Refrigerated Cooling of Microprocessors with Micro-Evaporation New Novel Two-Phase Cooling Cycles: A Green Steady-State Simulation Code

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Abstract. *The goal of the present study is to propose and simulate the performance of a novel hybrid two-phase cooling cycle with micro-evaporator elements (multi-microchannel evaporators) for direct cooling of the chips and auxiliary electronics on blade server boards. The specific focus was to work with two-phase cooling using dielectric refrigerants, using a compressor or liquid pump to drive the fluid, a micro-evaporator for cooling of the chip and a microfin tube-in-tube condenser for heat recovery, which can reduce the demand of cooling energy by an impressive amount. A green simulation code to design and evaluate the thermodynamic performance of the hybrid cooling cycle, integrated with the micro-evaporator, for two different cycle configurations at steady state conditions is presented. The simulation code was used to design and evaluate the performance of a two-phase cooling cycle applied on a blade server having 32 high performance microprocessors. Two working fluids were considered, namely HFC134a and a new more environmentally friendly refrigerant HFO1234ze, both of which are dielectric fluids. The new HFO1234ze refrigerant is found to have similar cooling characteristics as HFC134a, but with a slight (23%) pumping power penalty (much less than a single-phase water system's pumping power penalty). An evaluation was also made with respect to the pumping power consumption to drive a two-phase HFC134a cooling cycle compared to a water-cooled cycle. The results so far demonstrate that the pumping power consumption of a water-cooled cycle is on the order of 5 times that of a two-phase HFC134a refrigerant cycle. Finally, a cycle utilising a vapor compressor to drive the refrigerant was also considered. This cycle showed much higher power requirements, but compensated by a higher potential for energy recovery due to the higher condensing temperature attained, which means a higher economic value for the energy recovered. Another benefit observed was a lower volume of heat exchanger when compared with the other cases evaluated, which represents a savings in material.*

Keywords*: datacenter, microprocessor, hybrid two-phase cooling cycle, micro-evaporator.*

1. INTRODUCTION

Currently, the most widely used cooling technology is refrigerated air cooling of a data center's numerous servers. According to recent articles published at ASHRAE Winter Annual Meeting at Dallas (January, 2007), typically 40% or more of the refrigerated air flow bypasses the server racks in datacenters. This poor energetic performance in one of industries leading technological sectors is quite startling and motivates the search for a green thermal solution for future generations of high performance servers. The objective is to consume much less energy to operate and cool while also recovering the waste heat rejected by the data center. This is the topic of research addressed here.

The cost of energy to operate a server for 4 years is now on the same order as the first cost to purchase the server itself, meaning that the choice of future servers should be evaluated on their total 4 year cost, not just their first cost. Based on the above issues, thermal designers of data centers and server manufacturers now seem to agree that there is an immediate need to improve the server cooling process by implementing liquid or two-phase cooling directly in the server on the chip level itself, eliminating the poorly performing air as a coolant all together. Therefore, there is a clear need for a detailed design and evaluation of these new cooling strategies in order to arrive at an improved solution. The new cooling technology should provide more efficient heat transfer from the chips, memories, etc. using water-cooled or boiling-cooled elements, eliminating air as a means of heat transfer, while also reducing electrical energy consumption for driving the cooling system by a significant amount. Current data centers consume on the order of 40-45% of their electricity usage for cooling purposes. Since data centers often dissipate on the order of 5 to 15MW of heat, this makes heat recovery an important energetic and environmental issue to consider. Heat recovery will also greatly reduce the CO_2 footprint of the system.

Recent publications show the development of primarily four competing technologies for cooling chips: microchannel single-phase (water) flow, porous media flow, jet impingement cooling and microchannel two-phase flow (Agostini *et al.*, 2007). The first three technologies are characterized negatively for the relatively high pumping power to keep the temperature gradient in the fluid from inlet to outlet within acceptable limits, i.e. to minimize the axial temperature gradient along the chip and its differential expansion with the thermal interface material. Two-phase flow in microchannels, i.e., evaporation of dielectric refrigerants, is a promising medium to long term solution, despite the higher complexity involved. This solution consumes little pumping power (only 1/10 as much as water cooling according to Hannemann *et al.* (2004)), has good temperature uniformity (Agostini *et al.*, 2008), very high heat transfer coefficients when using high aspect ratio microchannels (as high as 270'000W/m^{2}K according to Madhour *et al.* (2010)), and provides high heat flux dissipation (up to 340 W/cm^2 or more according to Park and Thome (2010)). Possible problems with flow instabilities have been resolved using micro-orifices at the channel inlets (Agostini *et al.*, 2007) while the prediction methods of local heat transfer coefficients (Thome *et al.*, 2004; Consolini and Thome, 2010), the critical heat flux (Mauro *et al.*, 2010), and pressure loss (Cioncolini *et al.*, 2009) in the two phase region are still improving.

In this context, the objective of the present study is to propose and analyze potential two-phase cooling cycles able to maintain the temperature of the chip below its upper operating limit (about 85°C) and to recover energy from the cycle's condenser for reuse, such as for heating a building, residence, hospital, district heating, etc. To do this, an in-house integrated simulation code was developed which has the capability to design the components and determine the performance of a hybrid cooling cycle (liquid pump or vapor compressor as the driver of the working fluid) under steady state conditions and for different working fluids. The code is also able to evaluate the performance of the cooling cycles for single-phase and two-phase flow in the micro-evaporators.

The new refrigerant HFO1234ze of Honeywell Inc. is considered here as a potential substitute of HFC134a. This fluid has a "Global Warning Potential" of only 6 against 1410 of HFC134a, i.e. it is considered as an immediate/future replacement for HFC134a. Both HFC134a and HFO1234ze are dielectric fluids and thus compatible with electronics. HFC134a is currently the most widely used refrigerant for refrigeration and air-conditioning systems.

