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〈Original Article〉

A Study on the Boiling Heat Transfer Characteristics Using Loop Type Thermosyphon

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Flexible two-phase thermosyphons are devices that can transfer large amounts of heat flux with boiling and condensation of working fluid resulting from small temperature differences. A flexible two-phase thermosyphon consists of a evaporator, an insulation unit, and a condenser. The working fluid inside the evaporator is evaporated by heating the evaporator in the lower part of the flexible two-phase thermosyphon and the evaporated steam rises to the condenser in the upper part to transfer heat in response to the cooling fluid outside the tube. The resultant condensed working fluid flows downward along the inside surface of the tube due to gravity. These processes form a cycle. Using R134a refrigerant as the working fluid of a loop type flexible two-phase thermosyphon heat exchanger, an experiment was conducted to analyse changes in boiling heat transfer performances according to differences in the temperature of the oil for heating of the evaporator, the temperature variations of the refrigerant, and the mass flows. According to the results of the present study, the circulation rate of the refrigerant increased and the pressure in the evaporator also increased proportionally as the temperature of the oil in the evaporator increased. In addition, the heat transfer rate of the boiler increased as the temperature of the oil in the evaporator increased.

Keywords : Loop type thermosyphon, Boiling heat transfer, Evaporator, Heat flux, Working fluid

Introduction

Flexible thermosyphon technology can improve cooling performance because these systems release heat using the latent heat resulting from the boiling and condensation of the working fluid so that high performance systems can be implemented and the prices, sizes, and power consumption of the systems can be reduced (Rhi et al., 1977; Han et al., 2005). As such, cooling systems with high speed rotating shafts using the loop type two-phase flexible thermosyphon technology can be made at much lower prices compared to existing cooling systems because they do not require devices such as cooling fluid circulating pumps and chillers (Kim et al., 1998). Chen and Chang (1988) identified an excursive unstable phenomenon from static instability and found out that the unstable phenomenon appeared because areas where pressure drop increased when flow velocity decreased existed in boiling channels. Knaani and Zvirin (1994) constructed an electronic component cooling system using a two-phase loop thermosyphon and conducted

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experimental and theoretical studies on refrigerant pressure drops and rises at individual points of the cycle. Lee et al. (1995) studied the instability caused by nucleation in static instability in a natural circulation loop installed with two parallel vertical tubes. According to the study results, a new geysering phenomenon occurred as slug bubbles were generated in the channel. Currently, cooling systems using cooling water or oil as a cooling fluid are widely utilized in South Korea and other countries too. However, since such existing cooling systems are composed of many additional elements such as heat exchangers for heat removal, cooling fluid circulating pumps, cooling fluid circulating pipe, chillers for cooling fluid heat removal, cooling fluid flow control valves, and cooling fluid flow control devices, not only their structures are complicated but also they are expensive, consume large amounts of electricity, induce corrosion, and cause damage to electrical components when leaks occur(Liao et al., 2000). In addition, such existing cooling systems use driving devices such as pumps to circulate cooling fluids and this leads to shortcomings such as lower reliability and increased sizes and weights. When seawater or fresh water is used as a cooling fluid, leaks of the cooling fluid will bring about fatal damage and corrosion and contamination will shorten the life the cooling system and will lead to excessive maintenance costs(Faghri et al., 1989). However, cooling systems using loop type two-phase flexible thermosiphon technology can improve cooling performance because these systems release heat using the latent heat resulting from the boiling and condensation of the working fluid so that high performance systems can be implemented and the prices, sizes, and power consumption of the systems can be reduced(Karl et al., 1981).

In the present study, experimental studies were conducted on the heat transfer performance of the boiler of a loop type two-phase flexible thermosyphon heat exchanger that can discharge the large amounts of heat generated by the high speed rotation of the high speed rotation shafts of high voltage motors, generators, and large lathes. Simulations were conducted to evaluate the heat transfer performance, and the results were verified through comparison with experimental results.

