

PERFORMANCE OF HEAT PIPE FOR DIFFERENT WORKING FLUIDS AND FILL RATIOS

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Abstract: An attempt is made to design, fabricate and test a miniature heat pipe with 5 mm diameter and 150 mm length with a thermal capacity of 10 W. Experiments were conducted with and without working fluid for different thermal loads to assess the performance of heat pipe. The working fluids chosen for the study were same as those commonly used namely, water, methanol and acetone. The temperature distribution across the heat pipe was measured and recorded using thermocouples. The performance of the heat pipe was quantified in terms of thermal resistance and overall heat transfer coefficient. The amount of liquid filled was varied and the variation of the performance parameters for varying liquid inventory is observed. Finally, optimum liquid fill ratio is identified in terms of lower temperature difference and thermal resistance and higher heat transfer coefficient. The data reported in this study will serve as a good database for the researchers in this field. Overall heat transfer coefficient of the Miniature heat pipe is found to be the maximum for the Acetone as working fluid.

Keywords: Heat Pipe, Working Fluid Inventory, Fill Ratio

INTRODUCTION

The heat pipe consists of a hollow tube closed at both ends and partially filled with a liquid that boils at a desired temperature. One end of the tube is immersed in the warm region and the other end in the cold region. The objective is to transfer heat through the pipe from warmer to the colder region.

Many researchers investigated the characteristics and dominating parameters of heat pipe both experimentally and theoretically^{1, 2}. A mathematical model and its numeric solution for the laminar two-phase flow of liquid and vapor of working fluid in the capillary structure of micro heat pipes was investigated by Garcia et al.³. The mathematical model is formulated for steady state, one-dimensional flow for the vapor and quasi-one-dimensional flow for the liquid. The reported results are the longitudinal distributions of the mass flow rate, the pressure and the traverse section area of the phases, and the curvature radius of the liquid-vapor interface, for a micro heat pipe with capillary structure of cross section form of four tips astroid. The relation between the maximum heat transfer capacity and the capillary pressure is analyzed. To verify the mathematical models, the results obtained are compared with data reported in the specialized literature.

Based on the momentum conservation and Laplace-Young equations, an analytical expression for the minimum meniscus radius was derived and an expression for the maximum capillary heat transport

limit in micro/small heat pipes was obtained⁴. These expressions incorporated the shear stresses at the liquid/solid and liquid/vapor interfaces, contact angle effects, vapor pressure drop, tilt angle, groove dimensions, and channel angle effects.

The original analytical model for predicting the maximum heat transport capacity in micro heat pipes, as developed by Cotter, has been re-evaluated in light of the currently available experimental data⁵. As is the case for most models, the original model assumed a fixed evaporator region and while it yields trends that are consistent with the experimental results, it significantly overpredicts the maximum heat transport capacity. In an effort to provide a more accurate predictive tool, a semi-empirical correlation has been developed.

A detailed mathematical model for predicting the heat transport capability and temperature gradients that contribute to the overall axial temperature drop as a function of heat transfer in a micro heat pipe has been developed⁶. The model utilizes a third-order ordinary differential equation, which governs the fluid flow and heat transfer in the

A	Heat transfer surface area at the evaporator (m ²)
h	Overall heat transfer Coefficient (W/m ² °C)
Q	Heat Input (W)
R	Thermal Resistance (°C/W)
T _c	Average Condenser Temperature (°C)
T _e	Average Evaporator Temperature (°C)

evaporating thin film region; an analytical solution for the two-dimension heat conduction equation, which governs the macro evaporating film region in the triangular corners; the effects of the vapor flow on the liquid flow in the micro heat pipe; the flow and condensation of the thin film caused by the surface tension in the condenser; and the capillary flow along the axial direction of the micro heat pipe. MHP has unique physical phenomenon contrary to micro heat pipes in view of affects of the operating limit, liquid locking, and length. That is, while the liquid blocking phenomenon occurs in the micro heat pipes of less than 1 mm, the condenser liquid are accumulated at the end of the condenser and heat is not delivered completely. The phenomenon of reducing the vapor temperature and thus reducing the maximum rate of heat transfer occurs in MHP if the condenser is cooled excessively. And there appear significant effects caused by the entire length of a heat pipe and the effect by the capillary limit among operating limits of a heat pipe⁷. The selection of the proper heat pipe cooling solution is dependent upon the developer's specifications, design constraints and budget.