2. LITERATURE REVIEW

Hannemann et al. (2004) have proposed a pumped liquid multiphase cooling system (PLMC) to cool microprocessors and microcontrollers of high-end devices such as computers, telecommunications switches, high-energy laser arrays and high-power radars. According to them, their system could handle applications with 100W heat loads (single computer chip) as well as applications with short time periods of kW heat loads (radar). Their PLMC consists of a liquid pump, a high performance cold plate (evaporator) and a condenser with a low acoustic noise air mover to dissipate the heat in the ambient air. A comparison between a single-phase liquid loop (water) and the system proposed with HFC134a was made for a 200W heat load. The HFC134a system had a mass flow rate, a pumping power and a condenser size that were 4.6, 10 and 2 times smaller than the water-cooled system. The coolant temperature rise was 10^{o}C for the water but negligible for HFC134a. They emphasized the significant benefits from efficiency, size and weight that were provided with the PLMC solution.

Mongia et al. (2006) designed and built a small-scale refrigeration system applicable for a notebook computer. The system included a minicompressor, a microchannel condenser, a microchannel evaporator and a capillary tube as the throttling device and is considered to be the first refrigeration system developed that can fit within the tight confines of a notebook and operate with high refrigeration efficiencies. HC600a (isobutane) was the working fluid, chosen from an evaluation of 40 candidate refrigerants. According to them, HC600a presented the best efficiency at a low pressure ratio and was readily available. Although it is flammable the system only required a very small charge (a few milliliters). For a baseline operating condition, when the evaporator and condenser temperatures and the heat load were 50^{o}C, 90^{o}C and 50W, the coefficient of performance (COP) obtained was 2.25. The COP reached 3.70 when the evaporator and condenser temperatures increased and decreased by 10^{o}C from the baseline conditions and the heat load was reduced to 44W. The small-scale refrigeration system achieved 25-30% of the Carnot efficiency (ideal COP for a Carnot cycle), values comparable with those obtained in today's household refrigerators.

Trutassanawin et al. (2006) designed, built and evaluated the performance of a miniature-scale refrigeration system (MSRS) suitable for electronics cooling applications. Their MSRS had the following components: a commercial small-scale compressor, a microchannel condenser, a manual needle valve as the expansion device, a cold plate microchannel evaporator, a heat spreader and two compressor cooling fans. A suction accumulator to avoid liquid flow to the compressor, an oil filter to return oil to the compressor and guarantee good lubrication, and heat sources to simulate the chips were also installed. HFC134a was the working fluid. System performance measurements were conducted at evaporator temperatures from 10^{o}C to 20^{o}C and condenser temperatures from 40^{o}C to 60^{o}C. The cooling capacity of the system varied from 121W to 268W with a COP of 1.9 to 3.2 at pressure ratios of 1.9 to 3.2. Their MSRS was able to dissipate CPU heat fluxes of approximately 40-75W/cm^2 and keep the junction temperature below 85^{o}C for a chip size of 1.9cm^2. It was concluded that a new compressor design for electronics cooling applications was needed to achieve better performance of the system (the most significant losses occurred in the compressor, which was not designed for the operating conditions of electronics cooling). It was also recommended to study the development of an automatic expansion device and a suitable control strategy for the MSRS.

Thome et al. (2007) surveyed the advances in thermal modeling for flow boiling of low pressure refrigerants in multi-microchannel evaporators for cooling of microprocessors. According to them, multi-microchannel evaporators hold promise to replace the actual air-cooling systems and can compete with water-cooling to remove high heat fluxes, higher than 300W/cm^2, while maintaining the chip safely below its maximum working temperature, providing a nearly uniform chip base temperature (Agostini *et al.*, 2008) and minimizing energy consumption. Variables such as critical heat fluxes, flow boiling heat transfer coefficients and two-phase friction factors were evaluated and characterized as important design parameters to the micro-evaporator for high heat flux applications.

Thome and Bruch (2008) simulated two-phase cooling elements for microprocessors with micro-evaporation. Heat fluxes of 50W/cm^2 and 150W/cm^2 in a micro-evaporator with channels 75µm wide, 680µm high and 6mm long with 100µm thick fins were simulated for flow boiling. The size of the chip was assumed to be 12mm by 18mm and the micro-evaporator was considered with the fluid inlet at the centerline of the chip and outlets at both sides, i.e. a split flow design to reduce the pressure drop but increase the critical heat flux. Results of pumping power, critical heat flux, and junction and fluid temperatures were generated for HFC134a at an inlet saturation temperature of 55^{o}C (chosen to allow for heat recovery). The following conclusions were reached: i) the influence of mass flux on the fluid, chip and wall temperatures was small, ii) for the heat flux of 150W/cm^2, the chip temperature was 70^{o}C or less, i.e. well below its operational limit of 85^{o}C, iii) the junction-to-fluid temperature difference was only 15K for the heat flux of 150W/cm^2, which is lower than that with liquid cooling systems, iv) the fluid working temperature could still be raised by 10K to a junction temperature of 80^{o}C while rejecting heat at 65^{o}C for reuse, and v) the critical heat flux increased with the mass flux and the lower limit was about 150W/cm^2 for 250kg/m^{2}s. The channel width had a significant effect on the wall and junction temperatures, and there was a turning point at about 100µm when considering 1000kg/m^{2}s of mass flux and 150W/cm^2 of base heat flux, at which these temperatures reached a minimum. For the same mass flux and base heat flux, the reduction of channel width also reduced the energy consumption to drive the flow (pumping power).

