An experimental study on a sodium loop-type heat pipe for thermal transport from a high-temperature solar receiver

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Abstract

A loop-type heat pipe was fabricated and tested to transport high-temperature thermal energy from a solar receiver in a CSP application. The purpose of the heat pipe in this study was to transport an 800-W thermal load at 1000 K to a specific energy conversion device 0.5 m in distance from the solar receiver. The container wall and transport lines of the loop-type heat pipe were made of stainless steel 304, and the working fluid was sodium. The evaporator and condenser were disk-type containers with diameters of 122 and 216 mm, respectively, though both had a height of 20 mm. The diameters of the vapor and liquid lines were 12.7 and 9.53 mm, respectively. The total length of the loop was approximately 1.4 m. A pillar structure with a diameter of 43 mm and height of 114 mm was installed in the center region of the condenser to interface with an AMTEC device that had a unique geometry. It was desired that the upper surface of the condenser as well as the whole outer surface of the pillar would have temperatures as close as possible during heat discharge. As one of the unique features in this study, the liquid line was located close to the vapor line for a preheating effect to reduce the frozen-startup time. The influence of the fill charge ratio of the working fluid as well as operating conditions such as the thermal load and the cooling condition in the condenser were analyzed with respect to the performance indices of the heat pipe, including the effective thermal conductance, the thermal resistance, and the maximum temperature difference. The optimum fill charge ratio of the heat pipe was 32\% based on the evaporator volume.

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Keywords: Loop heat pipe; sodium; experiment; pillar structure; vapor injector; fill charge; effective thermal conductance

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1. Introduction

Concentrated solar power (CSP) systems include many components that should operate in high-temperature environments. In some CSP applications, the thermal energy collected by the solar receiver must be transferred to a direct thermal-to-electricity energy conversion device such as a Stirling engine or alkali-metal thermal to electric converter (AMTEC), which may have to be located a certain distance apart from the receiver, depending on the design of the specific system [1, 2]. Such thermal transport systems typically require a small temperature drop, a high heat flux, operational reliability, and the least possible amount of power consumption. The purpose of this study was to investigate the performance of a loop-type heat pipe when it was applied to a thermal transport device between a high-concentration solar receiver and an AMTEC device with a specified interface geometry.

Heat pipes can achieve extremely high thermal transport capabilities utilizing a liquid-vapor phase change of the working fluid. For a high-temperature application over 450°C (723 K), however, the working fluid must be a liquid metal. A significant number of studies can be found in the literature dealing with liquid metal heat pipes for solar receiver applications [3-6]. However, most of them deal with heat pipes embedded into or attached to the receiver to improve the isothermal characteristics of the receiver wall or to enhance heat transfer with the heated fluid (thermal carrier). Relatively little information is available on either thermal transport to a distant location or the performance of a loop-type heat pipe for CSP applications. In the study by Wu et al., for example, although a capillary pumped loop (CPL) – a kind of heat pipe – was included as a thermal transport device in a dish/AMTEC system, no details were provided as to its configuration or performance [7].

In comparison with the traditional type of heat pipe, which is the most popular single-container geometry, the loop-type heat pipe has the potential to enhance thermal transport capabilities by separating the liquid and vapor lines and thus reducing the fluid dynamic resistance at the liquid-vapor interface that results from liquid and vapor flowing in the opposite direction to one another. Furthermore, the capillary structure can be removed in the vapor line. As a result, a well-designed loop-type heat pipe can achieve higher thermal transport capabilities than a traditional heat pipe having the same diameter in transportation lines.

