Effect of working-fluid filling ratio and cooling-water flow rate on the performance of solar collector with closed-loop oscillating heat pipe†

Kim-Bao Nguyen1, Seok-Hun Yoon2 and Jae Hyuk Choi2,*

1Graduate School of Marine System Engineering Korea Maritime University, Busan, 606-791, Korea
2Division of Marine System Engineering Korea Maritime University, Busan, 606-791, Korea

(Manuscript Received February 8, 2011; Revised September 7, 2011; Accepted October 16, 2011)

Abstract

This study experimentally investigated the effect of the working-fluid filling ratio (FR) and the cooling-water flow rate (CWFR) on the top heat loss and the performance of a solar collector equipped with a closed-loop oscillating heat pipe (CLOHP). The CLOHP was composed of a heating section, a cooling section, and an adiabatic section; it had a 0.002-m internal diameter and eight turns. The heating section was attached to a copper plate coated with black chrome, which absorbed energy from a solar simulator that had 12 halogen lamps and was controlled by a voltage regulator. The cooling section was inserted into the collector’s cooling box, which was made of a transparent acrylic plate. The FR of the working fluid ranged from 30% to 80% with a 10% interval, and the CWFRs were 0.15 l/min, 0.30 l/min, and 0.45 l/min. The experimental results show that the solar collector equipped with the CLOHP has good performance at working-fluid FRs of 60% and 70% with low flow rates of 0.15 l/min and 0.30 l/min.

Keywords: Filling ratio (FR); Cooling-water flow rate (CWFR); Closed-loop oscillating heat pipe (CLOHP); Solar simulator

1. Introduction

A closed-loop oscillating heat pipe (CLOHP) is a highly efficient device for heat transport. In general, it consists of an evacuated tube filled with a certain amount of working fluid. Heat transportation occurs through continuous evaporation and condensation of the working fluid. It gives high performance because of the high latent heat of vaporization and condensation of the working fluid and also has a simple structure and a fast thermal response. During the past decade, some researchers have studied the thermal transfer characteristics of the oscillating heat pipes (OHPs) [1-10].

Recently, in view of the preference for environment-friendly energy systems, the importance of solar energy applications such as solar collectors has increased, and some studies have been conducted on characteristics and performance of solar collectors [3-10]. Rittidech et al. [3] assessed the performance of a solar collector by using CLOHPs. Their results confirmed that the anticipated fluctuation in the efficiency of the collector depended on the time of day, solar irradiation, ambient air temperature, and mean temperature of the absorber plate. Piyanun et al. [4], who measured the performance of horizontal CLOHPs (HCLOHPs), showed that the operation start-up temperature of an HCLOHP greatly depends on the evaporator temperature that relates to the number of turns. The minimum operation start-up temperature of 50°C only occurred for a maximum number of 26 turns. The proper working fluid for an HCLOHP with a 2-mm inner diameter was water, whereas for a 1-mm-inner-diameter HCLOHP, both water and ethanol were acceptable. Schreyer et al. [5] experimentally investigated the use of refrigerant R11 in a thermosyphon solar collector for residential applications. The results showed that for two identical collectors, the maximum instantaneous efficiency of a collector charged with a boiling refrigerant was 6% larger than that of a solar collector circulating a hydraulic fluid. Hammad et al. [6] measured the performance of a solar collector using a heat pipe, which functioned under low-temperature conditions. Their experimental results indicated that the collector had an efficiency of about 60%, which was comparable that of a water-cooled collector. Mahasudan et al. [10] conducted a study on the minimization of heat loss in normal and reverse flat-plate collector configurations. The heat loss was minimized by optimizing the absorber plate to cover glass separation, providing a low-emissivity surface on the collector’s back face and by placing an additional reflector supported on glass wool insulation behind the back face.

Most heat loss in commercial flat-plate solar collectors oc-
curs due to the convective and radiative heat transfer from the absorber surface to the glass cover, and it greatly depends on the temperature of the absorber plate [11]. Thus, to enhance the collector efficiency the convective and radiative heat loss from the collector must be minimized.

In the present study, the heating section of a collector is mainly composed of a CLOHP and an absorber plate. Therefore, the heat transport capability of the heat pipes is strongly influenced by the temperature of the absorber plate, and the temperature of each part of the collector depends on the FR of the working fluid and the operating condition of the solar collector. On the other hand, the heat transport capability of the CLOHP depends on the FR of the working fluid and the cooling-water flow rate (CWFR).

