Flat Heat Pipe Design, Construction, and Analysis

G. Voegler

July 1999

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States, nor the United States Department of Energy, nor any of their employees, nor any of their contractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

KNOLLS ATOMIC POWER LABORATORY SCHENECTADY, NEW YORK 12301

Operated for the U.S. Department of Energy by KAPL, Inc. a Lockheed Martin Company
DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, make any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.
DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.
ABSTRACT

Heat pipe technology can effectively meet the demand for an isothermal emitter in a thermophotovoltaic (TPV) energy conversion system as it utilizes near isobaric phase changes to transfer heat at a uniform temperature. A flat heat pipe offers many advantages over the conventional cylindrical design, such as increased surface area to volume ratio and the ability to stack or layer the energy conversion system on both sides of the flat heat pipe to utilize available energy. Flat heat pipes present unique engineering demands and are also difficult to construct. This paper details the design, construction and partial analysis of a low temperature flat heat pipe in order to determine the feasibility of implementing flat heat pipes into TPV energy conversion systems.

INTRODUCTION

Figure 1 depicts a conceptual TPV energy conversion system utilizing heat pipes. Combustion gases from a heat source such as a gas turbine combustor flow through channels on which heat pipes are mounted. These heat pipes can serve as emitter surfaces. Across from the hot side heat pipes, TPV cells can be mounted to cold side heat pipes which are heat pipes in contact with the thermal sink. An isothermal emitting surface is called for in TPV energy conversion systems because the voltage outputs of the TPV cells are very sensitive to the wavelength bandwidth of the emitting surface. The emitter's wavelength bandwidth is a function of temperature. On the cold side, the TPV cells could also utilize a flat heat pipe, but this is less critical. Flat heat pipes are not new to the industry, several companies have designed them for space or computer applications (Tanzer, 1983, Tanzer, et al., 1985, Rankin, 1984, Tanzer, et al., 1986).

The flat heat pipe of this study was designed to meet the following criteria:

1. Construction to be made of Monel 400.
2. Exhibit two-dimensional heat transfer behavior
3. Feature dimensions of 1.22 m long (4 ft) by 30.48 cm wide (1 ft) by 1.27 cm thick (0.5 inch).
4. Maintain an operating temperature of up to 100°C (212°F).
5. Possess a means of internal support for structural rigidity without sacrificing two-dimensional flow capabilities.

The following experimental and theoretical goals were set:

1. Monitor the condenser region temperature profile via thermocouples and infrared videography.
2. Test performance at different heat inputs, fluid charges, and varying inclination angles.
3. Modify heat pipe equations for flat geometry.
4. Analyze experimental data to determine overall performance and feasibility of flat heat pipes for a TPV energy conversion system.

METHODOLOGY

This investigation proceeded with the following methodology. First, modifications to the theory of cylindrical heat pipes were extended to flat heat pipes. These included
modifying the design limits such as the boiling and capillary limits (Faghri, 1995), and designing for new structural limits.

Next, a subscale model of the final design was constructed to resolve fabrication difficulties before attempting the final full scale design. Instrument calibration and basic heat pipe operational procedures, such as fluid charging, were perfected with this model. The subscale model also served to provide insight that was incorporated into revisions for the full scale model. The full scale heat pipe was then constructed, tested, and analysed under the test conditions.

Finally, the raw data from the experiments were collected and reduced. The infrared camera data of the condenser end surface temperatures was used to aid in explaining the purely quantitative thermocouple data as well as to provide visual understanding of heat pipe phenomena. The temperatures of regions at the same distance axially from the heater end were compared for various tests and analysed for thermal performance assessment at varying heat inputs, fluid charge, and angles of inclination.

**FLAT HEAT PIPE PROPERTIES**

Flat heat pipes are similar to cylindrical heat pipes. The only real difference between the two is geometrical. While this may seem a minor difference, it presents many challenges from an engineering standpoint.

Typically, heat pipes are used to transfer quantities of heat across a distance with only a slight temperature loss from end to end. The cylindrical design works well to serve this purpose. However, when designing an emitter for a TPV energy conversion system, it is advantageous to have a large surface area to volume ratio in order to maximize the power density of the system. A flat heat pipe was conceived for this purpose. Due to the different surface area geometry, flat heat pipes also have different flow and structural design considerations than those of cylindrical heat pipes.