Considering that a solar receiver is usually mounted on a tracking system and the receiver is inclined due to a zenith angle, a thermosyphon heat pipe can be utilized. This eliminates the capillary structure for the liquid return, as long as the condenser section of the heat pipe is located in a higher position in the gravity field. Therefore, a loop-type thermosyphon was selected for the purpose of this study. However, a capillary structure was employed in the liquid line, not to secure a liquid return but to avoid a formation of a solid plug in the liquid line and to ensure the distribution of a solidified layer around the inner wall after the heat pipe operation ceased. This was done to enhance the startup capability of the loop-type thermosyphon. Otherwise, it would have taken a considerable amount of time to melt the entire solid plug in the liquid line before startup was achieved. The results of this study may find extended applications in projects with similar requirements.
2. Fabrication of the experimental heat pipe

The loop-type heat pipe was designed and fabricated to meet the interface requirements for the heat source and heat sink, as depicted in Fig.1. The evaporator of the heat pipe contacted the end wall of a cavity-type solar receiver, and the condenser section contacted the heat-supply surface of the AMTEC, which consisted of a flat disk surface and a hollow cylindrical surface in the center. The container wall and transport lines of the loop-type heat pipe were made of stainless steel 304, and the working fluid was sodium. The evaporator and condenser were disk-type containers with diameters of 122 and 216 mm, respectively, though both had a height of 20 mm. The diameters of the vapor and liquid lines were 12.7 and 9.53 mm, respectively, and both lines had a length of 0.5 m. The total length of the loop was approximately 1.4 m. A pillar structure with a diameter of 43 mm and height of 114 mm was installed at the center region of the condenser to interface with the central heat-supply region of the AMTEC in this study. Two layers of stainless steel screen mesh (with a mesh number of 60) were attached to the inner walls of the liquid line, the condenser, and the bottom surface of the evaporator, in order to distribute the condensate (liquid) over the inner surface. This was to avoid the formation of a solid sodium plug that might retard the startup time or block the circulation of the liquid and thus cause local overheating in the heat pipe. The detailed dimensions of the heat pipe sections are summarized in Table 1.

The working fluid charge ratio was varied between 112% and 129% (34.2 g and 39.6 g of sodium in mass) based on the void volume of the screen attached to the inner wall. This was done to identify the optimal performance of the heat pipe. These values are referred to in the remainder of this paper. The fluid charge values in the above corresponded to 30.4% and 35.2%, respectively, based on the evaporator inner volume. It was desired that the upper surface of the condenser as well as the whole outer surface of the pillar should have temperatures as close as possible during heat discharge.

![Fig. 1. A schematic of the loop-type heat pipe and its thermal contacts considered in this study](image)

<table>
<thead>
<tr>
<th>Section</th>
<th>Outer diameter (mm)</th>
<th>Wall thickness (mm)</th>
<th>Volume (cm³)</th>
<th>Length (L) or height(H) (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporator</td>
<td>122</td>
<td>3</td>
<td>148</td>
<td>(H) 20</td>
</tr>
<tr>
<td>Vapor line</td>
<td>12.7</td>
<td>1</td>
<td>66.3</td>
<td>(L) 738</td>
</tr>
<tr>
<td>Liquid line</td>
<td>9.53</td>
<td>1</td>
<td>21.4</td>
<td>(L) 582</td>
</tr>
<tr>
<td>Condenser disk</td>
<td>216</td>
<td>3</td>
<td>510.2</td>
<td>(H) 20</td>
</tr>
<tr>
<td>Condenser pillar</td>
<td>42.7</td>
<td>2.8</td>
<td>119.6</td>
<td>(H) 130</td>
</tr>
</tbody>
</table>
One of the unique features in this study was that the liquid line was located close to the vapor line to enhance the thermal exchange. Thermal insulation was applied after wrapping the two lines together. For a liquid-metal loop-type heat pipe in which the liquid and vapor lines are separated with a considerable distance, startup would take significantly longer since the working fluid would have been in a frozen state at the ambient temperature. Preheating with independent heating is usually required to melt the frozen working fluid in the condenser or in the liquid line. With a close or in-contact configuration of the liquid and vapor lines, as in this study, the startup time can be reduced with the preheating effect due to the partial amount of heat from the vapor line transferring to the frozen working fluid in the liquid line through pipe walls.

