EXPERIMENTAL INVESTIGATIONS OF A GRAVITY ASSISTED HEAT PIPE AT DIFFERENT VACUUM CONDITIONS AND FILL CHARGE RATIO

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Abstract

Heat pipes are found useful as a cooling means for modern electronic devices, which utilize latent heat of vaporization of working fluid instead of sensible heat. As a result the effective thermal conductivity may be several orders of magnitude higher than that of an ordinary conductor. In the present study an attempt is made to fabricate and test a gravity assisted heat pipe with 5 mm inner diameter, 8 mm outer diameter and 150 mm length with a thermal capacity of 10 Watts. Experiments were conducted with and without working fluid (distilled water) for 25-100% fill charge ratios of the evaporator volume and for different vacuum levels i.e., 1000 mbar, 400 mbar and 150 mbar, with the input thermal load varying from 2 to 10 Watts. The fill ratio of working fluid as a percentage of evaporator volume is found to have significant effect on the performance of heat pipe with respect to the overall heat transfer coefficient and thermal resistance. The fill ratios of working fluid greater than 25% and less than 60% of the evaporator volume shows better results in terms of increased heat transfer coefficient and decreased thermal resistance. At increased vacuum levels inside the heat pipe, the operating temperature decreases and system responds faster to the thermal loads. The data reported in this study shall serve as a good database for the researchers in this field.
Keywords: Heat Pipe, Fluid Inventory, Gravity assisted Heat Pipe, Thermosyphon

1. Introduction

A heat pipe is a simple device with no moving parts in it. It can transfer large quantities of heat over considerable distance without any external power input. They are often referred to as the superconductor of heat as they possess an extraordinary heat transfer capacity and rate with almost no heat loss [1]. The vapour generated in the evaporator section of gravity assisted closed two-phase thermosyphon rises up to the condenser section where it condenses and returns to the evaporator as a falling liquid film [2].

Numerous investigations have been made to obtain the thermal performance of the wickless heat pipe with water as the working fluid in common [3-6].

<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>Subscripts</th>
<th>Notes</th>
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<tr>
<td>$A$  cross-section area (m$^2$)</td>
<td>$d$</td>
<td>diameter (m)</td>
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<tr>
<td>$K$  thermal conductivity</td>
<td>$h$</td>
<td>heat transfer coefficient</td>
</tr>
<tr>
<td>(W/m $^0$C)</td>
<td></td>
<td>(W/m$^2$ $^0$C)</td>
</tr>
<tr>
<td>$L$  length (m)</td>
<td>$l$</td>
<td>width of cooling fin</td>
</tr>
<tr>
<td>$P$  saturation pressure (N/m$^2$)</td>
<td>$m$</td>
<td>fin factor for uniform c/s area in m$^{-1}$</td>
</tr>
<tr>
<td>perimeter (m)</td>
<td>$r$</td>
<td>radius (m)</td>
</tr>
<tr>
<td>$Q$  heat input (W)</td>
<td>$\lambda$</td>
<td>latent heat of vaporization</td>
</tr>
<tr>
<td>$R$  thermal resistance ($^0$C/W)</td>
<td>$\rho$</td>
<td>density (kg/m$^3$)</td>
</tr>
<tr>
<td>$T$  Temperature ($^0$C)</td>
<td>$\eta$</td>
<td>Efficiency</td>
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Yong Joo Park et al [3] in their investigation observed that, the heat transfer coefficient of the
evaporator increased with increase in power input and the effect of the fill charge ratio was nearly negligible. Saleem et al [4] demonstrated that using wick structure, the heat transfer coefficient was augmented by 55%, 25% and 75% at tilt angles of 30°, 60° and 90° with reference to horizontal position respectively. At 90° tilt angle, without wick, evaporator temperature reached steady state at 180 °C whereas the condenser temperature reached steady state at 140 °C, while transporting 100 W of heat. Ong and Haider-E-Alahi [5] investigated the hysteresis effect on thermosyphon for different fluids viz., R-22, R-134a and distilled water. The effects of different fill ratios with respect to evaporator volume and the difference between the evaporator and condenser temperature on overall heat transfer coefficient were analyzed. Water performs better and requires higher starting temperatures compared to R-22, R-134a. Further at a temperature difference of 10 °C between the evaporator and condenser, the overall heat transfer coefficient was found to be 250 W/m²°C. As quoted by Amir Faghri [2], Imura et al [6] found that the ratio of the liquid-fill volume to the total volume had a little effect on the mean heat transfer co-efficient in the evaporator section. It was also observed that for fill ratio less than 0.10 the liquid film breaks down into rivulets, decreasing the heat transfer co-efficient. The optimal fill ratio was determined to be in the range 0.10 to 0.20 i.e., between 30%-60% fill ratio of evaporator volume. Seok Hwan Moon et al [7] carried out an experimental study on the performance of miniature heat pipes with woven wire wick. It was found that the thermal resistance is reduced with an increase in the thermal load. Further, optimal fill ratio was shown to be at minimum thermal resistance at 29.3 % and 31% of the total volume for 3 mm and 4 mm diameter heat pipes respectively. Maximum thermal resistance of 3 °C/W and minimum of 1.5 °C/W was observed at 2W and 8W respectively. Seok Hwan Moon et al [8] in another study investigated on using miniature heat pipe with woven-wire wick for cooling a notebook PC. The cross-sectional area of the pipe was reduced by about 30% of the original by pressing 4 mm to 2 mm diameter. When the wall thickness was reduced from 0.4 mm-0.25 mm the heat transfer limits and the thermal resistance were improved by 10%. Thermal resistance decreased at increased thermal load.

