Cascade Thermosiphon Loop Module for Data Center Cooling

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Abstract

With an increasing trend of energy consumption required to cool data centers, designing and implementing efficient and cost-effective cooling schemes is the most urgent issue for sustainable operation of data centers. A theoretical concept of cascade thermosiphon loop module as an efficient cooling scheme for data centers is examined in the project. A cascade thermosiphon loop module which is a thermosiphon loop with an additional liquid supply path at the middle section of the evaporator is able to enhance boiling heat transfer due to induced flow boiling, and it can maintain uniform thermal performance along the heater in vertical orientation. In order to estimate the performance of the cooling loop, heat transfer and pressure drop model are constructed. Based on the model developed, proper channel dimensions are selected for boiling enhancement and preventing early dry-out. Results show that the wall superheat decreases by 15K when the cascade thermosiphon loop is implemented compared to the pool boiling in vertical orientation, and cascade design can reduce the deviation of heat transfer coefficient by 35% in comparison with the thermosiphon loop without the liquid supply path.

Keywords: Thermosiphon, Boiling heat transfer, Pool boiling, Confinement boiling, Immersion cooling, Data center cooling

1. Introduction

1.1. Background

One of the most probable future image of data centers would be tremendous energy-absorbing facilities created by humankind. In the U.S., data centers consumed more than 100 bilion KWh in 2011. It is not the only problem of certain countries. In a global perspective, the energy consumed by the data center operation accounted for 1.5% of the total energy consumption of the world, which is equivalent to the amount of energy spent by 25,000 American households [1]. This trend is just the beginning of the early stage of rapid growth of the data centers and it is clear that the energy consumption rate for data centers keeps increasing in the future.

Among a significant amount of energy consumed by data centers, 50-75 percent of energy is consumed to cool down high performance computing devices [2]. Conventionally, data centers have been using air cooling schemes, which circulate the cold air flow through servers mounting computing devices. However, as the power density of computing devices increases, conventional air cooling schemes cannot meet the cooling requirement anymore. As an alternative, liquid cooling and two-phase cooling scheme are viable solutions for cooling data centers.

Immersion cooling scheme is one of promising approaches for data center cooling since it does not require additional pumping power and has high heat transfer coefficient. However, immersion cooling scheme still has problems. Typical servers cooled by immersion cooling scheme have vertical arrangement of computing devices to save space and to benefit from enhancement of bubble detaching driven by buoyancy. This vertical arrangement of computing device is susceptible to temperature non-uniformity since bubbly flow generated from computing devices located at lower part of a server can negatively affect cooling performance of computing devices located at upper part. Also, conventional immersion cooling schemes require excessive fluid volume compared to the volume of servers itself. Dielectric working fluids used in immersion cooling scheme typically cost high, and it implies that immersion cooling scheme can be an inefficient way when the total const of equipment is considered.

1.2. Problem definition

To improve the existing problems of immersion cooling scheme, a new cooling module for data centers should be designed with consideration of geometric constraints as follows. First, a new cooling module should be able to maintain uniform cooling performance on each vertically arranged computing device as shown in **Figure 1**.

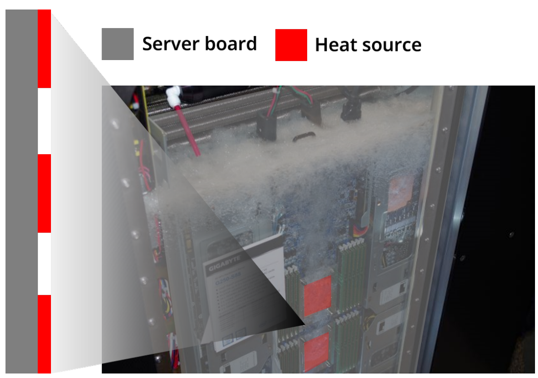

Figure 1. Vertically arranged heat sources

Second, a new cooling module should be capable to sustain cooling performance in narrow gaps between server boards. A lot of data centers nowadays require bulky space for facilities and it becomes one of the main concerns when building a new data center. Thus, the new cooling module should allow compact design of servers and racks, and to achieve this design requirement, it should be able to adopt the cooling module in narrow space.