Flow properties in cylinders are different from those in rectangular geometries such as flat plates and/or boxes. The flow of a thin film over a flat sheet (such as in the liquid return loop of a flat heat pipe) is not the same as the flow of a cylindrical circumferential film. Also, vapor flow through a cylindrical space differs from vapor flow through a rectangular cross section. The flow property differences can alter the steady-state limitations in flat heat pipe design.

The limit most affected in the design of a moderate temperature (100 C) flat heat pipe is the capillary limit. This is significant because the capillary limit is typically the highest heating limit at which a heat pipe is designed to operate under steady-state conditions. The capillary limit involves the flow of the liquid in the wick’s return path and the pressure loss of the vapor and liquid through their respective areas of the heat pipe. Also, since the U.S. Naval Academy’s (USNA) flat heat pipes do not have adiabatic sections, a more accurate approximation of the mean liquid return length in the wick is needed. The capillary limit for a flat heat pipe when only one side of the evaporator region is heated and the liquid returns only on that one side can be derived as Equation 1. In Equation 1, \( q_e \) is the maximum heat flux into the evaporator section. Equation 2 defines \( K \), the permeability for a wire mesh wick. To achieve a large capillary limit, \( r_{eff} \) the effective pore size of the wire mesh in the wick, must be minimized in order to achieve a large capillary pumping head. Decreasing \( r_{eff} \) also decreases permeability, which can lower the capillary limit by restricting the flow of the return liquid to the evaporator portion of the wick. Proper flat heat pipe design involves optimization of both of these factors to obtain a high capillary pumping head, while still maintaining an acceptable permeability.

\[
q_e = \frac{2 \sigma k_f}{r_{eff} L} \left( \frac{12 \mu_v}{\rho_v \delta^2} + \frac{\mu_i}{2 \rho_l K} \right)^{-1} \tag{1}
\]

\[
K = \frac{2 \mu \varphi A_i L}{\rho_l V_p f A_S} \tag{2}
\]

The permeability of the wire mesh wick calculated by Equation 2 can be approximated by Equation 3. Sample calculations were performed using both equations and it was found that both equations yielded nearly equal values.

\[
K = \frac{\varphi D_h^2}{32} \tag{3}
\]

Two heating configurations were considered for the full scale flat heat pipe. The first involved the entire heater section mounted on one side of the evaporator region. For this configuration, the capillary limit is found by Equation 1. The other arrangement of the evaporator section consisted of heaters mounted to both sides of the evaporator region. For this heater configuration, the capillary limit is found from Equation 4.

\[
q_e = \frac{2 \sigma k_f}{r_{eff} L} \left( \frac{12 \mu_v}{\rho_v \delta^2} + \frac{\mu_i}{2 \rho_l K} \right)^{-1} \tag{4}
\]

This capillary limit (two heated sides) is theoretically higher than the limit with only one side heated. This is because the two sides of the heat pipe work as parallel fluid circuits with the total flow resistance being less when both sides are heated and returning working fluid versus only one side heated at the same flux and returning working fluid on that side.
The boiling limit in a flat heat pipe is similar to that of a cylindrical heat pipe. Both involve pressure balances at the onset of nucleation (pressure difference from a bubble situated at a nucleation site to the vapor space through the liquid in the wick). The boiling limit equation for flat heat pipes is given by Equation 5.

\[
q_b = \frac{2A_{\text{hin}}^* R T_{\text{sat}}^2 \rho^k}{P_{\text{sat}} h f g t_w} \left( \frac{1}{r_b} - \frac{1}{r_w} \right)
\]  

(5)

The boiling limit can be expected to be a very large value. Flat heat pipes are constructed of high strength metals with polished surfaces. These metals feature very small surface defect sizes and, therefore, minute nucleation site sizes for the onset of boiling. This drives the maximum heat input for the boiling limit up to a large value. This limit is typically much higher than the capillary limit.