From a system viewpoint, Thome and Bruch (2008) showed an approximate comparison of performances of liquid water cooling versus two-phase cooling. For the same pumping power consumption to drive the fluids, two-phase cooling allowed the chip to operate about 13K cooler than water-cooling or it could operate at the same junction temperature but consume less pumping power using a lower refrigerant flow rate. The two-phase cooling system appeared to be more energy-efficient than classical air-cooling or direct liquid cooling systems while also exhausting the heat at higher reusable temperatures. Regarding the choice between a pump and a compressor as the driver for a micro-evaporation heat sink system, they emphasized that the choice depends on the economic value of the re-used energy. The system with a compressor is ideal for energy re-use because of the higher heat rejection temperature; however the additional energy consumed by the compressor compared to the pump has to be justified by the re-use application.

Mauro et al. (2010) evaluated the performance of a multi-microchannel copper heat sink with respect to critical heat flux (CHF) and two-phase pressure drop. A heat sink with 29 parallel channels (199µm wide and 756µm deep) was tested experimentally with a split flow system with one central inlet at the middle of the channels and two outlets at either end. Three working fluids were tested (HFC134a, HFC236fa and HFC245fa) and also the parametric effects of mass velocity, saturation temperature and inlet temperature. The analysis of their results showed that a significantly higher CHF was obtainable with the split flow system compared to the single inlet-single outlet system (Park and Thome, 2010), providing also a much lower pressure drop. For the same mass velocity, the increase in CHF exceeded 80% for all working fluids evaluated due to the shorter heated length of a split system design. For the same total refrigerant mass flow rate, an increase of 24% for HFC134a and 43% for HFC236fa were obtained (no comparable data were available for HFC245fa). They concluded that the split flow system had the benefit of much larger CHF values with reduced pressure drops and further developments in the design of split flow system could yield an interesting energetic solution for cooling of computer chips.

Zhou et al. (2010) developed a steady-state model of a refrigeration system for high heat flux electronics cooling. The refrigeration system proposed consists of multiple evaporators (microchannel technology), liquid accumulator with an integrated heater, variable speed compressor, condenser and electric expansion valves (EEV). To obtain more efficient heat transfer and higher critical heat flux, the evaporators were considered to operate only with two-phase flow. To guarantee the safe operation of the refrigeration system the authors considered the presence of an integrated heater-accumulator to fully evaporate the two-phase flow coming out of the evaporator, which naturally represents a decrease of the cycle COP. A parametric study to evaluate the effects of external inputs on the system performance (secondary fluid temperature in the condenser, evaporator heat load, compressor speed, EEV percentage opening and heat supplied to the accumulator) and a Pareto optimization to find the optimal system operating conditions were also developed. A heat load of 1500W and 2500W, which represent respectively a heat flux of 94kW/m^2 and 156.6kW/m^2 were considered. The main points observed were: i) the system COP can be improved without compromising the critical heat flux when handling higher heat flux, ii) higher critical heat fluxes are achieved with a smaller EEV opening and higher heat input supplied to the accumulator and iii) a trade-off between the system COP and CHF is necessary to prevent the device burnout, i.e. imposed heat flux must be lower than the CHF considering a safety margin. Finally, they presented a preliminary validation of the model with initial experimental data showing a satisfactory prediction ability of the model. The authors do not mention anything about the geometry assumed for the evaporators.

The present study is a continuation of the earlier study developed by Marcinichen and Thome (2010). They proposed a two-phase cooling cycle considering a liquid pump as the driver of the fluids, a micro-evaporator for cooling the chip and its auxiliary electronics, and a microfin tube-in-tube condenser for heat recovery. A standard length (30cm) and internal diameter (3mm) was defined for the pipes joining the components, which were assumed to be only straight and horizontal, i.e. the effects of bends and static height difference between components and pipes were ignored. The performance of the cooling cycle considered 3 different working fluids; HFC134a, HFO1234ze and water (in an analogous single-phase cooling cycle). The results showed that for a design of the cooling cycle so that the total pressure drop is about 1bar, the liquid water cooling cycle had a pumping power consumption 16.5 times that obtained for the two-phase HFC134a cooling cycle. When comparing with the HFO1234ze cooling cycle, which showed a total pressure drop of 1.209bar, the difference drops to 13.2 times. It is important to mention that the simulations presented were considered as a benchmark and that the energetic comparison should be applied to an actual server's specifications, which is the subject that is considered here.

3. HYBRID TWO-PHASE COOLING CYCLE

Figure 1 depicts the proposed hybrid two-phase cooling cycle, i.e., a multi-purpose cooling cycle able to interchange the cycle driven by a liquid pump or a vapor compressor. The change of cycle would be accomplished by means of shut-off valves 1 to 7 (SOV). The choice of the cooling cycle would depend on the demand for heat recovery, or whether cycle maintenance is required (repair of the compressor or pump with one mode as a backup to the other mode). The microprocessors cannot operate without cooling and thus the interchangeability of the cycles represents a safety mechanism in case of failure of the pump or compressor. The "cons" of the hybrid cycle would be mainly the higher initial cost but certainly the advantages (system online reliability, controllability, cycle interchangeability and flexibility in heat recovery) may prove to justify the higher initial cost. Furthermore, this hybrid cycle represents a plug-and-play option where any one of the three cycles can be installed based on the particular application to standardize the design, thus minimizing engineering costs.

