# Study on Performance Evaluation of Oscillating Heat Pipe Heat Exchanger for Low Temperature Waste Heat Recovery

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**Key words**: Oscillating heat pipe (OHP), Waste heat recovery, Thermosyphon, Shell and tube heat exchanger, Thermal resistance, Figure of merit

**ABSTRACT**: The performance of heat exchanger using oscillating heat pipe (OHP) for low temperature waste heat recovery was evaluated. OHP used in this study was made from low finned copper tubes connected by many turns to become the closed loop of serpentine structure. The OHP heat exchanger was formed into shell and tube type. R-22 and R-141b were used as the working fluids of OHP with a fill ratio of 40 vol.%. Water was used as the working fluid of shell side. As the experimental parameters, the inlet temperature difference between heating and cooling water and the mass velocity of water were changed. The mass velocity of water was changed from 30 kg/m$^2$s to 92 kg/m$^2$s. The experimental results showed that the heat recovery rate linearly increased as the mass velocity and the inlet temperature difference of water increased. Finally, the performance of OHP heat exchanger was evaluated by $\varepsilon$-NTU method. It was found that the effectiveness would be 80% if NTU were about 1.5.

## Nomenclature

$A$ : area [m$^2$]
$C$ : heat capacity rate [kW/K]
$c_p$ : specific heat [kJ/kgK]
$D$ : diameter [m]
$G$ : mass velocity [kg/m$^2$s]
$h$ : fin height [m]
$h_{fg}$ : latent heat of evaporation [J/kg]
$L$ : interval of baffle [m]
$m$ : mass flow rate [kg/s]
$n$ : tube number
NTU : number of transfer unit
$p$ : tube pitch [m]
$Q$ : heat transfer rate [kW]
$R$ : thermal resistance [K/kW]
$S$ : surface [m$^2$]
$s$ : interval of fin [m]
$T$ : temperature [K]
$U_t$ : overall heat transfer coefficient [W/m$^2$K]
$w$ : fin thickness [m]

### Greeks symbols

$\varepsilon$ : effectiveness
$\phi$ : figure of Merit [Ws$^{0.5}$/m$^2$K$^{0.75}$]
$\rho$ : density [kg/m$^3$]
$\mu$ : viscosity [Ns/m$^2$]
$\lambda$ : thermal conductivity [W/mK]

### Subscripts

$b$ : baffle
$c$ : side of condenser

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*ex* : heat exchanger
*f* : fin
*h* : side of evaporator
*hp* : heat pipe
*i* : inlet
*l* : liquid
*min* : minimum
*o* : outlet
*t* : total
*w* : wall
1, 2 : column of tube

## 1. Introduction

The oscillating heat pipe (OHP) is a very promising heat transfer device.[1] In addition to its excellent heat transfer performance, it has a simple structure: in contrast with conventional heat pipes, there is no wick structure to return the condensed working fluid back to the evaporating part. The OHP is made from copper tubes connected by many turns to become the closed loop of serpentine structure as shown in Fig. 1. The working fluid is charged into the OHP. The diameter of the OHP must be sufficiently small so that vapor plugs can be formed by capillary action. The OHP is operated within a 0.1~5 mm inner diameter range. The OHP can operate successfully for all heating modes. The heat input, which is the driving force, increases the pressure of the vapor plugs in the evaporating part. In turn, this pressure increase will push neighboring vapor plugs and liquid slugs toward the condensing part, which is at a lower pressure. However, due to the continuous heating, small bubbles formed by nucleate boiling. The bubbles grow and coalesce to become vapor plugs. The flow of the vapor plugs and liquid slugs moves to the condensing part by pressure difference. The heat transfer continuously occurs. As a result, thermal energy is rapidly transferred from the evaporating part to the condensing part as well as the oscillation and circulation of liquid slugs and vapor plugs occurring in the OHP.[1,2]

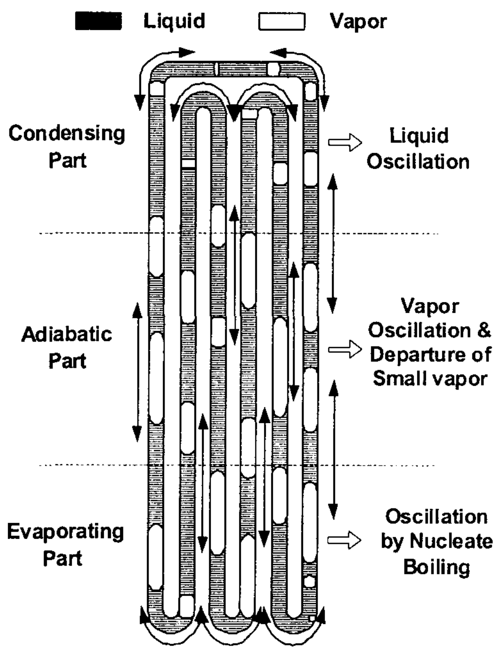



Fig. 1 Operation mechanism of oscillating heat pipe.

