Influence of Working Fluids to Heat Transfer Characteristics of Heat Exchanger using Oscillating Capillary Tube Heat Pipe for Low Temperature Waste Heat Recovery

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Key words: Oscillating capillary tube heat pipe (OCHP), Thermosyphon, Thermal resistance, Figure of Merit

Abstract

Heat transfer characteristics of a heat exchanger for low temperature waste heat recovery using oscillating capillary tube heat pipe (OCHP) were evaluated against the charging ratio variation of working fluid and various working fluids. R-142b, R-22 and R-290 were used as working fluids. The heat exchanger was composed of heat pipe with capillary tube bundles, having a 2.6 mm in outside diameter, 1.4 mm in inside diameter with 101 m length and 140 turns. Charging ratio of working fluid was 40% and 50%. Water was used as secondary fluid. Inlet temperature and mass velocity for each secondary fluid were 297 K, 280 K and 9~27 kg/m^{2}s, respectively. From experimental results, it was found that heat transfer performance of R-22 was higher than those of R-142b and R-290 and it was proportional to Figure of Merit for thermosyphon. As a result, it was thought that R-22 was the most reasonable working fluid of waste heat recovery for low temperature waste heat recovery.

Nomenclature

D : diameter [m]
H : length [m]
h_{fg} : latent heat of evaporation [J/kg]
Q : heat transfer rate [W]
R : thermal resistance [K/W]
Re : Reynolds number
T : temperature [K]

Greek symbols

Φ : figure of Merit
σ : surface tension [N/m]
ρ : density [kg/m^3]
λ : thermal conductivity [W/mK]

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Subscripts

c : capillary
c : condensing
e : evaporation
i : inside
l : liquid
tot : total
v : vapor
1 : heat Pipe
2 : thermosyphon

1. Introduction

The temperature of bath in fish culture largely influence on the growth rate of fish. For this reason, in domestic fish farms, water was supplied to fish farm after water temperature of 7℃ was risen to 21℃ using boiler during winter. Thus, boiler is also operated to raise water temperature. Like this, expense used to maintain temperature appropriately is a main reason of rise of cost price with labor cost of cultivation.(1) Therefore, there are many demands to develop waste heat recovery of high performance can be applied to conventional fish breeding system for saving fuel expenses. Heat exchanger was used to recover waste heat as the general form of waste heat recovery.(2) In waste heat recovery of fish farm, shell & tube type and plate type heat exchangers are being used to recover waste heat. In recent, heat exchanger system applying principle of heat pipe is being considered as a method for recover waste heat effectively.

In heat pipe type heat exchanger, because additive power source was not needed for the circulation of primary fluid and secondary fluid of high and low temperature side flow outside of tube, pressure drop is low and required power can be reduced. And, as heat transfer coefficient was high compared with heat transfer area, miniaturization of heat exchanger is possible. And, first of all, a large quantity of heat can be transported with low temperature(3,4) using heat pipe heat exchanger.

Especially, oscillating capillary tube heat pipe (OCHP) recently developed can be made flexible and it stand high pressure as well as it can be made to low cost and short process.(5,6)

Therefore, in this study, based on heat transfer performance to evaluate influence of working fluid and its charging ratio on heat transfer of OCHP heat exchanger, it was intended to obtain basic data to develop waste heat recovery of high performance which can recover waste heat of low temperature using OCHP.

2. Operation principle and conventional studies for OCHP

OCHP is a device to transport heat by the self-excited oscillation of fluid. At this time, working fluid was not returned to evaporating part by wick, which was widely used to return liquid of condensing part to evaporating part in conventional heat pipe. Its structure was made up of closed loop of capillary copper tubes shown in Fig. 1.