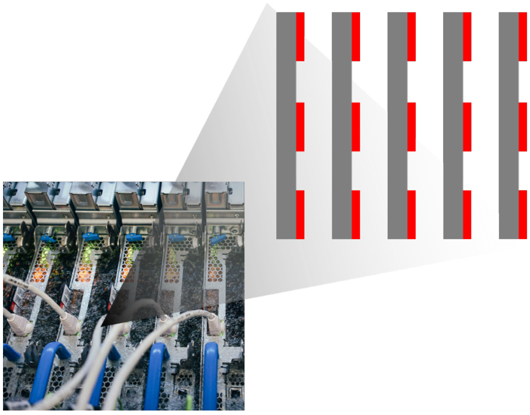

Figure 2. Narrow gap between each server board

1.3. *Design concept of cascade thermosiphon loop*

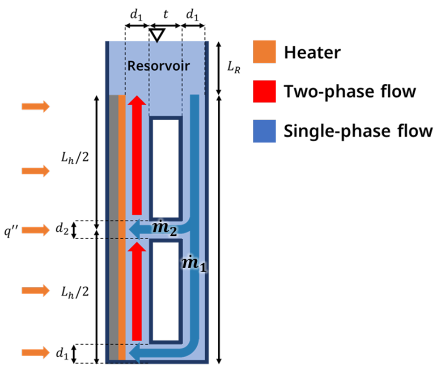

Figure 3. Schematic of the cascade thermosiphon loop

Mukherjee and Mudawar [3] proposed the concept of pumpless cooling loop based on the working principle of thermosiphon. In the cooling loop, due to the phase change occurring in the evaporator section, hydrostatic pressure difference between the evaporator and condensor drives liquid circulation. The authors reported that the liquid circulation induces flow boiling and increases CHF as the tube diameter of the evaporator section decreases. This concept of cooling loop will be adopted in this project since it can enhance boiling heat transfer in the limited space and working fluid volume.

In addition to the liquid circulation, to maintain uniform thermal performance along the heater, additional liquid supply path is made in the middle section of the evaporator. This additional liquid supply is expected to be beneficial for maintaining uniform thermal performance as it can prevent a large deviation of quality along the heater. In this project, an optimum number of liquid supply path will not be discussed for simplicity, instead design guidelines for channel height d_1 and d_2 will be suggested.

2. Theoretical Modeling

2.1. *Pressure drop model of cascade thermosiphon loop*

To estimate mass flow rates passing through each channel of the cascade thermosiphon loop for a given geometry, a pressure drop model for the cascade thermosiphon loop is suggested. The pressure drop model consists of two pressure balance equations, and based on this model, two unknown mass flow rates in each channel of the cascade thermosiphon loop can be determined.

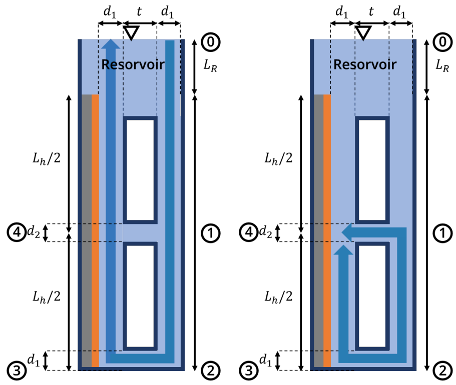

Figure 4. Flow path schematic for two pressure equations

The first pressure equation is obtained with the pressure balance between hydrostatic pressure difference and other pressure losses including frictional, gravitational, and accelerational pressure drop. Detailed expression for each pressure drop component is given in appendix.

$$P_0 - P_1 = \Delta P_{fr,0-1} - \rho_f g(L_R + L_h/2) \quad (1)$$
$$P_1 - P_2 = \Delta P_{fr,1-2} - \rho_f g(L_h/2) \quad (2)$$
$$P_2 - P_3 = \Delta P_{fr,2-3} \quad (3)$$
$$P_3 - P_4 = \Delta P_{fr,3-4} + \Delta P_{A,3-4} + \Delta P_{G,3-4} \quad (4)$$
$$P_4 - P_0 = \Delta P_{fr,4-0} + \Delta P_{A,4-0} + \Delta P_{G,4-0} \quad (5)$$

By adding the equations (1)-(5), the pressure balance between hydrostatic pressure difference and other pressure losses is given as follows.