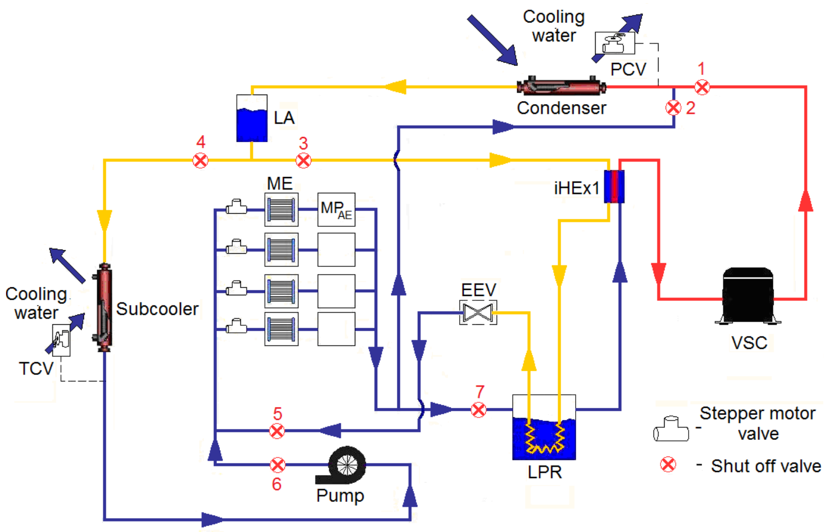

Figure 1. Hybrid two-phase cooling cycle.

The goal is to control the chip temperature to a pre-established level by controlling the inlet conditions of the micro-evaporator (pressure, subcooling and mass flow rate). It is imperative to keep the micro-evaporator (ME) outlet vapor quality below that of the critical vapor quality, which is associated with the critical heat flux. Due to this limitation, additional latent heat is available for further evaporation, which can be used by other low heat flux generating components in the blade server.

Another parameter that must be controlled is the condensing pressure (condensing temperature). The aim, during the winter, is to recover the energy dissipated by the refrigerant in the condenser to heat buildings, residences, district heating, etc. In order to accomplish this, the idea is to use a variable speed compressor (VSC) and an electric expansion valve (EEV), as will be discussed below.

Figure 2 depicts the two-phase cooling cycle where the flow rate is controlled by a liquid pump. The components considered and their main functions are presented below:

a) Variable speed liquid pump: controls the mass flow rate circulating in the system.

b) Stepper motor valve: controls the liquid flow rate to control the outlet vapor quality in each micro-evaporator (0% to 100%).

c) Micro-evaporator (ME): transfers the heat generated by the microprocessor to the refrigerant.

d) Microchannel cold plate for auxiliary electronics (MP_{AE}): additional component used to cool the auxiliary electronics using the remaining latent heat, which is available due to the limitations enforced on the micro-evaporator.

e) Pressure control valve (PCV): controls the condensing pressure.

f) Condenser: counter-flow tube-in-tube exchanger.

g) Liquid accumulator (LA): guarantees that there is only saturated liquid at the subcooler inlet, independent of changes in thermal load.

h) Temperature control valve (TCV): controls the subcooling at the inlet of liquid pump.

This cycle is characterized in having low initial costs, a low vapor quality at the ME outlet, a high overall efficiency, low maintenance costs and a low condensing temperature. This is a good operating option when the energy dissipated in the condenser is not recovered, typically during the summer season. However, the heat can still be recovered if there is an appropriate demand for low quality heat (low exergy).

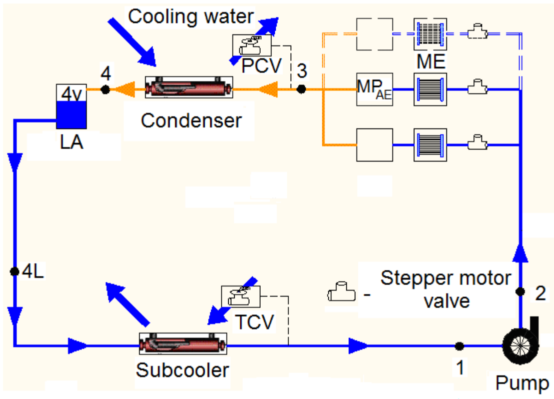

Figure 2. Schematic of the liquid pump cooling cycle.

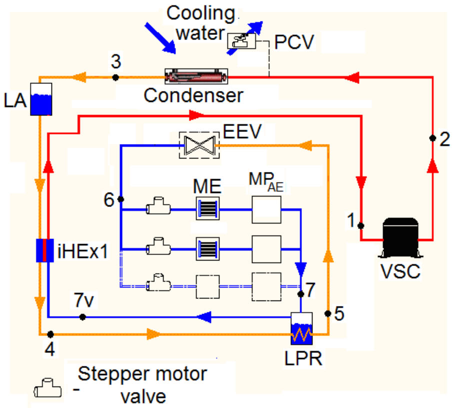

Figure 3. Schematic of the vapor compression cooling cycle.

Figure 3 shows a two-phase cooling cycle where a vapor compressor is the driver of the working fluid. The components considered and their main functions are:

a) Variable speed compressor: controls the ME inlet pressure and consequently the level of inlet subcooling.

b) Pressure control valve (PCV): controls the condensing pressure.

c) Condenser: counter-flow tube-in-tube exchanger.

d) Liquid accumulator: guarantees that there is only saturated liquid at the internal heat exchanger (iHEx1) inlet.

e) Internal heat exchanger liquid line/suction line (iHEx1): increases the performance of the cooling system.

f) Electric expansion valve (EEV): controls the low pressure receiver level.

g) Low pressure receiver (LPR): this component can be seen as a second internal heat exchanger liquid line/suction line, which increases the EEV inlet subcooling and allows an overfeed to the ME since the ME outlet returns to this receiver.

h) Stepper motor valve: controls the liquid flow rate to control the outlet vapor quality in each micro-evaporator (0% to 100%).

i) Micro-evaporator (ME): transfers the heat away from the microprocessor.

j) Microchannel cold plate for auxiliary electronics (MP_{AE}): cools the auxiliary electronics.

This cycle is characterized by a high condensing temperature (high heat recovery potential), a high range of controllability of the ME inlet subcooling (characteristic of systems with VSC and EEV), a medium overall efficiency when compared with the liquid pumping cooling cycle (uncertain, evaluate potential for heat recovery in the condenser). This is a good operating option when the energy dissipated in the condenser is recovered for other use, typically during the winter season when considering a district heating application (high exergy).