Its basic operation was due to irregular circulation and the self-excited oscillation to axial direction of working fluid and vapor within the loop. Nucleate boiling was occurred as much as heat rate given to the heating part. Bubbles occurred by nucleate boiling were coalesced and its flow pattern was changed to the slug flow of vapor and liquid. Slug flow induced

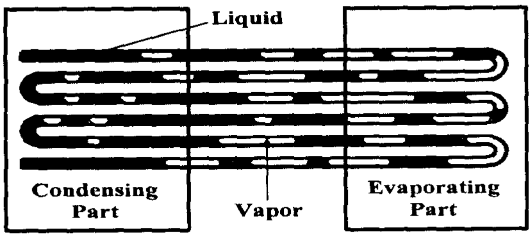


Fig. 1 Basic concept of OCHP.

oscillation to the axial direction as well as occurrence of the pressure wave. Convective heat transfer and latent heat were occurred owing to vapor bubbles.(5,7)

Nishio et al.(8 11) conducted experiments and analysis for heat transfer performance to charging ratio and working fluids in OCHP composed of glass and copper tube when distilled water, ethanol and R-141b were used as working fluids. It was reported that heat transfer performance of OCHP was superior when the charging ratio was 30~60(vol.%) and there was little difference in heat transfer performance to variation of working fluid. But, the more the ratio of capillary ascending length to inside diameter(D_i/H_c) of working fluid was large, the higher heat transfer rate was.

Gi et al.(12) conducted heat transfer characteristics to variation of charging ratio of R-142b after making OCHP of loop and non-loop type using teflon tube of inside diameter 2 mm and 4 mm and reported that maximum heat transfer rate was obtained when charging ratio was 50~60(vol.%) in case of loop type and charging ratio was 30(vol.%) in case of non-loop type.

In recent, Lee et al.(6) conducted heat transfer performance when R-141b was used as a working fluid in OCHP which was composed of aluminum extruded tube and reported that heat transfer performance was excellent when the charging ratio was about 40(vol.%).

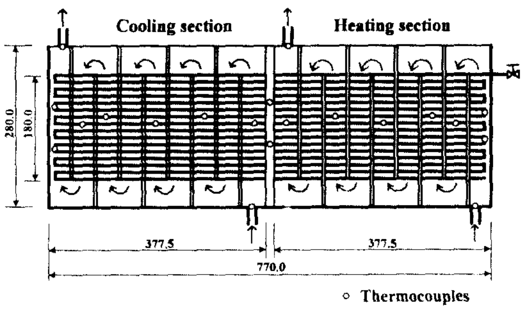


Fig. 2 Schematic diagram of heat exchanger for low temperature waste heat recovery.

3. Experimental apparatus and methods

3.1 Experimental apparatus

In Fig. 2, the schematic diagram of waste heat recovery of low temperature using OCHP made in this study was represented. The experimental apparatus was composed of waste heat recovery, the circulation part of heating and cooling water to supply heat rate to evaporating and condensing part, and data acquisition system. Waste heat recovery was composed to the serpentine of outside diameter 2.6 mm, inside diameter 1.4 mm, total length 101 m, and the turn number 140. The length of heat pipe was 0.77 m and the length of evaporating and condensing part was 0.38 m, respectively. And, the baffles of each 8 pieces were installed to evaporating and condensing part to conduct heat transfer effectively. Heat exchanger was sufficiently insulated to prevent from heat loss to outside.

Before charging working fluid within heat pipe, inside of heat pipe was evacuated to 6.8 $\times 10^{-6}$ torr. using high vacuum system, which was connected rotary and diffusion pump in series. The charging cylinder was used to charge working fluid properly.

The experimental apparatus was composed of test section, secondary flow part to supply he-

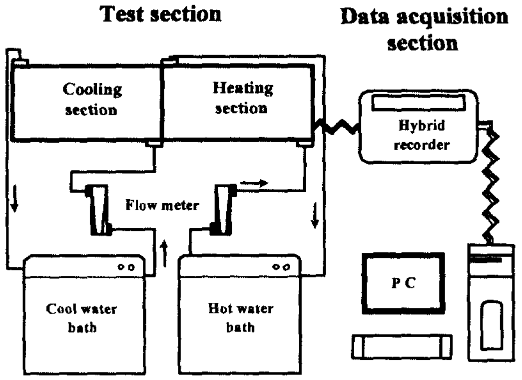


Fig. 3 Schematic diagram of experimental apparatus.