With water as the working fluid, the entrainment and viscous limits are not an operational limit consideration. The physical properties of water drive the entrainment and viscous limits to very large values within this project’s operating parameters (20°C to 100°C condenser temperature). Furthermore, the entrainment limit is usually not a factor in well-wicked heat pipes, since the wick retains the liquid to prevent entrainment by the vapor. Being difficult to calculate, the entrainment limit is found by comparing the shear force on the liquid due to the vapor flow with the surface tension of the liquid. For most applications, the entrainment limit is calculated by assuming that the Weber number equals one, where the Weber number is a ratio of vapor inertial forces to liquid surface tension forces (Peterson, 1994).

In the sonic limit, the vapor cannot travel faster than the local speed of sound in a constant area duct. The heat input that can be delivered is therefore limited by the maximum mass flow rate of the vapor. The sonic limit is found by Equation 6.

\[
q_s = \rho_v A_v h f g \sqrt{k R T_{\text{local}}}
\]  

(6)

FLAT HEAT PIPE STRUCTURAL CONSIDERATIONS

A flat heat pipe has structural problems that are not associated with cylindrical heat pipes. Cylindrical heat pipes are natural pressure vessels due to their shape, and can withstand larger pressure differences as well as the resulting compressive or tensile forces on their walls than flat heat pipes. Since water heat pipes can exist at pressures lower than atmospheric pressure when not in use, and can exceed atmospheric pressure at elevated, in-use temperatures, the cylindrical design has clear advantages. A heat pipe’s surface area is subjected to the pressure difference created between the heat pipe internal pressure and the external environmental pressure. This may result in a large force acting on the vessel, which can cause material failure. This was clearly a design consideration that needed to be solved.

The design operating point for the flat heat pipes in this design was one atmosphere of differential pressure, with the design point for stress loading taken at a near perfect vacuum. In this scenario, the 101.3 kPa (14.7 psia) of atmospheric pressure would be acting to deform the heat pipe with only the internal structural supports to counteract the pressure differential. Since one of the project goals was to design structural supports that did not inhibit two-dimensional flow, any internal beams or walls were ruled out as internal supports. This left only pin supports as an option. Not knowing how the pins were to be fastened to the vessel wall in the final design, the pins were modeled as both welded (rigidly attached) and not welded to the vessel wall. The vessel wall was then modeled as a beam, with either the side vessel wall and a pin or two pins acting as supports. This gave the maximum expected deflection for the flat heat pipe of a given thickness, material, and spacer distance. A spreadsheet was developed to analyze deflections for many different materials, material thicknesses, and pin spacings.

Beam deflection was not the only structural analysis conducted. To determine whether or not deflection was the limiting factor, a column buckling analysis for pins of varying lengths, diameters, materials, and types of attachments also had to be conducted.

SUBSCALE FLAT HEAT PIPE

A subscale heat pipe 0.3 m (1 ft) by 0.1 m (4 inches) by 1.27 cm (0.5 inch) was constructed first. Its purpose was to provide practical experience of flat heat pipe construction as well as to be used to help calibrate instruments and set up the laboratory spaces. In addition, it aided in proving the practicality of the design and revealing any structural faults.

Leak-testing and charging consisted in the fabrication of a charging apparatus. A 0.635 cm (0.25 inch) nipple was fitted to the end of the heat pipe and mated to the compressor. The heat pipe was then pressurized for leak testing. A valve in the charging apparatus could be throttled so that excessive pressure did not build up in the heat pipe. To leak test, soapy water was applied to the pressurized heat pipe in order to detect the leaks. Leaks would form bubbles in the soapy water.

Charging the heat pipe with working fluid was performed fairly simply as well. First, the heat pipe and charging assembly were mated to the vacuum pump and evacuated. In the fluid charging column was placed the correct amount of water to charge the heat pipe. These amounts were measured and marked, taking into account the volume that would occupy the fittings as well as the heat pipe. Once evacuation was complete, the valve attached to the vacuum pump and the valve attached to the heat pipe were closed, and then the vacuum pump was shut off. The valve attached to the fluid column was then opened, and the vacuum inside the fittings drew the water in to fully fill the pipe volume between the fittings. The heat pipe valve was then opened slightly to bleed in the necessary charge (as marked on the
column). All valves were then closed, and the charging apparatus removed, if necessary. All valves used were rated for special high-vacuum service.