Materials and methods

Fig. 1 shows the loop type two-phase flexible thermosyphon heat exchanger for high speed rotating shaft heat release. The loop type two-phase flexible thermosyphon heat exchanger is installed with an evaporator which is a heating section and a condenser which is a cooling section that are separated from each other. The evaporator and the condenser are assemblies of pipes composed of many pipes and they are constructed so that they are connected to each other through pipes gathered on the top and bottom. The evaporator and the condenser have conveying pipes in charge of working fluid flows connected to each other and the condenser is installed with a gas/liquid separator that gathers non-condensable gases and an exhaust gas valve. In loop type thermosyphon heat exchangers, because of the characteristics of thermosyphons, the condenser should be installed at a higher location than the evaporator without fail to obtain the pressure difference for circulation of the working fluid.

Fig. 2 shows schematic diagram of the evaporator and Fig. 3 shows evaporator. The evaporator of the 6.5 kW

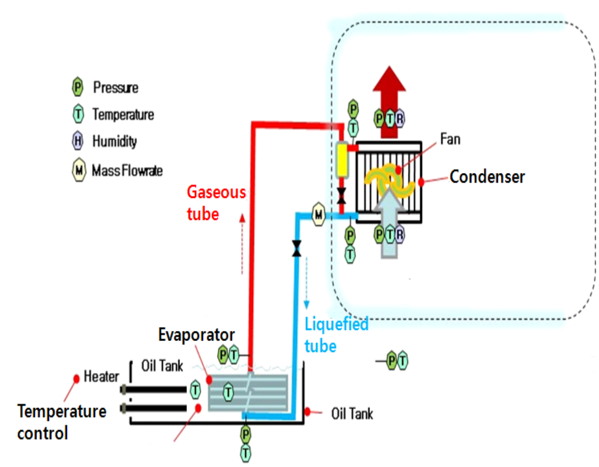

Fig. 1. Loop type two-phase thermosyphon heat exchangers for high speed rotating shaft heat release.

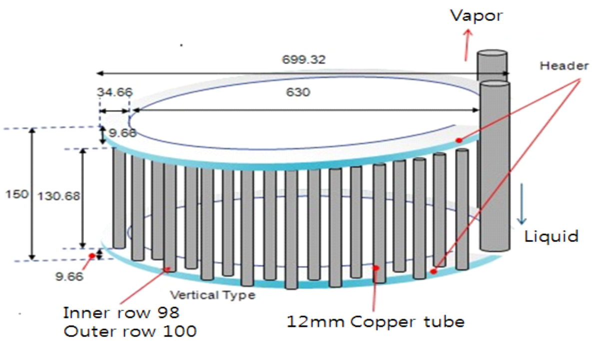

Fig. 2. Schematic diagram of the evaporator.

grade loop type thermosyphon heat exchanger made for the present study is 699 mm in the outside diameter, 630 mm in the inside diameter, and 150 mm in the height. In addition, 98 pieces of 12 mm diameter copper pipes were installed in the inside of the evaporator and 100 pieces of 12 mm diameter copper pipes were installed on the outside of the evaporator. Furthermore, L type 25 mm soft copper pipes were installed on the top and bottom of the evaporator. The condenser is 480mm wide, 1000 mm long and 68 mm high. Eighty 12 mm diameter, 1,000 mm long copper pipes were installed on the condenser. In addition, exhaust valves were attached to the condenser so that the working fluid can be filled and non-condensable gases can be exhausted. As shown in Fig. 2, for performance experiments of the 6.5 kW grade loop type thermosyphon heat exchanger, the height difference between the evaporator and the condenser was set to 1.75 m. The outside diameter of the conveying pipe that connects the evaporator with the condenser is 20 mm.