It is worth mentioning that the applicability of these cooling cycles is not restricted to only one microprocessor but can be applied to blade servers and clusters, which may have up to 64 blades per rack cabinet. Each blade, such as that shown in Figure 4, can have two (or more) microprocessors with a heat generation capacity higher than 150W. If the auxiliary electronics (memories, DC/DC, etc.) on the blade are included, the total heat generation per blade can be higher than 300W. Thus, the microchannel cold plate (MP_{AE}) described in the cooling cycles has the function to cool the auxiliary electronics that can represent about 60% of the total heat load on the blade, but will have a larger surface area compared to the CPU and thus a lower heat flux.

Finally, when considering an entire rack, a very sizable heat load is generated, which represents a good opportunity to recover the heat rejected. In this case, reuse of the heat removed from the blades for a secondary application will greatly reduce the CO_2 footprint of the system. For example, if we consider a data center with 50 vertical racks, where each rack has 64 blades and each blade dissipates 300W, the total potential heat to be recovered will be 0.96MW. Such a heat recovery system requires a secondary heat transfer fluid to pass through all the condensers (either water or a refrigerant) and then transport the heat to its destination.

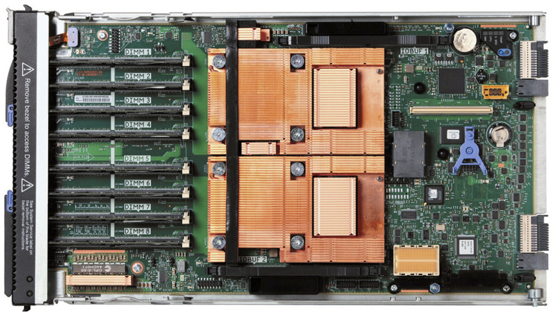

Figure 4. Typical blade with two microprocessors and a heat generation capacity higher than 300W.

4. GREEN SIMULATION CODE

A green in-house two-phase cooling cycle simulation code was developed to design and evaluate the performance of the liquid pump and vapor compressor cooling cycles under steady state conditions. The simulation code is able to design the condenser and subcooler, to evaluate the performance of the ME and various component coolers for a given heat load, and to calculate the pumping power consumption to drive the cooling cycle. The pressure drop of each component and the piping are also calculated. Table 1 shows the principal methods implemented in the code.

Table 1. Methods in the code.

Component	Type	Method
ME	Multi Microchannel	Heat transfer coefficient by Thome *et al.* (2004). Pressure drop as suggested by Ribatski *et al.* (2006). Critical heat flux by Revellin and Thome (2008).
Condenser (tube-in-tube)	inner tube: Spiral μ-fin (single-phase flow)	Heat transfer coefficient by Meyer and Olivier (2010).
		Pressure drop by Meyer and Olivier (2010).
	inner tube: Spiral μ-fin (two-phase flow)	Heat transfer coefficient by Cavallini (2000).
		Pressure drop by Cavallini (2000).
	annulus: Smooth (single-phase flow)	Heat transfer coefficient by Dittus and Boelter (1930).
		Pressure drop by Blasius (1913).
Subcooler (tube-in-tube)	inner tube: Ribbed (single-phase flow)	Heat transfer coefficient by Ravigururajan and Bergles (1985).
		Pressure drop by Ravigururajan and Bergles (1985).
	annulus: Smooth (single-phase flow)	Heat transfer coefficient by Dittus and Boelter (1930).
		Pressure drop by Blasius (1913).
Straight horizontal pipes	adiabatic (single-phase flow)	Pressure drop by Blasius (1913).
	adiabatic (two-phase flow)	Pressure drop by Muller-Steinhagen and Heck (1986).
Straight vertical pipes (upward)	adiabatic (single-phase flow)	Pressure drop by Blasius (1913) and Azzi *et al.* (2005).
	adiabatic (two-phase flow)	Pressure drop by Taitel *et al.* (1980), Barnea *et al.* (1982), Barnea (1986) and Liu and Wang (2008)
Straight vertical pipes (downward)	adiabatic (single-phase flow)	Pressure drop by Blasius (1913) and Azzi *et al.* (2005).
	adiabatic (two-phase flow)	Pressure drop by Barnea *et al.* (1982), Barnea (1986) and Perez-Tellez (2003).
Elbow (horizontal)	adiabatic (single-phase flow)	Pressure drop by Spedding et al. (2004).
	adiabatic (two-phase flow)	Pressure drop by Azzi *et al.* (2000).
Elbow (vertical)	adiabatic (single-phase flow)	Pressure drop by Spedding *et al.* (2004) and Azzi *et al.* (2005).
	adiabatic (two-phase flow)	Pressure drop by Azzi *et al.* (2005).

The input data required to run the simulation code are: (i) the geometrical parameters of ME's and heat exchangers, (ii) the heat load on the ME's and MP_{AE}'s, (iii) the evaporating temperature and subcooling at the ME inlet, the condensing temperature at the condenser inlet (only for the vapor compression cooling cycle), the water temperature at the condenser and subcooler inlet and outlet (secondary fluid flowing in the annulus), and (iv) the length and direction of the pipes and elbows joining the components.

The analysis of results were developed taking into account the thermal performance, the pumping power consumption and the total pressure drop of the cooling cycle. A comparison of 5 simulated cases considering different working fluids and cooling cycles was accomplished. For simulated cases 1 and 4, two-phase (TP) HFC134a and single-phase liquid water (SP_W) in the ME, the design considered for the liquid pump cooling system was such that the total pressure drop was about 1.5bar. The design constraint imposed on the condenser and the subcooler was that the pressure drops in the working and secondary fluids were, respectively, 0.05bar and 1bar. Cases 2, 3 and 5 considered the same heat exchanger geometries (ME, condenser and subcooler) as defined for the case 1. The difference in cases 2, 3 and 5 with respect to case 1 are that for case 2 the internal diameter of piping on the blade was reduced from 3mm to 2mm, for case 3 the working fluid was changed from HFC134a to HFO1234ze and for case 5 the vapor compression cooling cycle was considered.