Another unique feature in the design was that a vapor injector was installed at the center of the condenser, right beneath the pillar, to promote the vapor transfer to the inner surface of the pillar. This was achieved by extending the vapor line through the condenser side wall to the center region of the condenser before distributing the vapor to both the pillar region and other regions inside the condenser (See Fig.2-c).

Fig. 2 shows a photograph of the fabricated heat pipe and the thermocouple locations used to measure the temperature. K-type thermocouples with silica yarn insulation were used at 13 points at the wall along the working fluid circulation path.

![Fig. 2](image-url)

Fig. 2. The loop-type heat pipe in this study: (a) photograph; (b) locations of temperature measurement; (c) vapor injector in the condenser

<table>
<thead>
<tr>
<th>Thermocouple number</th>
<th>Location</th>
<th>Thermocouple number</th>
<th>Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>T1</td>
<td>Evaporator top surface near the fill-tube</td>
<td>T8</td>
<td>Condenser, disk top surface -1</td>
</tr>
<tr>
<td>T2</td>
<td>Evaporator top surface near the vapor-line</td>
<td>T9</td>
<td>Condenser, disk top surface -2</td>
</tr>
<tr>
<td>T3</td>
<td>Vapor-line, lower point</td>
<td>T10</td>
<td>Condenser, disk bottom surface, near purge tube</td>
</tr>
<tr>
<td>T4</td>
<td>Vapor-line, upper point</td>
<td>T11</td>
<td>Condenser disk bottom surface, near liquid-line</td>
</tr>
<tr>
<td>T5</td>
<td>Condenser, pillar side surface, lower point</td>
<td>T12</td>
<td>Liquid-line, upper point</td>
</tr>
<tr>
<td>T6</td>
<td>Condenser, pillar side surface, upper point</td>
<td>T13</td>
<td>Liquid-line, lower point</td>
</tr>
<tr>
<td>T7</td>
<td>Condenser, pillar top surface</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
3. Experimental method

The experiment was conducted in a laboratory environment with an electric heater attached to the evaporator bottom surface to simulate the heat supply from the end wall of a solar receiver. The maximum power and allowable temperature of the disk type heater were 1.2 kW and 1,000°C, respectively, and these values served as the constraints of the experimental conditions. The input thermal load was limited due to the maximum power and allowable temperature of the electric heater, and not by any operation limits of the heat pipe, e.g. capillary limit. Higher heat fluxes would be applicable during an on-site experiment with an actual solar concentrator and receiver. For the laboratory experiments, both an insulated boundary condition and a natural convection boundary condition were imposed for each fluid fill charge value. The input heat transfer rate to the evaporator varied between 500 and 800 W for the insulated condenser surface, and it varied between 800 and 1,100 W for the natural convection cooling condition at the condenser surface. Ceramic fiber blankets and tapes were used as insulation material. The inclination angle was varied between 30 to 90 degrees from horizontal, but most of the data were acquired at a 70-degree inclination with the liquid line on the downside. The experimental setup is shown schematically in Fig. 3.

The fluid charge amount was considered as the most important factor for heat pipe performance. For each fluid charge amount, the temperature variation was monitored during the transient operation, until a steady state was reached, as a function of the thermal load. The performance of the heat pipe was evaluated by the maximum wall temperature difference as well as the effective thermal conductance $k_{\text{eff}}$ and thermal resistance $R_{\text{th}}$ defined by the following equations.

\[
R_{\text{th}} = \frac{\bar{T}_{\text{exp}} - \bar{T}_{\text{cond}}}{q}
\]

\[
k_{\text{eff}} = \frac{q \cdot L_{\text{eff}}}{A_c \left( \bar{T}_{\text{exp}} - \bar{T}_{\text{cond}} \right)}
\]

where $\bar{T}_{\text{exp}}$ is the average temperature of the evaporator surface (measured at T1 and T2), $\bar{T}_{\text{cond}}$ is the average temperature of the condenser exposed surface (measured at T5-T9), $A_c$ is the cross-sectional area of the vapor line, and $q$ is the heat transfer rate through the heat pipe. The $q$ values were evaluated differently depending on the boundary conditions imposed on the condenser surface. For the uncovered condenser surface, $q$ was estimated as the sum of the radiation and natural convection heat loss rates based on the measured temperatures on the surface.