To obtain basic information on the heat loss and perform- ance of a flat-plate solar collector equipped with a CLOHP, the heat transport characteristics of such a solar collector were investigated for various FRs of working fluid, solar intensities, and CWFRs. In addition, the heat loss of the solar collector was analyzed to determine the details of the heat loss mechanism in the solar collector.

2. Analysis of top heat loss

Fig. 1 shows a schematic diagram of the heat loss components of a flat-plate solar collector. The heat loss to the surrounding is an important characteristic of a flat-plate solar collector. In general, in a flat-plate solar collector, heat is lost from the absorber plate through the glass cover, the back insulation, and the edge, and they are indicated as the top loss, bottom loss, and edge loss in Fig. 1, respectively. The heat loss occurs by all mechanisms as conductive, convective, and radiative heat transfer. The back loss and the edge loss can be minimized by proper selection of insulation material and its thickness. Therefore, the most important heat loss in a flat-plate solar collector is top heat loss. Procedures to determine the top heat loss of a flat-plate solar collector are provided below.

2.1 Convective heat transfer coefficient

The convective heat transfer coefficient, $h_{1c}$, for the heat transfer from the absorber plate to the cover glass that is inclined to the horizontal can be determined as follows [11]:

$$h_{1c} = \frac{\text{Nu} \cdot K}{L}.$$  

The Nusselt number, Nu, for air between the absorber plate and the cover is calculated as follows [11]:

$$\text{Nu} = 1 + 1.44[a]^+ [b] + [c]^+$$  

with

$$a = 1 - \frac{1708}{\text{Ra cos} \beta} + \frac{\sin(1.8 \beta)}{\text{Ra cos} \beta} - \frac{1708 b}{\text{Ra cos} \beta} - 1.$$  

The superscript “+” in formula (2) implies that only a positive value of the term in square brackets is to be considered, and the term is to be considered as zero if it has a negative value. Further, the angle of inclination, $\beta$, varies between 0°-75°. The Rayleigh number, Ra, is given as [11]

$$\text{Ra} = \frac{g \beta \Delta T L^3}{v \alpha}.$$  

If $75^\circ \leq \beta \leq 90^\circ$, $\text{Nu}$ is determined as [11]

$$\text{Nu} = [1, d, e]_{\text{max}}$$

with

$$d = 0.288 \left( \frac{\sin \beta \text{Ra}^{1/4}}{A} \right)^{1/3}, \quad e = 0.039(\sin \beta \text{Ra})^{1/3}.$$  

The Nusselt number in formula (4) is the maximum value of the three quantities separated by commas, and $A$ denotes the ratio of the length of the inclined collector plate to the spacing between the cover and the absorber plate.

The convective heat loss coefficient from the cover to the ambient air, $h_{2c}$, is given as [11]

$$h_{2c} = 2.8 + 3 V.$$  

where $V$ denotes the wind speed over the collector.

2.2 Radiative heat transfer coefficient

The radiative heat loss coefficient from the plate to the cover, $h_{rm}$, can be given as

$$h_{rm} = \varepsilon_\sigma \left( T_p^4 + 273^4 - (T_p + 273)^4 \right) \frac{1}{T_p - T_r}.$$  

Fig. 1. Heat losses in a flat-plate solar collector.
The effective emissivity of the plate-cover system, $\varepsilon_{pg}$, is calculated as

$$\varepsilon_{pg} = \left( \frac{1}{\varepsilon_{pg}} - 1 \right)^{-1}.$$  (7)

The radiative heat loss coefficient from the cover to the ambient air, $h_{2r}$, can be given as

$$h_{2r} = \varepsilon_{pg} \left( \frac{(T_e + 273)^4 - (T_r + 273)^4}{T_e - T_r} \right).$$  (8)

with

$$T_e = T_a - 6.$$  (9)

where $\varepsilon_{pg} = 0.90/0.94$ [12] depends on the temperature of the cover, and $\varepsilon_{pg} = 0.090$ [13].