A blade server with 16 blades was taken into consideration for the total heat load. Each blade, for example as that showed in Figure 4, presents two electronic systems in parallel and with each system being composed of one microprocessor (60W of heat load) and the auxiliary electronics (55.6W of heat load). Figure 5 shows the rack cabinet designed, with the piping configurations and components for the liquid pump cooling cycles with water and volatile working fluid (a, b) and (c) the vapor compression cooling cycle.

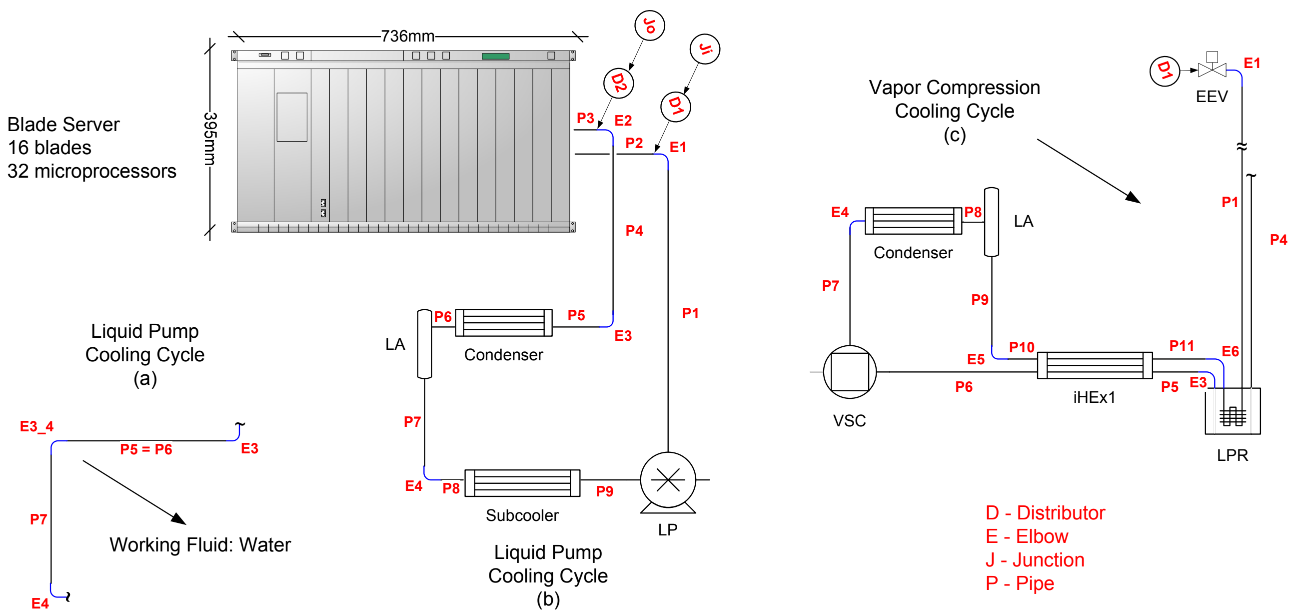

Figure 5. Rack cabinet – Frontal view.

In summary the following comparisons were made: i) SP_W versus TP_HFC134a, ii) TP_HFO1234ze versus TP_HFC134a, both comparisons for the liquid pump cooling cycle and iii) liquid pump cooling cycle versus vapor compression cooling cycle, both cycles with TP_HFC134a.

Table 2 shows the input data considered for each working fluid evaluated. The other thermodynamic parameters required to determine the total energy balance of the cycle come from the linkage to the methods shown earlier in Table 1.

Equation (1) shows the total energy balance of the cooling cycle:

$$\underbrace{Q_{ME} + Q_{AE}}_{\text{Heat load}} + W_{LP\_or\_VC} = Q_{cond} + Q_{sub} \qquad (1)$$

Q_{ME} and Q_{AE} are the heat loads associated with the microprocessor and the auxiliary electronics, which are transferred respectively by the ME and MP_{AE}. $W_{LP\_or\_VC}$ is the pumping power consumption of the driver, which can be the liquid pump or the vapor compressor. Finally, Q_{cond} and Q_{sub} are the heat transfer rate at the condenser and subcooler.

Table 2. Input data.

Component	Working fluid	Input data
Auxiliary Electronics	All of them	55.6W per half blade
ME	HFC134a HFO1234ze	Inlet evaporating temperature = 60ºC
		Inlet Subcooling = 0ºC
		Outlet vapor quality = 30%
		Q_{ME} = 60W per ME
	Water	Inlet temperature = 60ºC
		Outlet temperature = 62ºC
		Q_{ME} = 60W per ME
Condenser (tube-in-tube)	HFC134a and HFO1234ze	Inlet condensing temperature = 95ºC (case 5) Outlet vapor quality = 0%
	Secondary fluid: water	Inlet temperature = 15ºC
		Outlet temperature = inlet condensing temperature - 10K Outlet temperature = inlet condensing temperature - 5K (case 5)
Subcooler (tube-in-tube)	HFC134a and HFO1234ze	Inlet vapor quality = 0%
	Secondary fluid: Water	Inlet temperature = 15ºC
		Outlet temperature = subcooler inlet temperature - 10K
Compressor	HFC134a	Isentropic compression Condensing temperature = 95ºC
Liquid Pump	All of them	Isentropic pumping

It is worth pointing out that the condenser is a tube-in-tube heat exchanger with spiral microfins on the internal surface of the inner tube with a smooth external surface. For the tube-in-tube subcooler, the internal surface of the inner tube is again microfinned and smooth on the external surface. For both heat exchangers the objective was to find its length, while the other geometrical parameters were considered to be fixed. In the annulus, water was considered as the secondary fluid.