![Fig.3. Schematic of the experimental setup](image-url)
4. Results and discussion

4.1. Performance characteristics with natural convection cooling in the condenser

A natural convection cooling condition was imposed on the condenser to investigate the general characteristics of a loop-type heat pipe, although this would not be a practical condition for an AMTEC application. For the fill charge ratios ($\Phi$) between 112 and 129%, the time to reach each steady state was between 1.5 and 2 hours. The criterion for a steady state in this study was less than 1°C variation in every temperature measurement for 5 minutes.

Typical results for $\Phi = 112\%$ are shown in Fig. 4-a. The difference between the average temperatures in the evaporator and condenser was about 200°C for an 800-W input heat rate, but it decreased to 117°C for a 1,100-W input heat rate. The temperatures at T9 (condenser disk top, side) and T10 (condenser bottom, near purge line) were the lowest in every input heat rate. However, T8, which was on the opposite side of T9 on the condenser disk top surface, usually showed higher values than T9. For a 1,100-W thermal load, the maximum temperature difference on the condenser top surface was 80°C. Temperatures at T11 (condenser bottom, near liquid return line and vapor line) always showed the highest value in the condenser due to conduction from the vapor line nearby. Temperatures in the liquid return line, T12 and T13, also exhibited higher values than the average condenser temperature due to a preheating effect from the vapor line in the vicinity and also due to conduction from the evaporator through the pipe wall. These two temperatures rose to even higher values than T11 for 1,000-W and higher input heat rates. As T7 (pillar top surface) showed little difference from the average condenser temperature for input heat rates of 1,000 W and higher, it was assumed that the vapor injector functioned as expected.

A similar graph for $\Phi = 122\%$ is shown in Fig. 4-b. The difference between the average temperatures in the evaporator and condenser was 180°C for an 800-W input heat rate, but it decreased to 91°C for a 1,100-W input heat rate. In comparison with the $\Phi = 112\%$ case, these temperature differences decreased by about 20°C. For a 1,100-W thermal load, the maximum temperature difference on the condenser top surface was 50°C, which was 30°C less than that for $\Phi = 112\%$. In addition, temperatures at T12 and T13 exhibited lower values than that at T11, which represented a normal liquid circulation throughout the liquid line. For $\Phi = 129\%$, the difference between the average temperatures in the evaporator and condenser was 224°C for an 800-W input heat rate, but it decreased to 104°C for a 1,100-W input heat rate. For a 1,100-W thermal load, the maximum temperature difference on the condenser top surface was 90°C, which was 10°C larger than that for $\Phi = 112\%$.

Steady-state temperature distributions for $\Phi = 117\%$ are summarized in Fig. 5 for varying input heat rates. The difference between the average temperatures in the evaporator and condenser was 170°C for an 800-W input heat rate, but it decreased to 88°C for a 1,100-W input heat rate. It can be concluded that the performance of the heat pipe was enhanced since the temperature difference between the evaporator and the condenser decreased by approximately 30°C from that for $\Phi = 112\%$.

Fig. 4. Steady-state temperature distributions in the loop-type heat pipe as a function of input thermal load: (a) $\Phi = 112\%$; (b) $\Phi = 122\%$. 
In comparison with $\Phi = 122\%$, $T_{10}$ typically increased by 60 to 70°C for every input heat rate. For a 1,100-W thermal load, the maximum temperature difference on the condenser top surface was 20°C, which was 60°C less than that for $\Phi = 112\%$. The effective thermal conductance and the thermal resistance were 47,000 W/m-K and 0.13°C/W, respectively, based on the estimated heat dissipation through the condenser surface. The fluid charge of 117% was considered to be optimum based on the experimental results for four different fluid charge values.