$$\begin{aligned}\Delta P^* &= \rho_f g(L_R + L_h)\\ &= \Delta P_{fr,0-2} + \Delta P_{fr,2-3} + \Delta P_{fr,3-0} + \Delta P_{A,3-0} + \Delta P_{G,3-0}\end{aligned}$$

The second pressure equation is obtained with the pressure balance between node 1 and 4.

$$\begin{aligned}\Delta P_{fr,sp,1-4} = \Delta P_{fr,sp,1-2} - \rho_f g(L_h/2) + \Delta P_{fr,sp,2-3}\\ + \Delta P_{fr,tp,3-4} + \Delta P_{A,3-4} + \Delta P_{G,3-4}\end{aligned}$$

Each pressure drop term can be expressed by using mass flow rates, quality, and geometric parameters. Pressure drop equations are non-linear equations in terms of mass flow rates, and if the geometry of the loop is determined, the mass flow rates can be calculated by using the numerical non-linear equation solver.

2.2. Design parameters

Design parameters of a cascade thermosiphon loop are given in **Table 1**. Here, L_h, t, w, L_R indicate the length of the heater, wall thickness of the cooling module, width of the heater, and height of the reservoir, respectively. These specifications are determined by considering geometry of a commercial server board.

L_h [m]	t [m]	w [m]	L_R [m]
0.8	0.01	0.45	0.05

Table 1. Geometry of the heater and the cooilng module

Among them, channel height d_1 and d_2 are the most important parameters since they are directly related to the mass flow rate through the loop. Thus, throughout this project, channel height d_1 and d_2 will be decided and thermal performance evaluation will be conducted based on the geometry. Commericial dielectric fluid HFE-7100 is used as a working fluid, and all material properties are used at saturation temperature of atmospheric pressure. Applied heat flux is assumed 20 W/cm^2.

3. Results and Discussion

3.1. Mass flow rates depending on channel geometry

When the channel geometry is given, mass flow rates can be estimated by the theoretical model. When the channel height d_2 is fixed at 5mm, **Figure 5** shows relative pressure drop for each frictional, gravitational, and accelerational pressure drop components. At relatively low channel height d_1, accelerational pressure drop and frictional pressure drop dominate the total pressure drop. As the channel height d_1 increases, gravitational pressure drop increases.

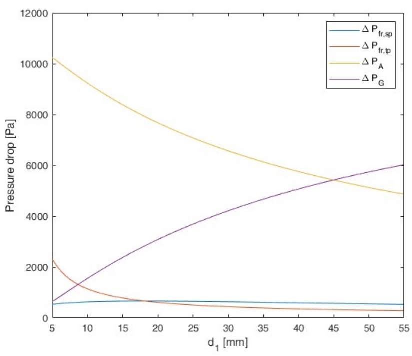

Figure 5. Relative magnitude of pressure drop components

As the channel height d_1 increases, the total mass flow rate also increases since the hydrostatic pressure gradient becomes larger, which is the main driving force for this loop. Channel height d_2 is set as 5mm since it has negligible effects on the total mass flow rate. For an optimum operation for this loop, optimum channel height d_1 should be determined considering the quality and flow boiling heat transfer; however, optimization process will not be discussed in this project due to the limited time.

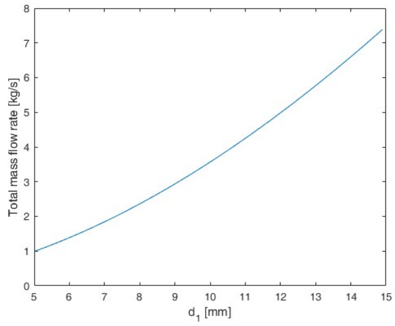

Figure 6. Total mass flow rate depending on d_1

3.2. Local heat transfer coefficient

Local heat transfer coefficient can be estimated after the local quality is calculated from the given geometry and operating conditions. When the channel height d_1 is 7mm, the exit quality is maintained about 0.4 and the total thickness of this module becomes below 3 cm. Since the total thickness of the module should be thin as much as possible, narrower channel height design can be taken into account; however, it results in comparably high exit quality. Thus, the design parameter d_1 is chosen as 7mm. When the channel height d_2 is decided as 5mm, mass flow rates and exit quality is given in **Table 2**.