Tables 3 and 4 show the geometrical parameters considered for the condenser, subcooler and ME. For the ME the same geometrical parameters were considered in all cases with the only exception being for case 4 (SP_W cooling cycle) where the orifice distribution plate, normally used at the inlet of ME's to avoid problems of mal-distribution in the channels in two-phase flow (Agostini et al., 2007), was not considered. It is important to mention that for case 4 a value of 2K was simulated as the maximum axial rise in the chip's temperature from inlet to outlet of the ME, which represents also the rise in the water temperature from inlet to outlet. The actual temperature rise could be more, depending on the computer manufacturer's design specifications. Increasing this temperature difference will decrease the water flow rate for its simulation, and hence also reduce its pressure drop and pumping power accordingly, but will increase the local temperature of the microprocessor at the exit. On the other hand, unless the water is charged into the server's cooling system on site, then glycol must be added to the water circuit before shipment to prevent freezing, which will increase the pressure drop by about 50%. For the other TP cooling cycle cases, the maximum axial rise in the chip's temperature obtained by the ME simulations was about 0.2K, which is due to the near ideal matching of the tandem fall in the local flow boiling heat transfer coefficient and saturation temperature (pressure) along the micro-evaporator.

Table 3. Geometrical parameters / condenser and subcooler.

Cooling cycle	**Liquid pump**			**Vapor compression**
Working fluid	**HFC134a or HFO1234ze**		**Water**	**HFC134a**
Heat exchanger	**Condenser**	**Subcooler**	**Subcooler**	**Condenser**
Inner tube	Spiral μ-fin	Ribbed	Ribbed	Spiral μ-fin
Fin tip diameter [mm]	10.8	5.53	14.00	10.8
Fin height [mm]	0.25	0.14	0.305	0.25
Thickness of the tube [mm]	0.35	0.27	0.635	0.35
Helix angle [º]	18	16	27	18
Apex angle [º]	50	---	---	50
Number of fins [-]	70	55	60	70
Outer tube	Smooth	Smooth	Smooth	Smooth
Internal diameter [mm]	17.52	8.72	21.33	17.52

Table 4. Geometrical parameters of the ME.

Micro-evaporator			
Fin height [μm]	1700	Length [mm]	13.5
Fin width [μm]	170	Width [mm]	18.5
Channel width [μm]	170	Area [cm^2]	2.5
Base thickness [mm]	1	Material	Copper
Split flow		1 inlet and 2 outlets	

Table 5 shows the results obtained by the methods developed to evaluate the performance of the ME's. The three-zone model (Thome *et al.*, 2004) was used for predicting the two-phase heat transfer since it was shown to predict many fluids and geometries with good accuracy (Dupont *et al.*, 2004), the numerically based model of Revellin and Thome (2008) was used for critical heat flux (CHF) calculations and the homogeneous model was used for two-phase pressure drops since it was found to predict microchannel pressure drops with relatively good accuracy (Ribatski *et al.*, 2006). The results show that a much higher mass flow rate of water is required for the SP_W cooling cycle than for the TP cooling cycles, which is justified by the latent heat of the refrigerants being 30-60 times the liquid specific heat of the water and because of the low maximum junction temperature rise defined as an input parameter. The pressure drop is low for all the fluids, in part as a consequence of the split flow design. For the outlet vapor quality considered (30%), the predicted CHF was higher than 6 times the actual maximum heat flux of 24W/cm^2. This safety factor is more than sufficient since the accuracy in predicting CHF is about ±20%.

Table 5. ME performance.

Cooling Cycle	**TP**		**SP_W**
Working Fluid	**HFC134a**	**HFO1234ze**	**Water**
Inlet evaporating temperature [ºC]	60	60	-----
Outlet evaporating temperature [ºC]	60	60	-----
Mass flow rate per ME [kg/h]	**5.18**	**5.32**	**25.81**
Outlet vapor quality [%]	30	30	-----
Inlet temperature [ºC]	60	60	60
Outlet temperature [ºC]	-----	-----	62
Pressure drop [bar]	0.001	0.001	0.005
Heat flux [W/cm^2]	24	24	24
CHF [W/cm^2]	154.4	165.6	-----

Tables 6 and 7 show the simulation results for the 5 cases mentioned beforehand. The pressure drop in each component and piping, pumping power consumption, total pressure drop, heat transfer rate in the heat exchangers and their calculated length, internal diameter of piping, water mass flow rate, outlet temperature at the heat exchangers and pressure drop (secondary fluid) are shown. It is important to remember that the SP_W cooling cycle does not consider the condenser and liquid accumulator, which are not required in that cycle (see Figure 5).

Table 6. Piping diameters, pumping power and pressure drop.

CASES	**1**	**2**	**3**	**4**	**5**
Cooling Cycle	**Liquid pump**				**Vapor compression**
Working Fluid	HFC134a	HFC134a	HFO1234ze	Water	HFC134a
Internal diameter of piping on the blade [mm]	3.0	2.0	3.0	3.0	3.0
Internal diameter of piping for single-phase flow [mm]	4.1	4.1	4.1	7.7	4.1
Internal diameter of piping for two-phase flow [mm]	7.0	7.0	7.0	----	7.0
Pressure drop in the piping on the blade [bar]	0.021	0.134	0.028	0.127	0.021
Total pressure drop [bar]	1.54	1.65	1.79	1.50	1.87
$W_{LP\_or\_VC}$ [W]	6.4	7.0	7.9	35	746.5
Components	**Percentage of pressure drop by component and piping [%]**				
ME	0.02	0.02	0.07	0.29	0.02
Condenser	2.86	2.67	3.36	----	1.62
Subcooler	3.53	3.19	3.08	3.51	----
Piping	94.61	95.06	94.35	96.20	90.43
Components (total)	5.39	4.94	5.65	3.80	9.57