ating and cooling water of constant temperature and flow rate to test section and data acquisition system as shown in Fig. 3. Thermocouples of two pieces were installed at the inlet and outlet of evaporating and condensing part to measure temperature of secondary flow. T type thermocouples of 16 pieces were installed on surface of heat exchanger to measure the surface temperature of tube, each 7 pieces of which was installed to evaporating and condensing part and two pieces were installed to adiabatic part.

Data acquisition system was composed of Hybrid recorder (DR230, Yokogawa Co.) and computer to measure temperature needed to understand heat transfer performance and after surface temperature of tube outside was stabilized, data was detected with intervals of 2 second for 20 minutes and they was processed with computer. And, volumetric flow meter (Dwyer Co.) of float type was used to measure flow rate of secondary flow.

3.2 Experimental methods

In wate circulation part, heating and cooling water of constant flow rate were supplied using flow rate control valve after maintaining to constant temperature using constant temperature water bath to supply constant heat rate. The heating and cooling water of 297 K and 280 K was supplied to the secondary fluid part. Mass flow rate supplied to condensing and evaporating part was varied from 2 kg/min to 6 kg/min.

In Table 1, the flow rate of secondary flow supplied to evaporating and condensing part was converted to mass velocity and Re.

3.3 Figure of Merit of and choice of working fluid

As the choice of working fluid is very important to stable operation both wick type heat pipe and thermosyphon, it was decided by the thermo physical properties. Figure of Merit(2) to working fluid of wick type heat pipe and thermosyphon can be expressed to equation (1) and (2), respectively. And, these are used to evaluate the effectiveness of various working fluids at the specific operation temperature. But until now, because evaluation index for OCHP was not established, these kinds of evaluation index cannot be directly applied to OCHP. But, in this study, some working fluid was represented using evaluation index for thermosyphon to understand thermal characteristics of working fluid. This is considered that each tubes composing OCHP is real inter-connected tube and capillary tube thermosyphon of wickless type.

$$\Phi_1 = \frac{h_{fg}\rho_l\sigma}{\mu_l} \tag{1}$$

$$\Phi_2 = \left[\frac{h_{fg}\lambda_l^3\rho_l^2}{\mu_l}\right]^{0.25} \tag{2}$$

Figure 4 showed the evaluation index when methanol, acetone, R-22, R-290, R-600, R-600a, R-141b and R-134a were used as working fluids. Comparing evaluation index of each working fluid around operation temperature 290 K, the evaluation index of methanol, acetone and

Table 1 Mass velocity and Reynolds number

Flow rate (kg/min)	Mass velocity (kg/m^{2}s)	Re No. of heating water	Re No. of cooling water
2	9.04	392.7	246.6
3	13.56	589.1	370.0
4	18.09	785.8	495.5
5	22.61	982.2	616.9
6	27.13	1178.5	740.2

R-22 was high, in order. But, since boiling points are 351 K and 337 K, in case of acetone and methanol, respectively and it was 305 K in case of R-141b, it is inappropriate for application when inlet temperature of heating water flow to evaporating part was decided to 297 K. Therefore, considering these points, thermal characteristics of R-22 were superior to compared with other refrigerants. This is the reason that thermal conductivity and density of liquid phase of R-22 is higher than those of another fluids although the latent heat of evaporation of R-22 was smaller than those of another working fluids used in this study.

Thus, among the working fluids of boiling point lower than 297 K, R-22 of high density of liquid phase, R-290 of large latent heat of vaporization, and R-142 of large viscosity of liquid phase were chosen as working fluids in OCHP and heat transfer performance was evaluated to working fluids. At this state, when the charging ratio of working fluid was 40 (vol.%) and 50 (vol.%), heat transfer performance was excellent from former OCHP researches. Therefore, 40 (vol.%) and 50 (vol.%) were chosen as the charging ratio of working fluid and heat transfer was evaluated for each charging ratio. And, bottom heating mode was chosen as heating mode in this study.