The experimental investigations on OHP have been carried out by visualization method to find the flow pattern and effective thermal conductivity of OHP. Kim and Lee et al.[3,4] reported that the oscillation of vapor bubbles, which caused by nucleate boiling, and the departure of small bubbles are considered as the representative flow patterns at the evaporating part and at the adiabatic part in OHP. They also concluded that the optimal operation state, which was the state that the oscillation of liquid slugs and vapor plugs was most active, was obtained at the charging ratio (fill ratio) of 40 vol.%. By the very rapid oscillation of vapor plugs and liquid slugs inside the tubes, a pseudo slug flow pattern near the annular flow pattern was formed. The thermal resistance between the surface of tube walls and the working fluid decreased and the heat transfer performance was enhanced.

When OHP are applied to a heat exchanger, the OHP heat exchanger has many advantages compared with other conventional heat pipe heat exchangers. It can increase its effectiveness by reducing the temperature difference between the evaporating and condensing part. By using low finned copper tubes, the thermal resistance between the fluid streams and the outside surface of OHP can be further reduced. Thus a finned OHP heat exchanger can transfer heat from one fluid stream to another fluid stream very effectively and it can be compact due to flexible, simple and small structure.

There are different methods for the evaluation of heat transfer performance of heat pipe heat exchangers. Kays and London[5] used the effectiveness-NTU (effectiveness-number of transfer units) method to predict the heat transfer performance of thermosyphon and heat pipe heat exchangers. Huang and Tsuei[6] numerically predicted the heat transfer performance of heat pipe heat exchanger by conduction model using finite difference method. Dunn and Reay[7] reported that the effectiveness of heat pipe heat exchanger increased when the length of heat pipe was in the range between 1 m and 6 m. When the tube diameter of heat pipe was increased from 12 mm to 19 mm, the effectiveness increased about 9%. Hsieh et al.[8] applied heat pipe heat exchanger for waste heat recovery. They represented the influence of tube array on the heat transfer performance of heat pipe heat exchanger. Lee and Bedrossian[9] studied on the characteristics of counter flow heat exchanger using heat pipe. They reported that maximum heat transfer rate has the relation with the flow rate ratio (between the hot and cold fluids) and the length ratio (between the evaporating part and the condensing part) of heat pipe.

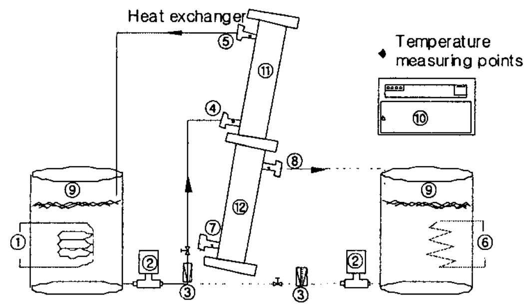
This study introduces a new design of heat pipe heat exchanger using OHP for low temperature waste heat recovery. The performance of OHP heat exchanger is investigated depending on various parameters such as mass velocities, inlet temperature difference, and working fluids. Finally, the heat transfer performance of OHP heat exchanger is evaluated by $\varepsilon$-NTU method. It was found that the effectiveness would be 80% if NTU were about 1.5.

## 2. Experimental apparatus and methods

### 2.1 Experimental apparatus

The schematic diagram of experimental apparatus is shown in Fig. 2. It consists of a test section, data acquisition system, and circulation systems of heating and cooling water.

The test section is an OHP heat exchanger as shown in Fig. 3. The data acquisition system was composed of a data logger (DR 230)




① Chiller ② Water pump ③ Flowmeter ④ Condenser inlet ⑤ Condenser outlet ⑥ Boiler ⑦ Evaporator inlet ⑧ Evaporator outlet ⑨ Constant temperature bath ⑩ Data logger ⑪ Condenser ⑫ Evaporator

Fig. 2 Schematic diagram of experimental apparatus.