### 2.3 Top loss coefficient

The total heat loss coefficient from the plate to the cover is

$$h_t = h_{1r} + h_{2r}.$$  (10)

and that from the cover to the ambient is

$$h_2 = h_{2r} + h_{2r}.$$  (11)

The total top heat loss coefficient from the plate to the ambient is given as

$$U_t = \left( \frac{1}{h_t} + \frac{1}{h_2} \right)^{-1}.$$  (12)

The total top heat loss of the collector is determined as

$$Q_t = U_t A (T_p - T_r).$$  (13)

To calculate the top heat loss from the plate, all heat loss components must be known. The formulas (1) to (13) show that the top heat loss of the collector depends on the construction parameter (air gap thickness, $L$); environment parameters (wind velocity, $V_1$ and ambient air temperature, $T_a$); temperature of the absorber plate, $T_p$, and that of the glass cover, $T_g$. Furthermore, the collector used the CLOHP to transport the thermal heat energy from the heating section to the cooling section. Therefore, its top heat loss, which is affected by the heat transport characteristics of the heat pipes, depends on the FR of its working fluid and the CWFR. That is why the solar collector equipped with the CLOHP was designed and constructed to thoroughly investigate the effect of the FR and the flow rate of cooling water on the top heat loss and fully assess the performance of such a solar collector.

### 3. Experimental setup and procedures

Fig. 2 shows a schematic of the experimental setup. The experimental setup consists of the body of the solar collector equipped with a CLOHP; a data logger; a pressure transducer for measuring the internal vacuum pressure of the CLOHP; a flow meter for cooling water; K-type thermocouples to measure the temperatures of the absorber plate, glass cover, and ambient air; and a water bath to maintain the temperature of the cooling water constant. The temperature of the cooling water in the water bath was set to 20°C before starting the experiment. Before the working fluid was charged into the heat pipe, the pressure inside the heat pipe was reduced to a vacuum pressure of $10^{-3}$ kPa by using a diffusion-type vacuum pump. The vacuum pressure inside the heat pipe, $P_1$, was measured again after one week with a pressure transducer and recorder to ensure that the vacuum was maintained. Measuring points for temperatures of the absorber plate and the glass cover are marked in Fig. 2. Two thermocouples, $T_1$ and $T_2$, were attached to adjacent copper tubes in the adiabatic section. All thermocouples for measuring temperature were connected to an MV2000 Yokogawa recorder.

Fig. 3 shows the detailed dimensions and configuration of the collector. The collector consisted of a heating section, an adiabatic section, and a cooling section. The height of the heat pipe was 0.465 m; that of the heating section, 0.245 m; that of the adiabatic section, 0.065 m; and that of the cooling section, 0.155 m. The CLOHP was made of copper tubes, and its heating section was attached to the absorber plate that was illuminated by the solar simulator. Twelve 300 W halogen lamps were used to simulate solar energy. The cooling section of the heat pipe was inserted into the collector’s cooling box, which was made of transparent acrylic plate. Aluminum was used to fabricate the frame of the collector. The cover was made of...
transparent glass and sealed with silicone. The bottom and four sides of the heating section of the collector were insulated by poly-urethane foam. Aluminum foil was also used to minimize the radiative heat loss through the bottom side. The solar simulator was fixed and arranged parallel with a controllable stand. The radiation intensity of the solar simulator was adjusted via the voltage regulator and recorded by means of the Yokogawa recorder via an LP-PYRA-50 pyranometer. In this study, it was adjusted to inclination angle of 30°. Solar irradiation intensity was adjusted to 545 W/m², 645 W/m², 718 W/m², and 825 W/m². The working fluid was filled with a filling ratio of 30% to 80% with a 10% interval. The CWFRs were 0.15 l/min, 0.3 l/min, and 0.45 l/min. The temperatures of absorber plate, glass cover, cooling water, and ambient air were determined in the steady-state condition.

4. Results and discussion

As shown in Fig. 2, two thermocouples were attached to adjacent copper tubes in the adiabatic section to confirm whether the copper tubes function as oscillating heat pipes or not. In a CLOHP, vapor bubbles alternating with liquid slugs circulate actively, thus leading to different temperatures at different positions. If the pressure inside the copper tubes changes intensively, the temperatures at two points on two adjacent copper tubes are always changed alternately. This implies that if the temperature of one tube is sometimes higher than that of the other tubes and vice versa, the copper tubes will work as oscillating heat pipes. If this phenomenon does not happen, it also implies that the working fluid inside the copper tubes does not circulate and the copper tubes will not function as a CLOHP. Therefore, the thermal energy in a heat pipe cannot be transported from the heating section to the cooling section; consequently, the collector does not function effectively.