From the literature it can be observed that the effect of varying vacuum on the performance of heat pipe is not reported and this motivated the present investigation. The advancement in science and technology reported in the literature has been taken in to account and an attempt has been made in this preliminary investigation to characterise a gravity assisted heat pipe. The heat pipe is tested with and without working fluid for different power inputs, vacuum levels and fill ratios with respect to evaporator volume and transient and steady state experiments are conducted to derive various output parameters such as slope of temperature rise curve, thermal resistance and convective heat transfer co-efficient.

2. Experimentation

The experimental setup consists of a gravity assisted heat pipe, data acquisition system, vacuum pump, digital Pirani gauge and a power supply unit. The installation of test rig is shown in Fig 1. The heat pipe is fabricated using a copper tube of 150 mm length, 5 mm inner and 8 mm outer diameter. The condenser section is brazed with six annually spaced rectangular cooling copper fins of length 50 mm, width 15 mm.
and of thickness 1 mm. Thermal load is supplied by a mica band heater having inner diameter of 8 mm, length 50 mm of 50W, 230V. This is used for providing the required heat source at the evaporator section. Omega-201 thermal grease is applied to reduce the contact thermal resistance between the heater and the

![Experimental Test Rig](image)

**Fig 1: Experimental Test Rig**

The evaporator and adiabatic sections of the heat pipe were insulated using glass wool (20 mm thick) to minimize the convective heat loss from the outer surface of the heat pipe. Dimmerstat and wattmeter were provided to control and measure the electrical power input to the heater respectively. The calibrated J- Type thermocouples (1 mm diameter) soldered at equal intervals on the wall of the heat pipe are used as temperature sensors.

![Location of thermocouples on gravity assisted heat pipe](image)

**Fig 2. Location of thermocouples on gravity assisted heat pipe**

Fig 2 shows the location of the thermocouples; three points each on the wall of the evaporator, adiabatic and condenser sections respectively. A fifteen channel DAQ (Data Acquisition System) is used to acquire the temperature data. A fan and a duct are installed at the condenser fins to achieve forced convection cooling. Working fluid is metered and charged through the fluid inlet valve and sealed under required vacuum level. Vacuum level is measured by a digital Pirani gauge (Hindhivac Pirani Model HPS-2) with pressure range of 1000-0.001mbars. The results of the present study may include the error rate in voltage by ±0.05V and current by ±0.01A and in the temperature measurement by ±0.1°C. Experiments were conducted with and without working fluid (distilled water) in the heat pipe. The heat pipe without working fluid essentially represents metallic conductor. Its performance is considered as the base for the evaluation of heat pipe (i.e. with working fluid in it). The transient tests were conducted on the heat pipe, in which the heater is put “on” and the temperature rise was observed at regular intervals till the steady state is achieved. Experiments were repeated for different heat inputs with different fill ratios and different vacuum
levels. Various plots were drawn to study the performance of gravity assisted heat pipe.

3. Results and Discussions

Fig. 3 presents the variation of evaporator temperature with time for 6W heat input at various vacuums with 60% fill ratio. The fill ratio of a heat pipe is defined as the ratio of amount of liquid in the pipe to the internal volume of the evaporator section of the pipe. It is observed that the steady state reaches early in the case of 400 mbar at 840 seconds and the temperature measured was 51°C with a slope of 1.32 °C/min. Slopes are extracted at 63.3% of steady state temperature value.

At 150 mbar vacuum steady state reached at 1020 seconds showing the same steady state temperature of 51°C with the slope of 1.29 °C/min, however at 1000 mbar vacuum the steady state is reached at elaborated time (1590 seconds) with a slope of 1.34 °C/min.

![Graph of evaporator temperature varying with time at different vacuums](image)

**Fig. 3. Variation of evaporator temperature with time for 6 watt heat input at various vacuums for 60% fill ratio.**

Fig. 4 presents the variation of the condenser temperature with time for 6W heat input at varied vacuums at 60% fill ratio. It is observed that the steady state reaches early in the case of 400 m bars at 840 seconds and the temperature recorded was 41°C with a slope of 0.889 °C/min. At 150 mbar vacuum, temperature reaches steady state of 41 °C with a slope of 0.872 °C/min. At 1000 mbar steady state is reached with a slope of 0.894°C/min.