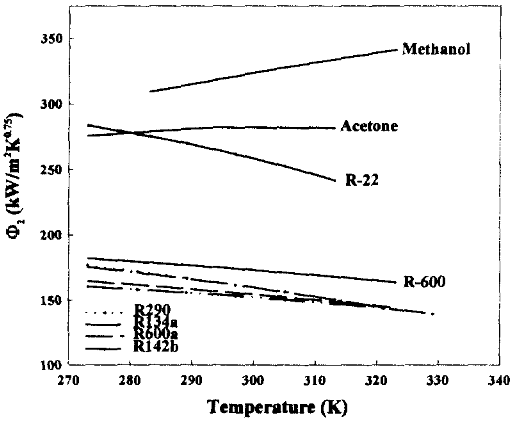


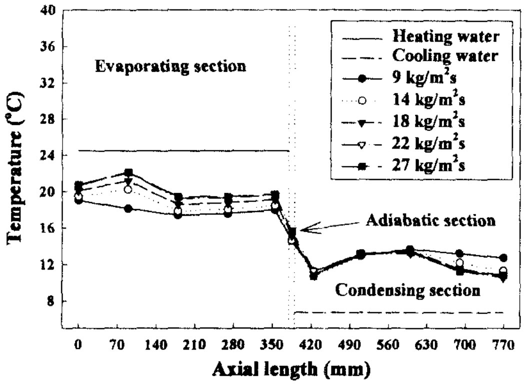
Fig. 4 Figure of merit for different working fluids in thermosyphon.

4. Experimental results and consideration

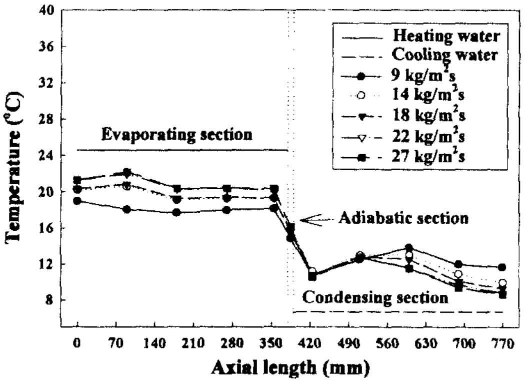
4.1 Wall temperature distribution of heat pipe

When the charging ratio was 40 (vol.%) and 50 (vol.%) of R-22, the surface temperature distribution of axial direction of heat pipe was represented in Fig. 5 (a) and (b) as a function flow rate of secondary fluid.

Generally, as the flow rate of secondary flow supplied to evaporating and condensing part was increased, surface temperature at evaporating part was ascended and surface temperature at the condensing part was descended and surface temperature at the adiabatic part was ascended.



(a) Charging ratio : 40 (vol.%)



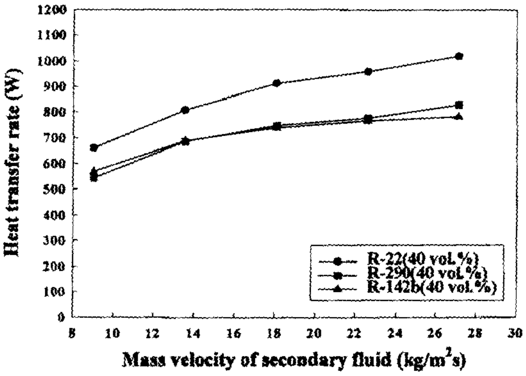
(b) Charging ratio : 50 (vol.%)

Fig. 5 Variation of wall temperature with mass velocity of secondary fluid.