To prevent leaks from occurring in the conveying zone that connected the evaporator with the condenser, the conveying zone was checked using high pressure nitrogen and the conveying zone was insulated with 25 mm thick glass fibers to reduce heat losses to the outside of pipes occurring when the working fluid flows through the steam and fluid flow pipes. To measure individual experimental data in the performance experimental facility, Pressure Transducers (P21AA, 0~2MPa) were installed at the working fluid inlet/outlet of the evaporator so that the saturated vapor pressure of the working fluid can be measured and reviewed in comparison with the saturation temperature. To measure pressure drops in the vapor tube, Pressure Transducers (P21AA, 0~2MPa) were installed at the inlet/outlet of the condenser. One thermocouple (K-type sheath dirt. 1.6 mm OD) each was installed at each of the inlet/outlet of the evaporator and the inlet/outlet of the condenser. To measure the oil temperature in the housing of the evaporator and the air temperature in the condenser, three each of thermocouples were installed at the inlet and out let respectively. The experimental data were measured and recorded using a Hybrid Recorder (64 channels). A refrigerant flowmeter was installed at the condenser outlet tube to measure the condensate refrigerant flow rate.

In loop type thermosyphon heat exchanger performance experiments, initial temperatures are very important. Therefore, the non-condensable gases remaining in the tube were discharged so that a constant degree of vacuum is maintained in normal states. R-134a was used as a working fluid. The quantity of the injected working fluid was measured on a scale. A Slidacs was installed to maintain the oil temperature in the oil housing that supplies heat to the evaporator at 20~80℃ when the experiment was conducted. The temperatures of the working fluid at the inlet and outlet of the evaporator and the condenser at individual layers were compared.

Fig. 3. The evaporator.

Results and discussion

Fig. 4 shows the shape of a preliminary simulation for evaporator phase change models and related boundary conditions. In the preliminary simulation, the RPI(Rensselaer Polytechnic Institute) model containing relational expressions for nucleation, growth, and breakaway of bubbles on walls provided by ANSYS FLUENT v.13 was applied and the boundary conditions provided by ANSYS FLUENT v.13 were applied for the simulation shape and boundary conditions.

Fig. 5 shows velocity distributions in oil housings. The resultant velocity distribution of the simulation to predict oil flows in the evaporator oil housing indicated that the effects of oil flows resulting from shaft rotation on the flows of the oil in the vicinity of the heat pipe should not be big. Since the effects of oil flow velocity in the oil housing on the flows of the oil in the vicinity of the heat pipe are predicted to be insignificant based on the results of the simulation to predict oil flows in the evaporator oil housing, the oil flows in the vicinity of the heat pipe are considered to be natural convection.

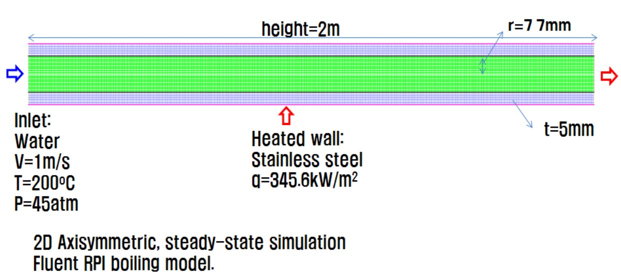

Fig. 4. The shape of the preliminary simulation and related boundary conditions for evaporator phase change models.

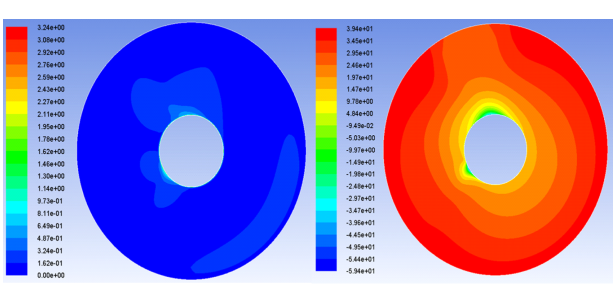

Fig. 5. Distributions of gas fraction, gas velocity, liquid velocity and liquid temperatures.

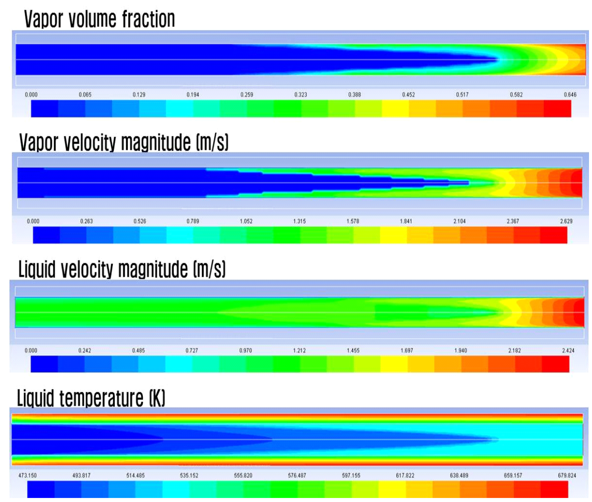

Fig. 6. Velocity distribution in the oil housing.

