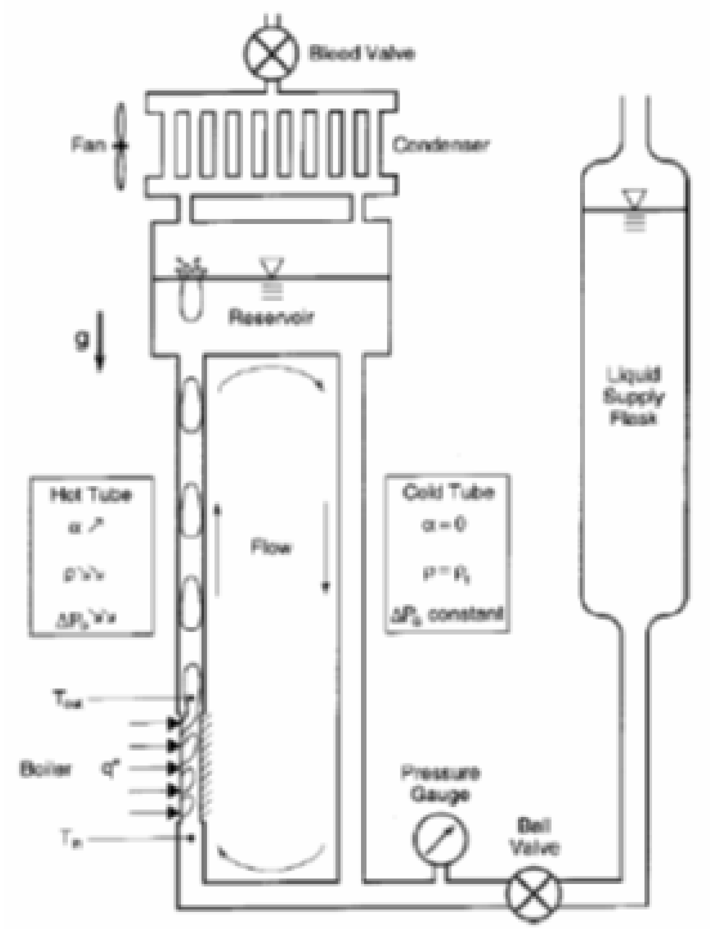

Figure 2. Schematic of a Flow Boiling Thermosyphon [2]

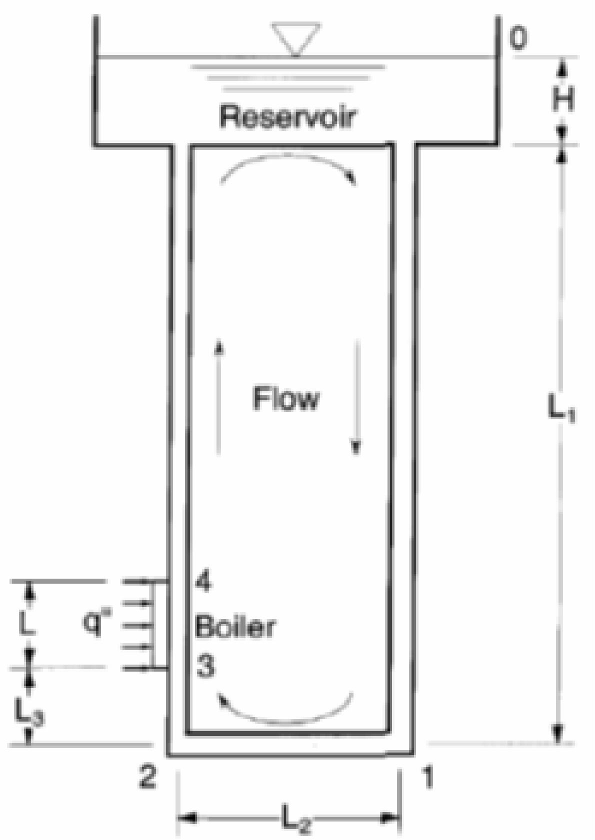

Figure 3. Modeled Configuration [2]

L_1 (m)	0.45
L_2 (m)	0.06
L_3 (m)	0.04
H (m)	0.05
d_{rise} (mm)	6.35
d_{fall} (mm)	6.35
L (m)	0.0213
As (cm^2)	4.55

Table 1. Geometrical Parameters of Study [2]