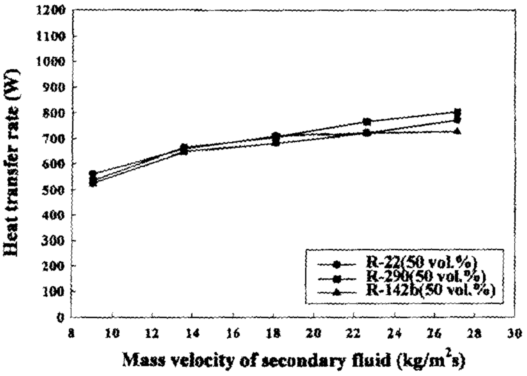
4.2 Heat transfer performance evaluation to working fluids and their charging ratio

When R-22, R-290, and R-142 were used as working fluids and their respective charging ratio was set to 40 (vol.%) and 50 (vol.%), heat transfer characteristics were represented in Fig. 6 (a) and (b) as a function of mass velocity of secondary flow which was supplied to evaporating and condensing parts.

As mass velocity of secondary fluid was increased, heat transport was also increased in three refrigerants. The heat transport of R-22 was higher than of those of R-290 and R-142b in same mass velocity. In heat transfer rate, there is a little difference in remaining two kinds of working fluids. This is agreed well with the results of Figure of Merit above referred and this suggested that Figure of Merit of thermosyphon could be applied to OCHP. But, it was difficult to compare with the ratio of inside diameter to capillary ascending length of working fluid proposed by Nishio et al.(11) because capillary ascending length of working fluids used in this study was not measured.



(a) Charging ratio : 40 (vol.%)



(b) Charging ratio : 50 (vol.%)

Fig. 6 Heat transfer rate with mass velocity of secondary fluid.

In Fig. 7, thermal resistance was represented to kinds of working fluid, their charging ratio and mass velocity of the secondary fluid, which was supplied to evaporating and condensing part. Thermal resistance is widely used as an index to represent heat transfer performance of heat pipe and it can be expressed to equation (3). Where, T_e and T_c are the average surface temperature of evaporating and condensing part of heat pipe.

$$R_{tot} = \frac{(T_e - T_c)}{Q} \tag{3}$$

As seen in experimental results, thermal resistance of charging ratio 40 (vol.%) of working fluid was lower than that of charging ratio 50 (vol.%). In case of R-22 of high performance evaluation index, thermal resistance was much lower than those of other working fluids. And, it was reconfirmed that heat transfer performance of R-22 was higher than those of

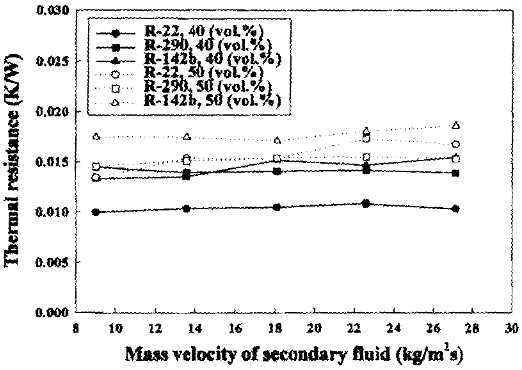


Fig. 7 Thermal resistance with mass velocity of secondary fluid.

two other refrigerants. And, there was not existed large difference in thermal resistance of R-141 and R-290 and this agreed to tendency given at Figure of Merit.

So, it was judged from that Figure of Merit can be used as an index for choice of working fluid because Figure of merit of working fluid for thermosyphon was proportion to heat transfer performance when bottom heating mode was applied to OCHP.

4.3 Comparison between experimental results and values predicted by correlation

When bottom heating mode was used to operate OCHP, even if gravity was not affect heat transfer performance in small sized tube, to some degree, gravity affect return process of condensed liquid to evaporating part from condensing part such as thermosyphon. And, as shown in above mentioned, condensed liquid in OCHP was returned to evaporating part not by wick but oscillation of working fluid.

In Fig. 8, when R-22 was used as working fluid and its charging ratio was 40 (vol.%) and 50 (vol.%), heat transfer rate measured in this study was converted to heat flux based on inside area of heat pipe and then it was compared with prediction correlations(13) for critical heat flux for thermosyphon which was proposed by Katto, Sakhuja, Tien & Chung and Imura et al.