After all leaks were eliminated, the heat pipe was painted flat black to raise its surface emissivity to approximately 0.95 so that it could be viewed best by an infrared camera. The subscale heat pipe was next successfully charged and subjected to a hot water bath. Within seconds, the opposite end of the heat pipe began to warm up. It worked as a heat pipe. The heat pipe was then monitored with the infrared camera. This facilitated a qualitative check of the heat pipe performance as well as the calibration of the infrared camera.

FULL SCALE HEAT PIPE CONSTRUCTION DETAILS

After the screens and Monel sheets were cut to the proper dimensions, their size and assigned hole placement were loaded into the Sheets Manufacturing Company's CNC Punch-Press machine's user interface. The Monel sheets for the vessel wall were punched first, making holes in the locations where pins were to be welded. Next, to avoid any tearing, stretching, or other kind of deformation in the screen; the screens were laid on top of each other, taped to the side of thin steel sheets (30mil), and sandwiched in between the sheets. They were then loaded onto the CNC Punch-Press machine and punched in the appropriate places so that the steel sheets could absorb the initial impact and prevent the movement and subsequent deformation of the screen. By using the computerized machine, it was possible to place all of the necessary holes in exactly the correct spots on both the sheets and the screen to ensure that the sides would line up perfectly.

Long copper bars with fine radius edges were utilized to facilitate bending the fullscale heat pipe to the required dimensions. The two layers of screen were then placed on top of the vessel, fastened with tape on the ends, and fastened down to the vessel by threading small screws through the pin spacer holes and attaching nuts on either sides. The screen was then tack-welded to the vessel wall in regular intervals between the pin spacer locations. Keeping the screen flush and in contact with the vessel presented a challenge. A small portion of the screen on one side of the heat pipe detached from the vessel in a small section of the heat pipe. This spot was noted in case it caused any performance problems. The end pieces and fitting were machined. The pins were then cut to the proper length, milled, and deburred to fit into the necessary space.

With all of the necessary components ready, the assembly began. First, the sides of the two separate halves were welded together, carefully sequencing the welds and using a heat trap to minimize deformation. After this, the ends were welded on and the singular end fitting was welded snug to the end to resemble the final appearance of the subscale heat pipe. Following this, the pins were placed in their proper spots and TIG welded from either side of the heat pipe.

EXPERIMENTAL SETUP AND PROCEDURE

The setup for experimentation as well as the procedure for experimentation had great importance to the success of the project. It was important to verify that the flat heat pipe does indeed perform as a heat pipe and to take data on its one-dimensional performance before proceeding on to any other tests. These one-dimensional tests were also the easiest way to understand and interpret the internal dynamics of the heat pipe as they influenced the temperature profile. A heat pipe is, by definition, a hermetically sealed vessel that transfers heat via phase change processes and has capillary pumping action to circulate the working fluid. It was necessary to test this capillary action. Finding the capillary limit of the heat pipe is therefore very important to not only understanding heat pipe performance but also to maximizing heat pipe performance. Experiments were conducted at differing angles of gravity assist for the liquid return path to test the capillary pumping action of the screen and gravity's effect on the heat pipe's performance.

EXPERIMENTAL SETUP

Sixteen high temperature 500 Watt strip heaters 1.905 cm (0.75 inch) wide with a heated section of 30 cm (11.81 inches) were utilized as the heat sources. They were wired in blocks of four to variable transformers. Their heating rates could also be adjusted independently in four blocks of four heaters each. The variable transformers were wired to Watt transducers which were wired into junction boxes. The electric signals from the Watt transducers were recorded by the computer to which these junction boxes were connected.

Thermal joint compound was spread between the heaters and the heat pipe to lower contact resistance and to ensure that most of the generated heat went into the heat pipe. The heat pipe was then insulated with a ceramic tile over the heater sections with aluminum foil air bubble insulation wrapped around the entire assembly. The same evacuation and charging assembly that was developed for the subscale heat pipe was used for the full scale heat pipe.