Comparing cases 1 and 4, TP_HFC134a and SP_W liquid pump cooling cycles, it is worth noting the difference regarding the internal diameter of piping and pumping power consumption. Case 4 presents the largest values, 7.7mm and 35W, respectively. A larger pipe diameter was necessary to guarantee the 1.5bar pressure drop defined as a design constraint, with the large pumping power being a consequence of the high mass flow rate of water (see Table 5). The pressure drop in the piping on the blade was 6 times lower for case 1. The total length of the heat exchangers, shown in Table 7, was highest for case 1. However, since case 4 imposed greater inner and outer tube diameters due to the 0.05bar and 1bar pressure drop design constraints in the working and secondary fluid sides, respectively, the volume occupied by the heat exchangers is highest for case 4 (24% higher than for case 1), which represents a higher material cost. Finally, it can also be observed for both cases that the pressure drop in the piping represents more than 90% of the total pressure drop (*viz.* Table 6).

Table 7. Heat exchangers simulations.

CASES	1	2	3	4	5
Condenser					
Length [m]	2.66	2.66	2.78	-----	2.62
Mass flow rate of secondary fluid [kg/h]	81.88	82.88	87.70	-----	50.72
Pressure drop secondary fluid side [bar]	0.954	0.976	1.129	-----	0.374
Pressure drop working fluid side [bar]	0.044	0.045	0.060	-----	0.021
Outlet temperature of secondary fluid [^{o}C]	**48.0**	**47.7**	**46.8**	**-----**	**90.0**
Heat transfer rate [W]	3176.65	3199.23	3274.01	-----	4445.70
Subcooler					
Length [m]	1.07	1.05	1.00	2.45	-----
Mass flow rate of secondary fluid [kg/h]	13.60	13.14	11.51	84.34	-----
Pressure drop secondary fluid side [bar]	0.981	0.906	0.687	1.045	-----
Pressure drop working fluid side [bar]	0.054	0.053	0.055	0.053	-----
Outlet temperature of secondary fluid [^{o}C]	**47.9**	**47.6**	**46.5**	**53.2**	-----
Heat transfer rate [W]	528.95	506.97	433.09	3734.20	-----
Total heat exchanger volume [cm^3]	**705.2**	**704.0**	**729.9**	**875.5**	**631.6**

Case 2 considers a smaller internal diameter of the piping on the blade, i.e. 2mm, which can be advantageous for installing the cooling system on the blade due to the limited space available. The smaller diameter piping is also inherently more flexible making it easier to conform to the layout of the electronic components on the motherboard, which leads to less stress exerted on the blade. The comparison of the results between cases 1 and 2 shows an increase in pumping power and total pressure drop of about 9.4% and 7.1%, respectively, although these values are still much lower than that obtained for the SP_W (case 4) presented earlier. Case 3, which considers the new environmentally friendly working fluid HFO1234ze, with the heat exchangers also having the same geometries as in case 1, showed a small increase in pumping power consumption and total pressure drop when compared with case 1, being on the order of 23.4% and 16.2%, respectively.

Case 5 shows the simulation results of the TP_HFC134a vapor compression cooling cycle, which considers the same piping diameters and heat exchanger geometries defined for case 1. A significant increase in pumping power is observed compared to case 1. However, what is of interest is the lower heat exchanger volume required, being the lowest of all the cases. As mentioned beforehand, this cycle would be a good operating option when the energy dissipated in the condenser is recovered for other uses, for example into the feedwater heaters of thermal power plants.

The vapor compression cooling cycle permits the recovery of the energy from the condenser for higher levels of condensing temperatures (high exergy, see Table 7). By inserting this recovered energy into a thermal power plant, one could potentially increase the water temperature exiting the plant's condenser from 45˚C (typical value) to 90˚C. This added energy will decrease the amount of fuel required to generate electricity, hence increasing the thermal performance of the plant by 2-3%, while also decreasing its and the data center's CO_2 footprint. Therefore, by implementing a vapor compression cycle into a data center would not only reduce its operating costs with millions being saved annually, but also have less of an impact on the environment.

Despite the high complexity of the vapor compression cycle, i.e. number of components and total length of the piping, a negligible increase was observed in the total pressure drop when comparing with the liquid pump cycle (case 1). It is also important to mention that the vapor compression cooling cycle showed a lower water mass flow rate and heat exchanger volume when compared with the other cycles proposed, which means respectively a lower pumping power of the secondary fluid and material costs of the condenser. Finally, it is worth mentioning that the higher heat transfer rate in the condenser is associated with the work imparted by the compressor.

5. CONCLUSIONS

A hybrid two-phase cooling cycle has been proposed and simulated to cool microprocessors and auxiliary electronics of blade server boards with two-phase evaporating flow in the micro-evaporator cooling elements. A simulation code was developed and 5 cases were simulated considering 3 different working fluids, HFC134a, HFO1234ze and water (in an analogous single-phase cooling cycle) and different internal diameters of the pipes and elbows joining the components. The results showed that for a design of the cooling cycle such that the total pressure drop is about 1.5bar, the liquid water cooling cycle has a pumping power consumption 5.5 times that obtained for the two-phase HFC134a cooling cycle, both considering a liquid pump as the driver of the fluid. When compared with the HFO1234ze cooling cycle, which showed a total pressure drop of 1.79bar, the difference drops to 4.4 times. The simulation of the vapor compressor cooling cycle showed higher pumping power consumption when compared with the other cycles simulated. However, this cycle can be justified when the waste heat at the condenser is recovered for applications such as district heating and preheating of boiler feedwater. The highest condensing temperature (higher secondary fluid temperature) and heat transfer rate (associated with the work imparted by the compressor) represent a higher economic value than that obtained with the liquid pump cooling cycles.

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