The input heat rate to the evaporator, $q_{\text{in}}$, was irrelevant to the estimation of thermal performance from Equations (1) and (2). Since there would have been considerable heat loss through the insulation due to a large temperature difference across the layer, the heat discharge rate through the condenser surface, $q_{\text{cond}}$, served as an effective heat transport, and its value should be reflected in the equations. The following equation was used to estimate $q_{\text{cond}}$ [8].

$$q_{\text{cond}} = q_{\text{conv}} + q_{\text{rad}} = \bar{h}A_s(T_s - T_{\text{amb}}) + \varepsilon A_s\sigma(T_s^4 - T_{\text{amb}}^4)$$

(3)

The heat transfer coefficient for natural convection was evaluated by $\bar{h} = (k / L)\bar{N}_U$, where $\bar{N}_U$ was determined combining the average Nusselt numbers for an inclined flat surface proposed by Fujii and Imura [9], which is valid in the range of $10^4 < Gr \cdot Pr \cdot \cos \theta < 10^5$, and that for an inclined cylindrical surface proposed by Al-Arabi and Salman [10], which is valid in the range of $Gr \cdot Pr < 2 \times 10^4$, as shown in Equations (4) and (5), respectively.

$$\bar{N}_U = 0.56(Gr \cdot Pr \cdot \cos \theta)^{1/4}$$

(4)

$$\bar{N}_U = \left[0.60 - 0.448\left(\sin \theta \right)^{1.03}\right] \cdot (Gr \cdot Pr)^{1/12} \cdot \left(\sin \theta \right)^{75}$$

(5)

The effective thermal conductance and the thermal resistance as a function of the heat discharge rate through the condenser and the fluid charge ratio are summarized in Fig. 6, and the typical numerical values are listed in Table 3.

Table 3. Summary of steady-state performance of the loop-type heat pipe with natural convection cooling in the condenser (*the heat flux values were calculated based on the heated surface in the evaporator)

<table>
<thead>
<tr>
<th>$\Phi$ (%)</th>
<th>$q_{\text{cond}}$ (W)</th>
<th>heat flux (kW/m²)*</th>
<th>$k_{\text{eff}}$ (W/m-K)</th>
<th>$R_{\text{th}}$ (°C/W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>112</td>
<td>571</td>
<td>48.8</td>
<td>30,000</td>
<td>0.20</td>
</tr>
<tr>
<td>117</td>
<td>671</td>
<td>57.4</td>
<td>47,000</td>
<td>0.13</td>
</tr>
<tr>
<td>122</td>
<td>645</td>
<td>55.2</td>
<td>43,000</td>
<td>0.14</td>
</tr>
<tr>
<td>129</td>
<td>611</td>
<td>52.3</td>
<td>36,000</td>
<td>0.17</td>
</tr>
</tbody>
</table>
4.2. Performance characteristics with an insulated condenser surface

The temperatures of the heat pipe would have increased continuously if all of the boundaries aside from the evaporator had been insulated perfectly. However, in all experiment sets, a converged, steady temperature distribution was established in the heat pipe after a certain elapsed time, corresponding to a specific input thermal load. This means that there was a heat loss rate across the insulation material due to both convection and radiation, which balanced the input heat rate. However, it was uncertain how much heat discharge rate actually occurred through the condenser and the other parts of the heat pipe. To estimate the effective thermal conductance and the thermal resistance in this case, therefore, the input heat rate $q_{in}$ was used for each $q$ in Equations (1) and (2).