Experimentation at various angles to the horizontal was an important part of the test criteria. It was therefore necessary to design a test stand that would allow the heat pipe to be rotated through various angles while remaining free to natural convection. For this, a wooden stand was designed that consisted of two legs fixed to a flat base. Attached to the stand were two clamps to hold the heat pipe in place while being able to rotate it. These were made from two C clamps welded together by a steel bar. From this bar protruded a threaded screw, which was fixed to the bar. This screw went through a wooden bearing block and the test stand legs, and was attached on the other side of the leg by a wing nut and a washer. The wing nut could be loosened and tightened to move the clamps through the various angles about the screw. This assembly rotated the heat pipe through the required test angles.

Temperature data were recorded in two ways. The first was via the infrared camera videography. The other means was by an array of thermocouples. The thermocouples were wired
into the same data acquisition junction boxes used by the Watt transducers. There were 8 junction boxes with 8 channels each for a total of 64 channels of data sampling. However, only 43 were used (39 thermocouples plus four Wattmeters). Figure 2 depicts the experimental setup.

**EXPERIMENTAL PROCEDURE**

The heat pipe was tested at different orientations. Figure 3 shows the nomenclature for the different orientations. The working fluid charge is also defined as that amount of fluid to fully saturate the wick. Thus, 125% charge refers to a 25% over saturated wick. The testing criteria (all heaters mounted on one side) were as follows:

1. With 100% fluid charge, test orientations 2, 3, 4, 5, 6 and 7 heating slowly to at least 400 Watts (more if conditions allow), allowing the heat pipe to reach equilibrium temperature for each heat input.
2. With 125% fluid charge, test orientations 3, 4, 5 and 7 heating with at least 400W allowing equilibrium conditions.
3. With 75% charge, test orientations 3, 4 and 5 heating with at least 400W allowing equilibrium conditions.
4. For all runs, data is to be collected over time by data acquisition software, which records continuously, and the infrared camera, which records at the user's discretion.

**ANALYSIS**

The flat heat pipe presented in this investigation was essentially developed to be an isothermal fin and was to be utilized as the cold side of a TPV energy conversion system. A fin, whether it is a heat pipe fin or a conventional fin, rejects heat to its environment mostly by combined convection and thermal radiation. Thus at steady state conditions,

\[
q_{in} = hA_s(T_s - T_{amb}) + \sigma e A_s(T_s^4 - T_{amb}^4) \tag{7}
\]

for an isothermal heat pipe fin. Figure 4 presents the individual convective and radiative heat transfer rejection terms of Equation 7 for various heat inputs. The rejection temperature is room ambient, taken as 25°C. As one can see, natural convection and thermal radiation are of comparable magnitudes for the range of heat input range of this investigation.

The natural convection coefficient, \(h\), is a function of heat pipe orientation. The following correlations (Kreith and Brohm, 1997), were utilized in this analysis to determine the Nusselt number, \(Nu\):

\[
Nu = 0.56(Gr_f Pr \cos \theta)^{1/4} \tag{8}
\]

for \(\theta\) less than 90°, and

\[
Nu = 0.15(Ra_f)^{1/3} \tag{9}
\]

for \(\theta\) equal to 90°, where \(\theta\) equals the angle taken from the horizontal.

Figure 5 shows the analytical solution of Equation 7 with Equations 8 and 9 for the case of 400W input and varying heat pipe orientations between the horizontal and the vertical (evaporator region on the bottom). Also depicted are experimental values taken at positions 3, 4 and 5. The experimental values for heat pipe temperature are an average of the thermocouple data taken at 400W for those cases. The heat pipe working fluid charge was 100% (a fully saturated wick).
can be seen, the experimental data reasonably fits the analytical data. For positions where the heat pipe evaporator region was above the condenser, the heat pipe could not maintain its capillary pumping force.