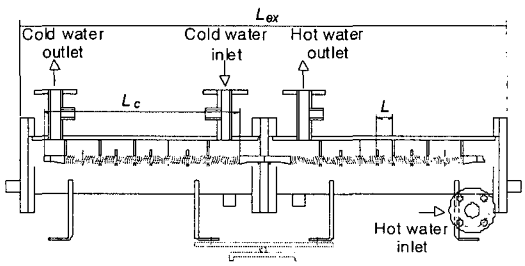



Fig. 3 Schematic diagram of OHP heat exchanger.

and personal computer. Each circulation system of heating and cooling water was composed of a constant temperature bath (9), water pump (2) and volumetric flow meter (3). The solid line indicates the circulation of cooling water system and the dotted line indicates the circulation of heating water system. The cooling and heating water system were maintained constant temperatures using chiller (1) and boiler (6), respectively.

The sheath type thermocouples were installed at the inlet and outlet of the heating and cooling water system to measure the temperatures of circulating water. Copper constantan thermocouples with an experimental uncertainty of ±0.2℃ were used for temperature measurements. Each sensor was calibrated to reduce experimental uncertainties and was connected to the data logger. The data were measured at a steady state for 100 times (at an interval of two seconds) and their average was used for data reduction.

The outer surface of the OHP heat exchanger was well insulated using insulating materials to prevent heat loss. The OHP was evacuated to $6.8\times10^{-6}$ Torr by using a high vacuum pumping system, which consists of a rotary and diffusion pump. R-22 and R-141b were used as the working fluids for OHP at charging ratio (fill ratio) of 40 vol.%. The charging cylinder (HPG-10, 96, Taiatsu Co.) of 10 mL was used to charge the working fluid exactly.

### 2.2 Structure of OHP heat exchanger

The schematic diagram of OHP heat exchanger was shown in Fig. 3. The specifications of the OHP heat exchanger were represented in Table 1.

The inlet and outlet nozzles (of the hot and cold fluids) on shell side of OHP heat exchanger have the same diameter of 50 mm. The OHP, which manufactured to the closed loop of serpentine type, was fixed into the heat exchanger. The baffles are spaced at constant intervals to conduct heat exchanger effectively.

Table 1 Specifications of OHP heat exchanger

| Heating mode | Bottom |
|---|---|
| $L_{ex}$ | 2.111 |
| $L_c$ | 0.89 |
| $L$ | 0.070 |
| Number of pass (EA) | 147 |
| Heat transfer surface area ($m^2$) | 12.75 |

Fig. 4 shows a schematic of the tube layout in OHP heat exchanger. The specifications of low finned copper tubes were represented in Fig. 5 and Table 2.

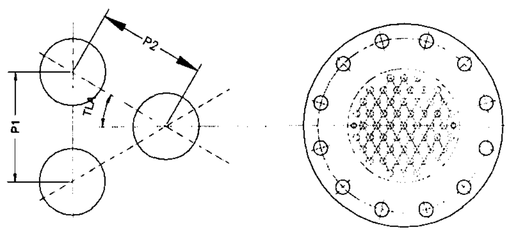



Fig. 4 Tube layout in OHP heat exchanger.

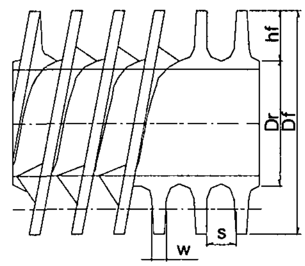



Fig. 5 Specifications of low finned tube.

Table 2 Dimensions of low finned copper tube

| Tube layout | Staggered |
|---|---|
| TLA (°)(triangular) | 30 |
| $P_1$ | 0.018 |
| $P_2$ | 0.018 |
| $w$ | 0.00048 |
| $s$ | 0.00115 |
| $h_f$ | 0.00193 |
| $D_r$ | 0.00714 |
| $D_f$ | 0.011 |

The minimum flow area for cross-flow in the plane diagonal adjacent to two rows in the staggered arrangements can be expressed by the following equation in case of outside finned tube[10]

$$S_{min} = 2n_t L\left\{p_2 - D_r - \frac{2wh}{(w+s)}\right\} \quad (1)$$

The mass flux at the core of actual OHP heat exchanger was calculated using the following Eq. (2).