Fig. 4 shows the effect of FRs on the temperature changes in the absorber plate (a) and the glass cover (b). In Fig. 4(a) and (b), it is found that the temperatures of the absorber plate for two cases depend strongly on the working-fluid FR. In Fig. 4(a), while the temperature of the absorber plate is very high at FRs of 30%, 40%, and 50%, it is low at FRs of 60% or 70%. The temperature of the absorber plate increases again at FRs of 70% or 80%. In Fig. 4(b), the temperature changes in the cover are not as large as those in the absorber plate for a given FR. This implies that there is no alternate formation of vapor bubbles and liquid slugs inside the heat pipes at FRs of 30% and 40%. In this case, the heat pipes contain only vapor, and the temperatures at two points on the copper tubes cannot change alternately or the temperature of one point is always higher than that of others. If a dry-out phenomenon occurs, the copper tubes cannot be used as a CLOHP. Therefore, thermal energy cannot be transported from the heating section to the cooling section. It stagnates inside the heating section and consequently causes the temperatures of the plate and the cover to be very high at FRs of 30% and 40%. However, in our previous study [7], a heat pipe exhibited the best performance when the FR of the working fluid was 30%.

For this study, the length of the heating section of the heat pipes is relatively greater than that in our previous study [7]. Therefore, it requires more working fluid to alternately generate vapor bubbles and liquid slugs or to avoid the dry-out phenomenon from occurring inside the heat pipes.

To understand the temperature changes at two points on the copper tubes and pressure inside the CLOHP, the fluctuations in those temperatures are plotted versus time at a solar irradiation of 545 W/m².
tion intensity of 645 W/m² in Fig. 5. In the figure, \( T_1 \) and \( T_2 \) denote the temperatures at the two points on the copper tubes, as marked in Fig. 2, and \( P \) indicates the internal pressure of the CLOHP at \( P_1 \).

As it is seen in Fig. 5(a) and (b), the internal pressure of the CLOHP remains almost constant. This implies that the dry-out phenomenon occurs inside the CLOHP, and there is no alternate formation of vapor bubbles and liquid slugs at FRs of 30% and 40%. On the other hand, in the case of high FRs((c)~(f)), the collector operates more stably and intensively, except for an FR of 50%. This is because at high FRs, a sufficient amount of the working fluid is available to alternately generate vapor bubbles and liquid slugs. Further, at FRs of 60% and 70%, the temperatures \( T_1 \) and \( T_2 \) on the heat pipes change very quickly. In addition, the pressure inside the heat pipes oscillates quite rapidly, and the magnitude of the pressure change increases with the FR until the FR becomes 70%. The heat pipes work more intensively due to these phenomena. Consequently, as shown in Fig. 4, the absorber plates at different FRs have different temperatures.

Fig. 6 shows the top heat loss coefficient of the collector versus the working-fluid FR. The results show that the top heat loss coefficient of the collector greatly depends on the FR. An interesting feature is that the top heat loss coefficient of the collector has almost the same value at FRs of 30%, 40%, and 50% and is much higher than that at high FRs of 60%, 70%, and 80%. This is because at the FRs of 30% and 40%, dry-out occurs, as described above, and the FR of 50% is insufficient for intensive working of the heat pipes.

Both the temperature of the absorber plate and the top heat
loss coefficient are high due to this phenomenon at the FRs of 30%, 40%, and 50%. In Fig. 6, the top heat loss coefficient decreases dramatically at the FR of 60% with solar irradiation intensities of 545 W/m² and 645 W/m². Otherwise, in the case of solar intensities of 718 W/m² and 825 W/m², it decreases dramatically at the FR of 70%. These results indicate that the working-fluid filling ratios of 60% and 70% are suitable for intensive and effective operation of the CLOHP.