For this experiment set, $q_{in}$ ranged from 500 W to 800 W. The maximum $q_{in}$ was limited by the electric heater surface temperature, which was about 200°C higher than the T1 or T2 temperatures. The experimental results are summarized in Fig.7 for four different values of $\Phi$. For all $\Phi$ values, $q_{in}$ of 500-W turned out to be insufficient for the full operation of the heat pipe, and it revealed that the difference in the average temperatures in the evaporator and the condenser were between 23 and 41°C. Among the four fluid charge ratios, the $\Phi = 117\%$ case exhibited the smallest temperature difference between the evaporator and the condenser for any input thermal loads. For $\Phi = 117$, 122, and 129\% (see Figs. 7-b, 7-c, and 7-d), T13 showed the lowest value with a significant decrease, by 90°C to 120°C, from T12 for any input thermal loads, which might be interpreted as a normal circulation of working fluid through the liquid line. For $\Phi = 112\%$ (see Fig. 7-a), however, T12 and T13 showed adverse temperature profiles for three input heat rates, possibly due to an insufficient liquid return. In consequence, the smallest overall temperature difference along the loop was achieved in the $\Phi = 112\%$ case for 600-W or higher input heat rates.

The effective thermal conductance and the thermal resistance were summarized against the input thermal load and the fluid charge ratio in Fig. 8. For an 800-W input thermal load, the effective thermal conductance exhibited extremely high values from 902 to 973 kW/m-K depending on the fluid charge ratio, and an extremely low thermal resistance value of 0.006°C/W regardless of the fluid charge value.

4.3. Frozen start-up characteristics

Frozen startup characteristics were investigated to identify the time required to initiate a normal operation as well as the time to reach a steady-state condition for each fill charge ratio. A normal operation implies that the working fluid completed a circulation along the loop. Fig. 9 summarizes the temperature variations in seven locations as the time elapsed after a 650-W thermal load was applied for $\Phi = 117\%$ with the insulated condenser surface. As the temperatures on the evaporator top surface led the temperature increase, the vapor line temperatures (T3 and T4) started to increase rapidly after 17 and 24 min. Then the liquid return line temperatures T11 and T12 increased abruptly with a little fluctuation at 26 min after the startup, until a normal operation began and all temperatures
exhibited smooth and stable increases in 31 min. The elapsed time required for a steady state was 120 min for $\Phi = 117\%$, and it was about 140 min for the other fill charges.

Similar experiments were conducted for the case with natural convection cooling in the condenser, with a 950-W input thermal load. For $\Phi = 117\%$, the elapsed time to reach a steady state was 90 min, and it was 100 to 110 min for other fill charges. The elapsed time to initiate a normal operation increased with the fill charge ratio, and it was 20 min for $\Phi = 112\%$, but 25 min for $\Phi = 129\%$. It was presumed that the startup time could be reduced if the input thermal load was raised in the field operation, and if a better thermal contact was provided between the vapor and liquid lines.

Fig. 7. Steady-state temperature distribution in the heat pipe as a function of input thermal load: (a) $\Phi = 112\%$; (b) $\Phi = 117\%$; (c) $\Phi = 122\%$; (d) $\Phi = 129\%$

Fig. 8. Loop-type heat pipe performance against input thermal load and fluid charge ratio (insulated condenser surface): (a) $k_{ef}$; (b) $R_{th}$
5. Conclusions

Based on the experimental data, the loop-type heat pipe in this study is capable of transporting an 800-W thermal load with a heat discharge temperature in the condenser higher than 730°C.

The working fluid charge ratio affected considerably the heat pipe performance, in terms of the effective thermal conductance, thermal resistance, and isothermal characteristics along the loop. With the specified geometry, constraints, and working conditions presented in this study, the optimum fluid charge ratio was 117% based on the wick void volume, which corresponded to 32% fill charge based on the evaporator inner volume. The same fill charge ratio turned out to take the least amount of time to achieve a steady state from a frozen startup.

For natural convection cooling in the condenser, the combined heat discharge from the condenser surface (the effective thermal load) was estimated by the empirical correlations based on the measured temperatures. When the effective thermal conductance was correlated with the effective thermal load, the performance curves for the four different fluid charges exhibited only little difference.

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