q'' [W/cm^2]	$\dot{m}_1$ [kg/s]	$\dot{m}_2$ [kg/s]	Quality
20	1.213	0.584	0.445

Table 2. Operating condition of the cascade thermosiphon loop

By using the estimated mass flow rates and local quality, local heat transfer coefficient along the heater length can be calculated. Local flow boiling heat transfer in the cascade thermosiphon loop can be predicted by the correlation proposed by Gungor and Winterton [4].

$$h_{G-W} = h_l \left[1 + 3000 Bo^{0.86} + \left(\frac{x}{1-x} \right)^{0.75} \left(\frac{\rho_l}{\rho_v} \right)^{0.41} \right]$$

$$\text{where } Bo = \frac{q''}{G h_{lv}}$$

Local heat transfer was compared with the case of single-loop case, which has no additional mass flow rate in the middle section of the heater. Local heat transfer coefficient deviation of the cascade thermosiphon loop is 35% smaller than the single-loop case. It implies more uniform heat transfer performance can be achieved in the case of cascade thermosiphon loop due to the additional liquid supply along the heater length and reduced local quality deviation.

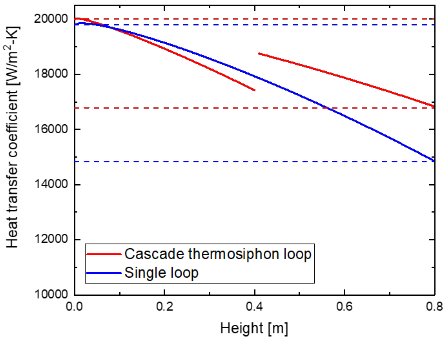

Figure 7. Caption

3.3. *Comparison with vertical oriented pool boiling situation*

Heater temperature was also compared with the case of vertical oriented pool boiling. Mohamed and Huseyin [5] conducted pool boiling experiment with the working fluid HFE-7100 and for various heater surface orientation. Estimated nucleate boiling heat flux was expressed as a function of wall superheat. By using the relation between wall heat flux and wall superheat, the estimated wall superheat for vertical oriented pool boiling is 25K.

$$q'' = \left[(0.25 \times 10^{-4} \Delta T_{sat}^{3.53})^{-8} + (1.9 \Delta T_{sat}^{0.71})^{-8}\right]^{1/8}$$

By integrating the local heat transfer coefficient of the cascade thermosiphon loop, the average heat transfer coefficient is obtained as 18,335 W/m^{2}K. Resulting wall superheat for the cascade thermosiphon loop is 10.8K, which is 15K smaller than the case of pool boiling.

4. Conclusion

1. Theoretical pressure drop model of a cascade thermosiphon loop is suggested and based on the model, pressure drop components are estimated. The result shows that at relatively smaller channel height d_1 accelerational and frictional pressure drop are the principal pressure drop components. On the other hand, at relatively larger channel height d_2, gravitational pressure drop grows rapidly.
2. Channel height d_1 and d_2 are decided as 7mm and 5mm by considering the exit quality and compact design requirement. Induced total mass flow rate is estimated as 1.795 kg/s.
3. Based on the proposed design, the local heat transfer coefficient distribution is estimated by using the flow boiling heat transfer coefficient. Due to the additional liquid supply in the middle section of the heater, sudden increase in the local heat transfer coefficient is observed. The difference between the lowest local heat transfer coefficient and the highest local heat transfer coefficient reduces by 35% compared to the case of single-looped thermosiphon.
4. Estimated average wall superheat is 10.8K in the case of cascade thermosiphon loop. The amount of wall superheat is 15K smaller than the case of pool boiling and this result implies that the cascade thermosiphon design can significantly enhance thermal performance of immersion cooling scheme.

Appendix A. Research area

My research area is focused on the thermal and hydraulic modeling of embedded liquid cooling device for semiconductors. Although I have been dealing with the case of single-phase cooling device, I have a lot of interests in two-phase cooling schemes and phase change phenomena.

Appendix B. Answer for the questions

Q1. Is the cascade thermosiphon structure always beneficial?

A: I think induced flow usually enhance the heat transfer performance; however, in the modeling, local flow pattern around the bridge channel was not considered. It means, for certain designs, liquid supply might prevent bubbles from escaping the heater surface. This can deteriorate the thermal performance of this module.