Fig. 6 shows the distributions of gas fraction, gas velocity, liquid velocity, and liquid temperatures in the pipe. It can be seen that in the pipe, the gas begins to be slowly generated from the wall and united on the center and this result was consistent with the gas velocity distribution. In addition, strong tendencies for the prediction of the position of gas generation and gas fractions in the pipe appeared according to wall heat flux.

Fig. 7 shows changes in the flow pattern of the system when heat flux increases following low degrees of undercooling. As shown in Fig. 7, when heat flux

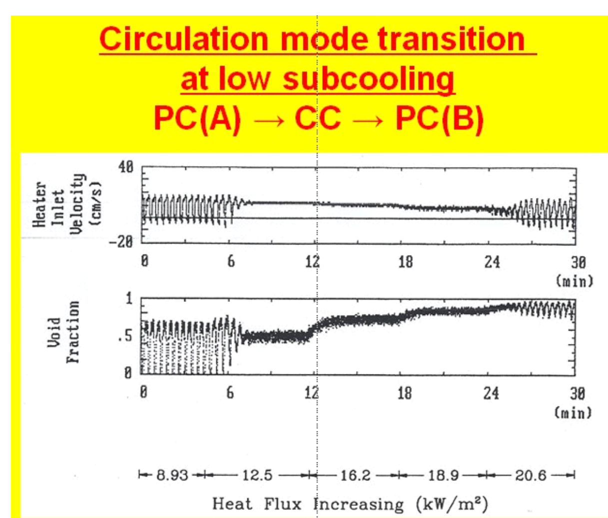

Fig. 7. Circulation mode transition at low subcooling.

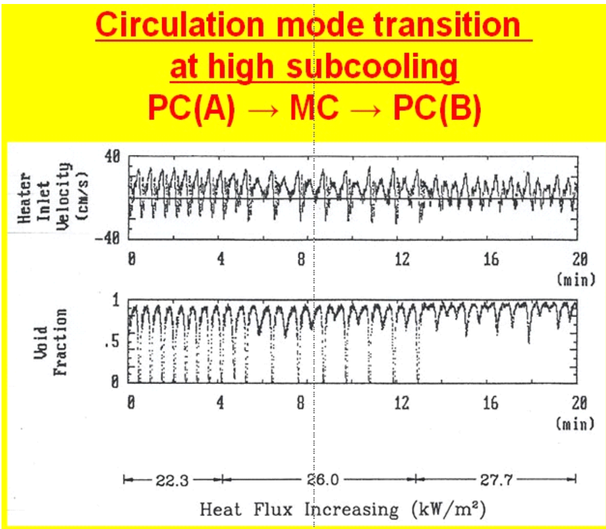

Fig. 8. Circulation mode transition at high subcooling.

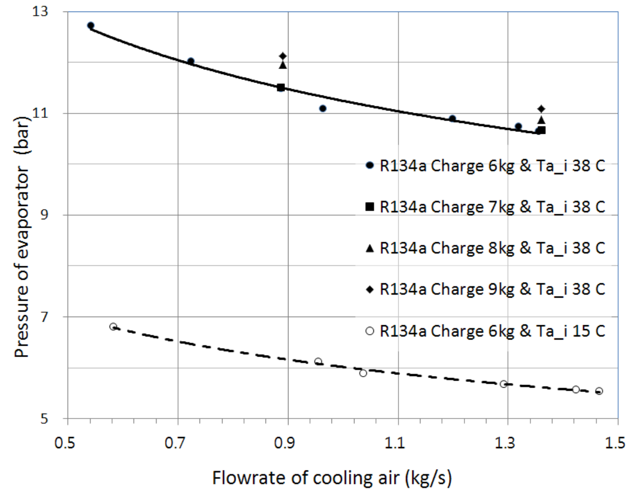

Fig. 9. Evaporator pressure according to condenser cooling air flow rates.

increases following low degrees of undercooling, as heat flux increased, the flow pattern of the system changed from the form of periodic circulations to the form of continuous circulations and back to the form of periodic circulations thereafter.