The pressure drop across the sections can be expressed as:

$$P_0 - P_1 = \Delta P_{F,0\rightarrow1} - \rho_f g(H + L_1) \quad (1)$$

$$P_1 - P_2 = \Delta P_{F,1\rightarrow2} \quad (2)$$

$$P_2 - P_3 = \Delta P_{F,2\rightarrow3} + \rho_f g L_3 \quad (3)$$

$$P_3 - P_4 = \Delta P_{F,3\rightarrow4} + \Delta P_{A,3\rightarrow4} + \Delta P_{G,3\rightarrow4} \quad (4)$$

$$P_4 - P_0 = \Delta P_{F,4\rightarrow0} + \Delta P_{G,4\rightarrow0} \quad (5)$$

$$\Delta P = {}_{F,4\rightarrow0} + \rho_{fg} (H + L_1 - L_3) = \Delta P_{F,0\rightarrow1} + \Delta P_{F,1\rightarrow2} + \Delta P_{F,2\rightarrow3} + \Delta P_{F,3\rightarrow4} + \Delta P_{F,4\rightarrow0} + \Delta P_{A,3\rightarrow4} + \Delta P_{G,3\rightarrow4} + \Delta P_{G,4\rightarrow0} \quad (6)$$

Appropriate expresions have been developed for all of the above pressure terms [2].These expressions were incorporated in an EES model used to predict the cooling performance of the thermosyphon for a variety of scenarios.

Results

First, the model was used to predict the maximum heat flux of the thermosyphon. Using water as the working fluid, a theoretical critical heat flux (CHF) of 535 W/cm^2 was predicted. However, the corresponding CHF was only 146.3 W/cm^2. The limiting factor is postulated to be the boiling phenomenon at the heat dissipating component surface. If boiling can be enhanced, significant improvement in CHF can be attaind.

Second, the model was used in the preliminary design of the thermosyphon. Figure 4 shows the variation of mass flow rate with heat flux for a heat flux range from 5 to 530 W/cm^2.

After an initial increase with the increase in heat flux, the mass flow rate begins to decrease. The explanation could reside in the fact that with increased heat flux, more vapor is generated by the boiler and the pressure drop due to flow acceleration becomes significant (Figure 5). Since the total pressure head is maintained constant, other pressure drop terms must decrease, which is realized by decreasing the mass flow rate through the system.

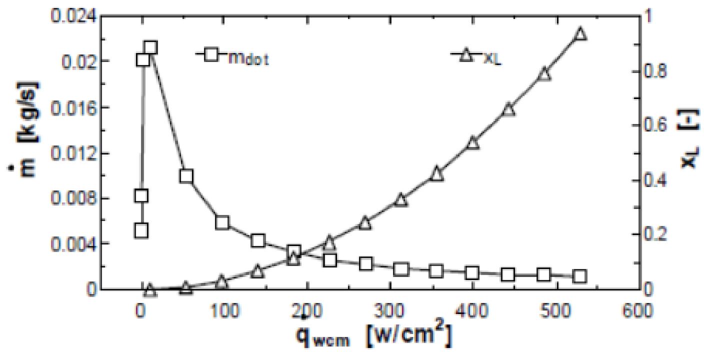

Figure 4. Variation of Mass Flow Rate with Heat Flux [2]

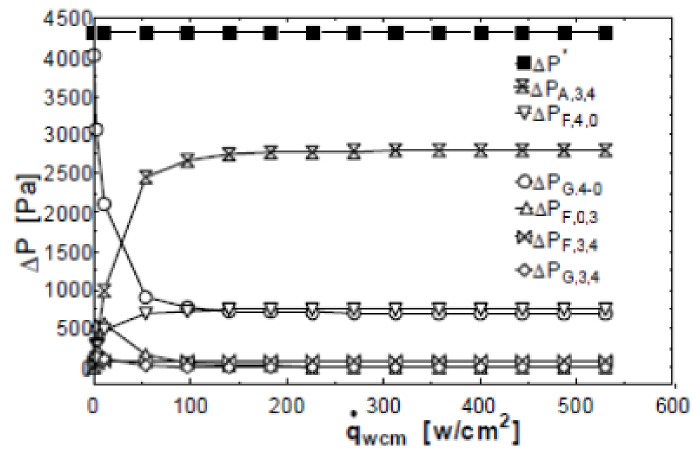

Figure 5. Pressure Drop at Different Heat Fluxes [2]

Figure 5 also indicates that $P_{A,3-4}$ is the largest pressure drop term. Based on this observation, it was concluded that it is worthwhile to study the effect of the boiler cross section area on the CHF. Figure 6 presents this effect.