$$G = \frac{m}{S_{min}} \quad (2)$$

### 2.3 Analysis of thermal resistance in OHP heat exchanger

The circuit diagram of thermal resistance in OHP heat exchanger is briefly presented in Fig. 6. Overall heat transfer resistance is composed of the convective thermal resistances $R_h$ and $R_c$ (between fins and water in evaporator and condenser), the conductive thermal resistances of fin side $R_{f,h}$ and $R_{f,c}$, the thermal resistances of wall side of OHP $R_{w,h}$ and $R_{w,c}$, and the axial conduction thermal resistance of OHP $R_{hp}$. Their relations can be expressed by Eq. (3).

$$R_t = R_c + R_{f,c} + R_{w,c} + R_{hp} + R_{w,h} + R_{f,h} + R_h \quad (3)$$

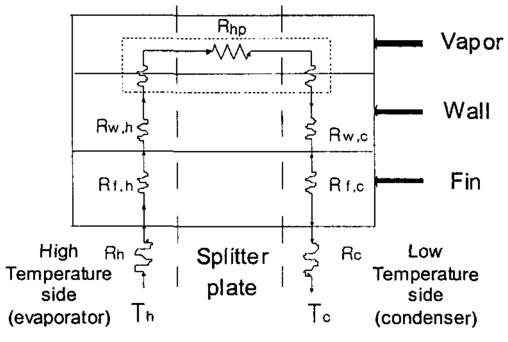



Fig. 6 Circuit diagram of thermal resistance in OHP heat exchanger.

### 2.4 Choice of working fluid for OHP

The fundamental principle of heat pipe is vaporization and condensation of working fluid. The choice of working fluid is an important factor for heat transfer performance and stable operation in heat pipes. Considerations in the choice of working fluid are thermal and physical properties of working fluid such as the effective working temperature range, vaporization pressure, vaporization latent heat, and viscosity coefficient.

In thermosyphons, the working fluid is chosen based on Figure of Merit as expressed by Eq. (4). A large Merit index means a high heat transfer performance. This index can be used to evaluate effectiveness of various working fluids in case of heat pipes operating under the influence of gravity at specific temperature.[11]

$$\phi = \left[\frac{h_{fg}\lambda_l^3 \rho_l^2}{\mu_l}\right]^{0.25} \quad (4)$$

At the normal temperature, the performance index of R-718, R-717, R-22, R-290, R-141b, and R-600a are generally represented in Fig. 7. Except for water and ammonia, the perform-

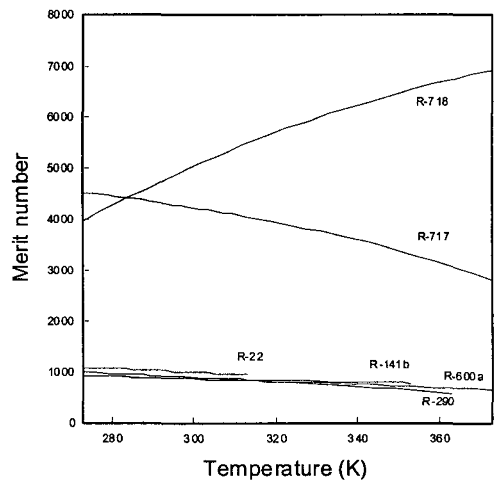



Fig. 7 Figure of Merit for various working fluid in thermosyphon.

Table 3 Experimental conditions

| Working fluid | R-141b | R-22 |
|---|---|---|
| Difference of inlet temperature (K) | 10, 15 | 10, 15 |
| Flow rate of water (ton/h) | 2~6 | 2~6 |
| Inlet temperature of evaporator (K) | 304, 309 | 304, 309 |
| Inlet temperature of condenser (K) | 294 | 294 |

ance index of R-22 is higher (compared with other refrigerants) around the operating temperature of 300 K because its thermal conductivity and density of liquid phase are high although the latent heat of evaporation is smaller than that of the other refrigerants.

For smooth operation at low temperature difference, the pressure difference between evaporating and condensing part in OHP has to be considered. Therefore, R-22 and R-141b are chosen as the working fluid to evaluate heat transfer performance of OHP heat exchanger in this study. Im and Ahn et al.[12,13] also proposed using R-22 and R-141b as the working fluids for smooth operation of heat pipe heat exchanger in low temperature waste heat recovery.