To compare the effects of varying charges, each charge experiment was conducted at a 400W heat input. The experiments were allowed to reach a steady-state condition for orientations 3, 4, and 5. The horizontal cases produced the best data and were examined for comparison. Although the differences in effects were not very pronounced, it was apparent that the 75% charge case displayed the quickest response time. This was because there was less water in the heat pipe and therefore less mass to absorb the evaporator input energy. Due to the presence of small leaks (discussed later) it became apparent that gas loading was present. Although the leak was very slow, the heat pipe data did reflect the presence of air, particularly at low heat inputs. At higher heat inputs (greater than 600W) data suggested that the air was pushed into the charging pipe at the end of the heat pipe. Differences between the 100% and 125% charge cases were not discernable. However, it is thought that 125% charge would create some liquid pooling in the evaporator section, disabling the capillary pumping action of the wick structure.

Figure 6 depicts a typical infrared video photograph for a portion of the condenser surface taken at 600W during a horizontal test case. The cooler left hand corner shows the presence of air. It should be pointed out that this area was small compared to the area of the entire heat pipe condenser surface.

CONCLUSIONS

Overall, this project was successful in determining the usefulness of flat heat pipes for thermophotovoltaic energy conversion. The goals met were as follows:

- A flat heat pipe of the set dimensions was designed entirely of Monel 400.
- The flat heat pipe was successfully constructed using pin spacers as internal supports and operated without biasing flow direction.
- Using water as a working fluid, an operating temperature in excess of 100°C was achieved.
- Cylindrical heat pipe equations were modified to flat geometry to design flat heat pipes.
- A flat heat pipe effectively served as an emitter for a thermophotovoltaic energy conversion system.

In addition, it was determined that:

- Increasing the angle of orientation from the horizontal required a higher surface temperature in the condenser region to reject the same amount of heat as the vertical orientation.
- The heat pipe did not operate completely without gas loading due to a small leak. However, even in the presence of gas loading, it was possible to maintain a near isothermal condenser surface.
REFERENCES


NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>area</td>
</tr>
<tr>
<td>$D_h$</td>
<td>hydraulic diameter</td>
</tr>
<tr>
<td>$f$</td>
<td>friction factor</td>
</tr>
<tr>
<td>$Gr_f$</td>
<td>Grashof number</td>
</tr>
<tr>
<td>$h$</td>
<td>heat transfer coefficient</td>
</tr>
<tr>
<td>$h_{lv}$</td>
<td>latent heat of vaporization</td>
</tr>
<tr>
<td>$K$</td>
<td>permeability</td>
</tr>
<tr>
<td>$k$</td>
<td>thermal conductivity</td>
</tr>
<tr>
<td>$k_{en}$</td>
<td>effective thermal conductivity in wick</td>
</tr>
<tr>
<td>$L$</td>
<td>length (over bar indicates mean length)</td>
</tr>
<tr>
<td>$Nu$</td>
<td>Nusselt number, hL/k</td>
</tr>
<tr>
<td>$P$</td>
<td>pressure</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$q$</td>
<td>heat input</td>
</tr>
<tr>
<td>$R$</td>
<td>gas constant</td>
</tr>
<tr>
<td>$Ra$</td>
<td>Rayleigh number</td>
</tr>
<tr>
<td>$r$</td>
<td>radius (i.e., nucleation site in wall or pore)</td>
</tr>
<tr>
<td>$r_{eff}$</td>
<td>effective pore radius in wick</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature</td>
</tr>
<tr>
<td>$t$</td>
<td>thickness</td>
</tr>
<tr>
<td>$V$</td>
<td>velocity in wick pores</td>
</tr>
</tbody>
</table>

Greek

- $\delta$ - thickness (wick or vapor space)
- $\varepsilon$ - emissivity
- $\mu$ - dynamic viscosity
- $\varphi$ - porosity in wick
- $\rho$ - density
- $\sigma$ - surface tension or Stephan Boltzmann constant
- $\theta$ - orientation

Subscripts

- $\text{amb}$ - ambient
- $\text{b}$ - boiling limit
- $\text{c}$ - capillary limit
- $\text{in}$ - heat input to evaporator section (one side in this case)
- $\text{l}$ - liquid
- $\text{local}$ - local property
- $s$ - sonic limit or condenser surface
- $\text{sat}$ - saturation property
- $\text{v}$ - vapor
- $w$ - wick
- $x$ - wick cross sectional area