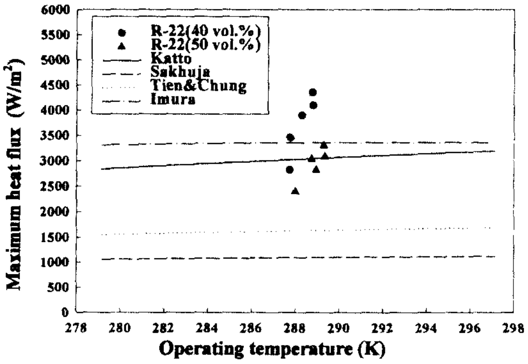


Fig. 8 Comparison of heat flux between experimental results and other correlations.

By the way, the charging ratio of working fluid applied to these correlations was not decided to values at the optimal operation conditions in which thermosyphon showed maximum heat flux. That is to say, if the optimal charging ratio range of working fluid of OCHP were 30~60 (vol.%),(6,8-12) usually, an optimal charging ratio of thermosyphon was decided at the small range compared with this range. Because of this reason, there is a difference on the heat transfer rate.(9) So, at the different optimal charging ratio state, it is unreasonable to compare critical heat flux of OCHP and thermosyphon. This is due to the difference of charging ratio of working fluid affecting directly to critical heat flux. Here, measured average surface temperature of adiabatic part was chosen as operation temperature, is a comparison basis. Inside diameter, the length of evaporation and condensing part were the same as conditions used in this study.

Within the operation temperature range 287~289 K, as a result of comparing predicted values based these correlations to experimental results, there is little difference between two values. But, among the predicted values, values predicted by Katto and Imura showed similar to this experimental results. This is the reason that correlations of Sakhuja, Tien & Chung were obtained on the assumption of incompressible vapor flow, constant liquid film thickness and homogeneous model at one dimensional steady state and thus is was a little different form real flow. On the other hand, In case of Imura's correlation which was well known for correlation to predict critical heat flux of thermosyphon within the low charging ratio, predicted value was about 10% as small as the experimental value when the charging ratio of working fluid R-22 was 40 (vol.%). But, the experimental result when the charging ratio was 50 (vol.%) was well agreed with value predicted by Katto within ±5%. It was judge that this is the reason that the value predicted by Katto showed similar value to some degree with this experimental result because Katto's correlation can be predicted very well when the charging ratio of working fluid was high. But, considering that heat transfer rate obtained through experiments when the charging ratio of R-22 was 50 (vol.%) was smaller than that of 40 (vol.%), critical heat flux of thermosyphon is a little small compared with oscillating heat pipe and as above mentioned, it was judged that this is due to the difference of charging ratio of working fluid to satisfy optimal operation condition of heat pipe.

5. Conclusions

The following conclusions were obtained through this study.

(1) Heat transfer performance when the charging ratio of working fluid 40 (vol.%) was higher than that of 50 (vol.%) in all working fluids.

(2) The heat transfer performance of R-22 was higher than those of two other refrigerants when R-142, R-22 and R-290 were used as working fluids of oscillating capillary tube heat pipe for low temperature.

(3) Heat rate of about 100 W per unit temperature difference was transported and based on this result, heat exchanger using OCHP developed in this study can be applied to waste heat recovery of low temperature difference for sea water and fish farm.

(4) It was found that to some degree, heat transfer performance of OCHP for low temperature is correlated to Figure of Merit, the performance evaluation index of working fluid for thermosyphon.

(5) Comparing heat transfer rate at the optimal state of OCHP with predicted value of critical heat flux of thermosyphon when R-22 was used as working fluid, heat transfer rate of oscillating heat pipe was a little higher than that of thermosyphon. But, to obtain more accurate result, many reviews are needed for thermal conditions and optimal flow of working fluid, of which heat transfer was the best.

Acknowledgements

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