Fig. 7 shows the distribution of the top heat loss rate of the collector versus time at different FRs and solar radiation intensities. In this figure, an interesting feature is that the top heat loss rate of the collector exhibits the minimum value at the FR of 60% or 70% for all solar irradiation intensities. On the other hand, a low FR causes the temperature of the absorber plate to be high; hence, the top heat loss coefficient is high. The high FR has a minimum value of the top heat loss rate of the collector because the heat pipe used in this study requires more working fluid to alternately generate vapor bubbles and liquid slugs since the length of the heating section of the heat pipes is large.

The effect of the FR on the performance of the collector can be estimated by the ratio of the top heat loss to the incident solar energy, \( Q_t/(I \cdot A_c) \), and predicted from the thermal efficiency. The ratio of the top heat loss to the incident solar energy and the thermal efficiency versus the FR are shown in Fig. 8 and Fig. 9, respectively. In Fig. 8, the ratio of the top heat loss to the incident solar energy has different values for different working-fluid FRs. This implies that the FR is a dominant factor in determining this ratio. Further, this ratio depends on not only the FR but also the solar irradiation intensity. In particular, at solar intensities of 545 W/m² and 645 W/m², the FR of 60% is sufficient to alternately produce vapor bubbles and liquid slugs. However, when solar intensities are 718 W/m² and 825 W/m², the working fluid inside the heat pipe is vaporized more quickly, and hence, the heat pipe needs more working fluid. Therefore, the FR of 70% is suitable for the heat pipes. At the FR of 80%, the ratio of the top heat loss to the incident solar energy is smaller than that at 50%, which indicates that the collector operates more effectively at the FR of 80% than at the FR of 50%.

Fig. 9 shows the thermal efficiency of the collector versus the working-fluid FR. The thermal efficiency attains the maximum value at the FR of 60% or 70% when the solar intensities are 545 W/m², 645 W/m², and 718 W/m². From Fig. 8 and Fig. 9, it can be concluded that the higher the ratio of the top heat loss to the incident solar energy, the more inferior the thermal performance of the collector.
Fig. 10 shows the effect of the FRs on the thermal efficiency of the collector for three solar intensities. In this figure, the values of thermal efficiency at 0.15 l/min and 0.30 l/min are higher than that of 0.45 l/min for the most part. A higher flow rate of cooling water creates an imbalance between vapor bubbles and liquid slugs because the bubbles in the cooling section of the heat pipes can be collapsed so quickly. In the CLOHP thermal energy can be transported mainly by sensible heat of liquid slugs, and liquid slugs are transported from the heating section to the cooling section by a pressure wave of the vapor bubbles. In the case of high CWFRs, there is a lack of vapor bubbles to generate a pressure wave because the bubbles collapse very quickly. Therefore, the heat transfer capability of the heat pipe and the thermal efficiency of the collector decrease.

5. Conclusions

This study investigated the effect of the working-fluid filling ratio (FR) and the cooling water flow rate (CWFR) on the top heat loss and performance of a flat-plate solar collector equipped with a CLOHP. The working-fluid FR has a dominant effect on the top heat loss and performance of the collector. The top heat loss coefficient and the top heat loss of the collector depend on both the FR and the solar radiation intensity. When the solar radiation intensities are 545 W/m², 645 W/m² and 718 W/m², the collector has good performance at filling ratios of working fluid of 60% and 70% due to the relatively long length of heating section of the heat pipe as compared with the previous researches. The dry-out phenomenon occurs at FRs of 30% and 40%. The CLOHP solar collector operates more effectively at filling ratios from 60% to 70%. Flow rates of cooling water of 0.15 l/min and 0.30 l/min give better performance than that of 0.45 l/min.

References


Kim-Bao Nguyen received his B.S. from Vietnam Maritime University in 2006 and M.S degrees from Korea Maritime University, Korea, in 2010. He has worked for VIMARU from 2006 to 2008.

Seok Hun Yoon received his B.S. and Ph.D degrees from Korea Maritime University. He is currently professor at the division of marine system engineering in Korea Maritime University, Korea. His research interests are thermal engineering, solar energy et al.

Jae Hyuk Choi received his B.S and M.S degrees from Korea Maritime University in 1996, 2000 and Ph.D degree from Hokkaido university in 2005. He is currently a professor in Korea Maritime University. Dr. Choi’s research interests are reduction of pollution emission (soot and NOx), high temperature combustion, laser diagnostics, alternative fuel, heat pipe and hydrogen production with high temperature electrolysis steam (HTES).