Fig. 8 shows changes in the flow pattern of the system when heat flux increases following high degrees of undercooling. When heat flux increases following low degrees of undercooling, as heat flux increased, the flow pattern of the system changed from the form of periodic circulations to the form of multi mode circulations and back to the form of periodic circulations thereafter. In addition, four flow patterns occurred across largely four areas and except for the continuous circulation pattern, all flows periodically fluctuated. That is, the continuous circulation pattern means a condition under which flows are continuous and uniform. Therefore, the system should designed so that continuous circulation patterns can be formed as flow patterns and operating conditions should be also controlled to maintain continuous circulation patterns for the loop thermosyphon to operate normally.

Fig. 9 and 10 show the results of evaporator experiments and Fig. 9 also shows changes in the pressure of the evaporator according to condenser cooling air flow rates. The refrigerant used in the experiments is 134a. The experiments were conducted at refrigerant charge rates of 6~9 kg. As the air flow rate in the condenser increased, evaporator pressure decreased. Therefore, it can be seen that as air flow rates in the condenser increased, the boiling steam generation rate of the loop thermosyphon heat exchanger increased.

Fig. 10 shows differences between oil temperatures and evaporator temperatures resulting from changes in the cooling air flow rate in the condenser. The refrigerant used in the experiments is 134a. The experiments were conducted at refrigerant charge rates of 6~9 kg and the air flow rates in the condenser in a range of 0.58~1.36 kg/s. As the condenser cooling air flow rate increased, the difference between oil temperatures and evaporator temperatures increased. Therefore, it can be seen that as cooling air flow rates in the condenser increased, the boiling heat transfer rate of the loop thermosyphon heat exchanger increased.

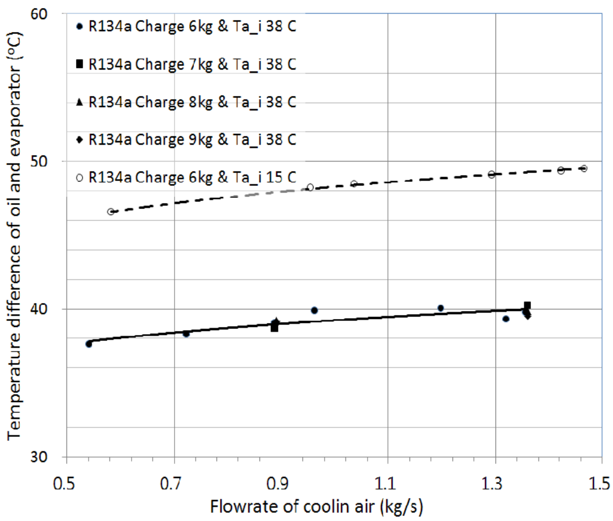

Fig. 10. Difference between oil temperatures and evaporator temperature according to condenser cooling air flow rates.

Conclusion

Experiments of loop type two-phase flexible thermosyphon heat exchanger for heat discharge from high speed rotating shafts were conducted using R-134a as a working fluid, experimental studies were conducted on changes in evaporator pressure and differences between oil temperatures and evaporator temperatures according to condenser cooling air flow rates, and the experimental results were compared with simulation results. The results are summarized as follows.

Strong tendencies for the prediction of the position of gas generation and gas fractions in the pipe appeared according to wall heat flux. As air flow rates in the condenser increased, the boiling steam generation rate of the loop thermosyphon heat exchanger increased. As condenser cooling air flow rates increased, differences between oil temperatures and evaporator temperatures increased. It can be seen that as cooling air flow rates in the condenser increased, the boiling heat transfer rate of the loop thermosyphon heat exchanger increased.

Acknowledgements

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