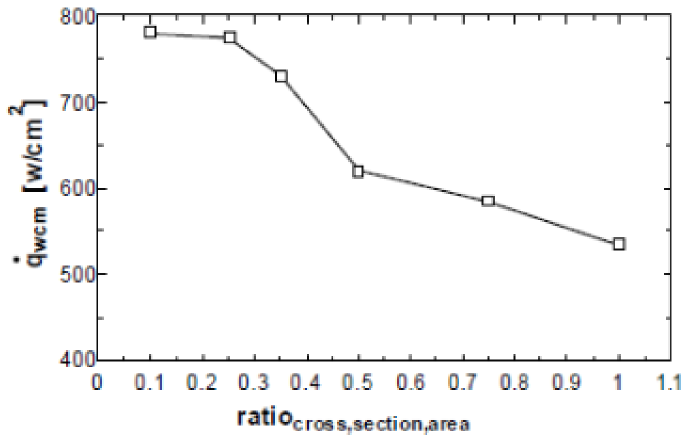

Figure 6. Effect of Boiler Cross Section on CHF [2]

Decreasing r_{area} from 1 to 0.1 (corresponding to outlet boiler gap size being about 3.3 times inlet boiler gap size) CHF increases from 530 W/cm^2 to 780 W/cm^2, a significant jump. Another significant pressure drop term is $P_{F,4-0}$ (Figure 5). This prompted the investigation of the effect of increasing the rising tube diameter on CHF. The tube diameter was increased from 6.35 mm to 11.95 mm in several steps. The results are captured in Figure 7.

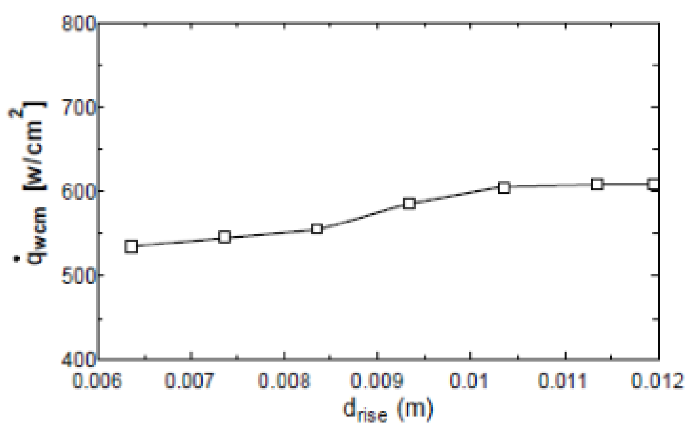

Figure 7. Effect of Rising Tube Diameter on CHF [2]

It can be seen that the increase in CHF, although present, is not significant and it levels off for tube diameters beyond 11.35 mm.

Conclusions

A pressure model was developed for a particular type of thermosyphon. The thermosyphon has a flow boiling configuration in the evaporator section, which results in superior thermal performance. The model was used to predict the theoretical cooling limit of the device as well as for preliminary thermosyphon design. The CHF value predicted by the model is far above the experimental data. In addition, enlarging the boiler cross-section area along the boiling surface can increase the maximum cooling density. Increasing the rising tube diameter does not have a significant effect on the CHF.

The model developed in [2] is a useful tool for parametrically studying of maximum thermal transport capacity of a flow boiling thermosyphon. However, the authors need to explain how the pressure drop expressions are used for critical heat flux prediction. Also, the model needs further refinement and validation. The predicted CHF values are way above measured data and the discrepancy has to be well understood in order for the model to become a trusted prediction tool.

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1. Palm, B., " A Short Course on Cooling of Electronics by Heat Pipes and Thermosyphons ", Nokia, Helsinki, November 1996.

2. Zhang, J., and Hugenroth, J.," Exploration of the Theoretical Limit of Thermosyphon Cooling System".

3. Mukherjee, S., and Mudawar, I., " Pumpless Loop for Narrow Channel and Micro-Channel Boiling", Journal of Electronics Packaging, Vol.125, pp. 431-441, 2003.

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