![Graph of condenser temperature varying with time at different vacuums](image)
Fig. 4. Variation of condenser temperature with time for 6 watt heat input at various vacuums for 60% fill ratio.

It is observed that the slopes are decreasing with increase in vacuum levels. This trend indicates that at increased vacuum levels inside the heat pipe the overall steady state temperature is reduced in addition to increased response of the system. Similar trends are observed for other fill ratios. A comparative plot of axial temp distribution for dry run, 400 mbar and 1000 mbar vacuum with 60% fill ratio is shown in Fig. 5. In case of dry run (Fig. 5), the slope of the axial temp distribution increases as the heat input increases and shows the larger temp difference between evaporator and condenser sections. In the case of 60% fill ratio and 1000 mbar vacuum, the slopes of axial temperature distribution are reduced for similar heat inputs. The slope is further reduced under 400 mbar as shown in the Fig. 5. The minimum temperature difference observed between the evaporator and condenser is 5°C for 2W, 8°C for 4W and 10°C for 6W heat inputs. Similar trend is observed in transient and axial temperature distribution plots for different percentage of fill ratios.

Fig. 5. Axial temperature distribution for dry run, 400 mbar and 1000 mbar vacuum for varied heat input at 60% fill ratio.

In Fig. 6 the thermal resistance tends to decrease with increase in thermal loads. This trend indicates that the temperature drop across evaporator and condenser does not increase at the rate at which heat input is increased. This leads to increase in vapour density and the heat transfer rate.
Fig. 6. Variation of thermal resistance with heat input at different fill ratios of evaporator volume at 400 mbar.

The results obtained are in well agreement with the experimental results of Seok Hwan Moon et al. [7]. Thermal resistance at 25% fill ratio of evaporator volume shows high levels at all heat inputs where as other fill ratios do not show much difference especially at higher heat transport conditions.

The overall heat transfer co-efficient of heat pipe is computed by

\[
h = \frac{Q}{A_e (T_e - T_c)} \text{ W/m}^2 \text{ °C} \quad (1)
\]

Fig. 7 shows the variation of overall heat transfer coefficient with the heat input for various fill charge ratios at 400 mbar. It is clear that overall heat transfer coefficient is increasing with increase in the heat input. Also it is evident that 60% fill ratio shows an increase in the heat transfer coefficient with a maximum value of 502 W/m\(^2\) °C at 8W heat input. 74% increase in the heat transfer coefficient is observed when compared with the dry run value of 134 W/m\(^2\) °C. However, this monotonous increasing trend of heat transfer coefficient with increased thermal load is limited by the burnout at highest heat input at this state the rate of condensate return will be lesser than the rate of evaporation leading to “starving” at the evaporator section.

Fig. 7. Variation of overall heat transfer coefficient with heat input at different fill charge ratio with respect to evaporator volume at 400 mbar.

Fig. 8 shows the variation of heat transfer coefficient with different fill ratios and vacuum pressures at 6W heat input. The result shows minimum value of heat transfer coefficient of 128 W/m\(^2\) °C for dry run and a maximum value of heat transfer coefficient of 477 W/m\(^2\) °C at 400 mbar at 60% fill ratio. The results are in good agreement with [5]. It is also observed that 65% increase in overall heat transfer coefficient at 400 mbar is obtained as compared with that of dry run.
Fig. 8. Variation of heat transfer coefficient with fill charge ratios at different vacuum levels for 6W heat input.

4. Conclusions

- A Gravity assisted heat pipe of a 10 w capacity has been successfully developed, fabricated and tested.
- Different operating characteristics were drawn at different heat inputs viz. 2w, 4w, 6w, 8w by varying the fill ratio with respect to the evaporator volume at different vacuum levels i.e.1000, 400 and 150 mbars.
- At increased vacuum levels inside the heat pipe, the operating temperature decreases and system responds faster to the thermal loads.
- The operating heat pipe with 60% fill ratio at 400 mbar vacuum has lesser overall resistance when compared to other vacuum levels and fill ratios. For a 6W heat input capacity, the thermal resistance observed for dry run is 6.22°C/W and that observed for 400 mbar vacuum is 1.66 °C/W at 60% fill ratio.
- The overall heat transfer coefficient of heat pipe increases with increase in heat input for the range of inputs tested for 60% fill ratio at 400 mbar vacuum. Also it is evident that 60% fill ratio shows a linear increase in the heat transfer coefficient with a maximum value of 478 W/m² °C for 6 W heat input.
- The fill ratios of working fluid greater than 25% of volume of evaporator and less than 60% of the volume of the evaporator shows better results in terms of increased heat transfer coefficient and decreased thermal resistance.
- In general, lower vacuum levels also indicate lower thermal resistance and higher heat transfer coefficients.

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References