Q2. Would it be possible to validate the model using numerical simulation?

A: Conventional loop heat pipe or loop thermosiphon were largely investigated by numerical models, and the cascade thermosiphon loop has similar working mechanisms with them. So I think it can be validated with the numerical model too.

Q3. If the channel diameter decreases to order of 100m, would it be possible to use capillary effect?

A: Great point of view. Initially, I wondered confined channel effect or capillary effect at very small channel diameters. Of course, capillary effect can induce additional liquid supply but early dry-out or moderate total pressure drop should also be taken into account.

Q4. Is there the optimum number of channels for the cascade design?

A: That's very insightful question and it would be the future research scope. When we think of some limiting cases such as the case when the number of channels goes to infinity, it becomes almost similar to pool boiling situation. For those point of view, I believe there might be the optimum number of channels.

Q5. Is there any reason for using different correlations for single-loop thermosiphon and cascade thermosiphon?

A: Actually, I used the same correlation for the single-loop thermosiphon loop and cascade thermosiphon loop. Maybe you were confused about the comparison with the pool boiling case. You can find the correlation for predicting wall superheat in the case of pool boiling in the reference.

Q6. Isn't there back flow at the branch?

A: When I estimated local pressure drop in each point, back flow was not observed since there wasn't adverse pressure gradient across the flow path. However, when we think of vigorous boiling situation, and when the channel height becomes very narrow, back flow should be considered as you mentioned it.

Q7. Why does the local heat transfer coefficient suddenly increase in the graph?

A: In the cascade thermosiphon loop design, there is an additional liquid supply at the branch. It reduces local quality and increases mass flux. Thus, the sudden increase in local heat transfer coefficient appear in the graph.

Q8. Could you explain more about how did you get the exit quality, 0.44?

A: After we estimate mass flow rates in each branch through the pressure drop model, exit quality can be estimated by using the energy balance equation. Of course, there wasn't any specific constraint for the exit quality, so we have to check whether it is a safe condition for normal operation. In this project, I just assumed this exit quality is safe from CHF.

Q9. Isn't HFE-7100 a classical refrigerant? Why doesn't it follow the usual nomenclature e.g. R-134a?

A: HFE-7100 is lately developed for the purpose of non-toxic refrigerant for environment. There are also several other refrigerants being developed. HFE-7100 consists of different chemicals compared to conventional refrigerants, so their nomenclature is different from the nomenclature starting from R.

Q10. Why is the local heat transfer coefficient lower at the beginning of the heater section?

A: Since the total mass flow rate in each case is equivalent, the mass flow rate at the beginning of the heater section is smaller in the case of cascade thermosiphon loop. Please recall that there is an additional branch, and the fraction of mass flow rate passes through the branch. So locally, the heat transfer coefficient is somewhat smaller.

Q11. Why did you set the channel height d_1 equal at the inlet and the bottom section?

A: Of course, you can set those channel height differently. That was just the simplification of the geometry.

Q12. Is there any tendency for the location of the branch or the number of branches?

A: Unfortunately, I couldn't conduct the parametric study for the location or the number of branches. But if the location of the branch becomes higher from the bottom, the hydrostatic pressure gradient was not enough to generate the liquid flow, and it implies the effect of liquid supply on the thermal performance is getting weaker. For the number of channels, When we think of some limiting cases such as the case when the number of channels goes to infinity, it becomes almost similar to pool boiling situation. For those point of view, I believe there might be the optimum number of channels.

Q13. Could you explain more about the key idea?

A: The key idea is quite similar to the working principle of conventional loop thermosiphon. When the boiling is initiated in the evaporator section, due to the density difference between the evaporator section and the condenser section, hydrostatic pressure gradient occurs. This pressure difference induces the flow. You can also refer to the reference [3], [6].

Q14. Is there any other way to solve the problem of non-uniform temperature distribution?

A: If there isn't a constraint for orientation, horizontal set-up might be helpful for reducing the local temperature difference.

Q15. Is there any downside of drawback to apply cascade design?

A: Of course, in the model, local flow pattern was not considered. So there might be an unexpected flow instability or early dry-out especially when the channel height becomes very narrow. Also, if the bubbly expansion hinders the additional liquid supply, the enhancement in heat transfer might not be large as we expected. Further design guidelines considering these effects should be suggested in the future research.

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