### 2.5 Experimental conditions

Experimental conditions were represented in Table 3. The inlet temperature difference of water into evaporator and condenser was constantly maintained at temperatures of 10 K and 15 K. The flow rates of water supplied to evaporator and condenser were adjusted from 2 ton/h to 6 ton/h to get optimal outlet temperatures and to recover waste heat effectively at low temperature difference.

## 3. Experimental results and considerations

### 3.1 Heat transfer performance to mass velocity

When R-22 and R-141b were used as working fluids of OHP and the difference of inlet temperature was constantly maintained at 10 K and 15 K. The variation of heat transfer rate of OHP heat exchanger with respect to mass velocity of secondary fluid is showed in Fig. 8. The heat transfer rate was calculated by Eq. (5).

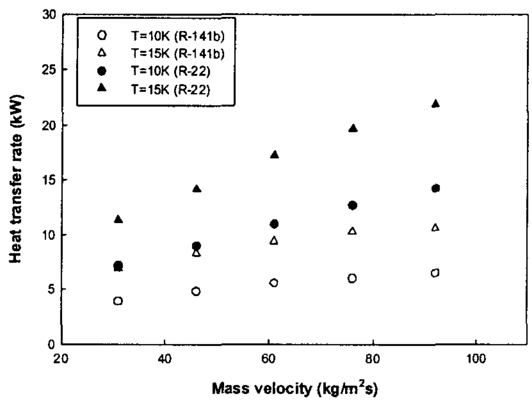



Fig. 8 Heat transfer rate with respect to mass velocity of secondary fluid.

$$Q = mc_p \Delta T \quad (5)$$

As mass velocity was increased, the heat transfer rate linearly increased due to the decrease of convective thermal resistance between the fin surface and secondary fluid. The heat transfer rate of R-22 was about two times higher than that of R-141b. It was due to the fact that the performance index of R-22 is higher than that of R-141b as shown in Fig. 7.

### 3.2 Heat transfer performance to inlet temperature difference

The influence of inlet temperature difference on the heat transfer rate of OHP heat exchanger is shown in Fig. 9. As the temperature

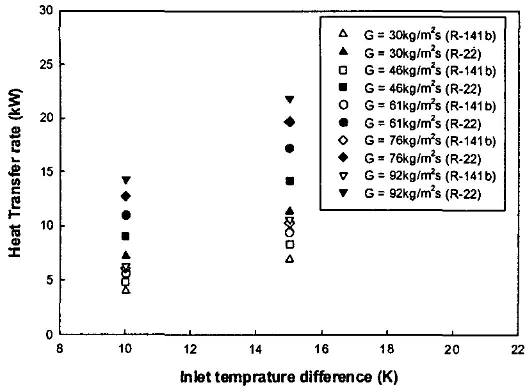



Fig. 9 Heat transfer rate with respect to inlet temperature difference.

difference was increased, the heat transfer rate increased. The heat transfer rate of R-22 is also higher than that of R-141b as mentioned above.

From experimental results on OHP by flow visualization method, it can be explained that when inlet temperature difference was increased, the heat added into the evaporating part of OHP increased. Bubbles were continuously generated in the evaporating part of OHP. These bubbles coalesced to become vapor plugs and the active oscillation phenomenon occurred between the evaporating and condensing parts. The thermal resistance of OHP, $R_{hp}$, decreased. Consequently, the heat transfer performance of OHP heat exchanger was enhanced.

### 3.3 Overall heat transfer coefficient to mass velocity

Overall heat transfer coefficient of OHP heat exchanger is obtained using Eq. (6).

$$U_t = \frac{mc_p \Delta T}{A \Delta T_m} \tag{6}$$

where, $\Delta T_m$ is the log mean temperature difference. It is calculated by Eq. (7) by measuring the inlet and outlet temperatures of heating and cooling water system.

$$\Delta T_m = \frac{(T_{h,i} - T_{c,i}) - (T_{h,o} - T_{c,o})}{\ln\left(\dfrac{T_{h,i} - T_{c,i}}{T_{h,o} - T_{c,o}}\right)} \tag{7}$$

The influence of mass velocity on overall heat transfer coefficient of OHP heat exchanger is shown in Fig. 10. When mass velocity was increased, the overall heat transfer coefficient increased. Also the overall heat transfer coefficient of R-22 was much higher than that of R-141b.

Increasing the mass velocities in OHP heat exchanger led to the decrease of the convective thermal resistances $R_h$ and $R_c$ (between fins and water in evaporator and condenser). The temperature difference between the evaporating and condensing part of OHP increased. This led to an increase in the pressure difference between the evaporating and condensing part of OHP. The oscillation of liquid slugs and vapor plugs occurred very actively. The thin liquid film was formed on the wall of the tubes due to the very rapid oscillation of liquid slugs and vapor plugs. The thermal resistance between the surface of tube walls and the working fluid decreased and the heat transfer performance was enhanced.

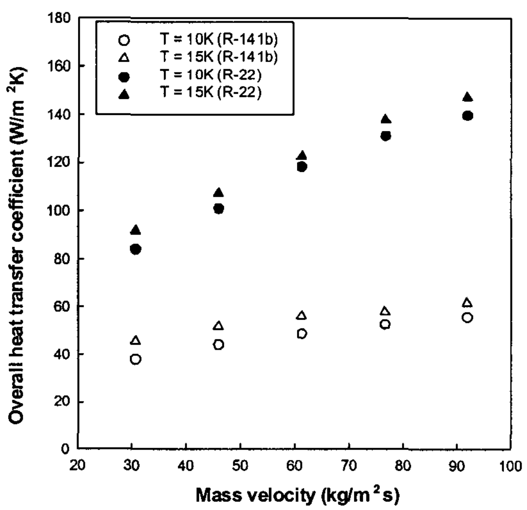



Fig. 10 Influence of mass velocity on overall heat transfer coefficient.

### 3.4 Performance evaluation of OHP heat exchanger by using $\varepsilon$-NTU method

The OHP heat exchanger is evaluated using $\varepsilon$-NTU method. The variation of effectiveness with respect to NTU can be expressed by the following equation.

$$\varepsilon = 1 - e^{-NTU} \tag{8}$$

where,

$$\mathrm{NTU} = \frac{U_t A}{C_{min}} \tag{9}$$

The overall heat transfer, $U_t$, can be found from Eq. (6) and the minimal heat capacity ratio, $C_{min}$, can be found by the following equation.

$$C_{min} = mc_p \tag{10}$$

The effectiveness of OHP heat exchanger with respect to NTU is shown in Fig. 11. Good agreement of the experimental results and the values calculated form Eq. (8) indicates that the OHP heat exchanger can be evaluated using $\varepsilon$-NTU method.

As shown in Fig. 11, the effectiveness of OHP heat exchanger would be 80% if NTU were about 1.5.

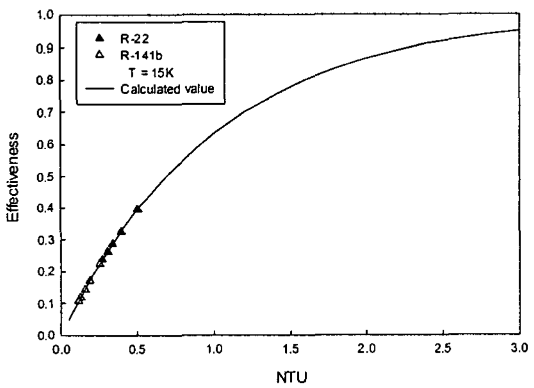



Fig. 11 Effectiveness of OHP heat exchanger with respect to NTU.

## 4. Conclusions

The following conclusions were obtained through the experiments and the evaluation of heat exchanger using oscillating heat pipe.

(1) The new waste heat recovery of high performance and easy in removal of fouling was developed.

(2) As the mass velocity and inlet temperature difference of secondary fluid were increased, the heat recovery rate was linearly increased. The heat recovery rate of R-22 is two times higher than that of R-141b under the same working conditions.

(3) Figure of Merit representing the performance of working fluid in thermosyphon is not agree with the waste heat recovery of OHP heat exchanger at low temperature difference.

(4) The effectiveness of OHP heat exchanger can be evaluated using $\varepsilon$-NTU method. The effectiveness would be 80% if NTU were about 1.5.

## Acknowledgements

This work was performed with financial support of Korea Science and Engineering Foundation (98M4-1506-00-01-2) and Research Center for Ocean Industrial Development at Pukyong National University. We gratefully acknowledge the support.

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