The characteristics of counter-flow heat exchanger units, using heat pipes or two-phase closed thermosyphon as the heat-transfer element, are studied experimentally and a simple analytical model was developed to predict the performance of such units using thermosyphon⁸. The maximum heat-transfer rate has a unique functional relationship between the ratio of two stream mass flow rates, and the ratio of heated to cooled lengths of the heat-transfer elements, regardless of element bundle geometries. A heat transfer analysis of an inclined two-phase closed thermosyphon was developed by Zuo and Gunnerson⁹. The inclination-induced circumferential flow was unfavorable with respect to dry out because the thin top-side liquid film was easier to boil off, but contrastingly was favorable with respect to flooding because the thick underside film corresponded to a large gravity force.

A network heat pipe concept employing the boiling heat-transfer mechanism in a narrow space was investigated by Cao and Gao¹⁰. Two flat-plate wickless network heat-pipes (or thermal spreaders) are designed, fabricated, and tested based on this concept. The fabricated thermal spreaders, which are made of copper or aluminum, are wickless, cross-grooved heat transfer devices that spread a concentrated heat source to a much larger surface area. As a result, the high heat flux generated in the concentrated heat source may be dissipated through a finned surface by air cooling. The network heat pipes are tested under different working conditions and orientations relative to the gravity vector, with water and methanol as the working fluids. The maximum heat fluxes achieved are about 40 W/cm² for methanol and 110W/cm² for water with a total heat input of 393 W.

EXPERIMENTAL METHODS

The heat pipe is fabricated using a copper tube of 150 mm length and 5 mm inner dia and 8 mm outer diameter (Fig. 1). Ni-Cr wire having inner diameter 8 mm and length 50 mm was used to make a heater of 230V, 50W capacity and heater was used for providing the required heat source at the evaporator. The evaporator and adiabatic sections of the heat pipe are insulated using asbestos to minimize the heat loss through these portions. Variac and multimeter were provided to control and measure the power input respectively. K- Type thermocouple wires were used as temperature sensors (thermocouples numbering are shown in the Fig. 1, the position of thermocouple 1 is 20 mm from the base, position of thermocouple 2, 3, 4, 5 and 6 are respectively 40, 70, 90, 120 and 140 mm from the base). A simple 8- channel digital temperature indicator is used to measure the temperature. Five copper fins of length 50mm, width 15 mm, and thickness 0.5mm were brazed on the condenser end. Experiments were conducted with dry run (i.e. without working fluid in the tube) and wet run (with working fluid inside). 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 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 various plots were drawn to study the performance of miniature heat pipe to optimize the fluid inventory.

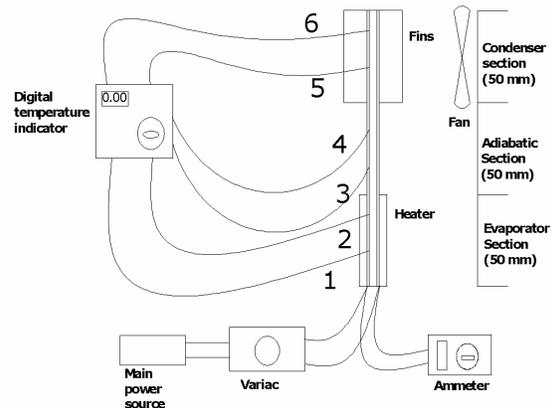


Fig. 1: Schematic Arrangement of experimental setup

EXPERIMENTAL PROCEDURE

The test section consists of three parts, as mentioned earlier, evaporator, adiabatic and condenser sections. In the experiment the heat transfer characteristics were measured for three different liquids (distilled water, methanol and acetone). Also the characteristics were measured for dry run condition (without any liquid). So, two miniature heat pipes were fabricated. For dry run

condition the heat pipe was sealed at bottom and top. In case of the heat pipe where liquids were used the bottom was sealed and top was sealed by a cork. Ni-Cr thermic wire was wound round the evaporator section. Power to the heater was provided from line supply through a variac. Fins were attached at the condenser section and a fan was directed towards the fins for forced convection to occur at this section. Six sets of thermocouple wires were fixed with the body by means of glue. At first each thermocouple sets were fused together at the top point and it was ensured that except the top point, they do not touch at any other points. Then they were attached with the body. The other ends of the thermocouple wires were connected with the digital thermocouple reader by means of connecting wires. Thermocouples were placed at six points on the surface of the heat pipe, two at evaporator section, two at adiabatic sections and two at condenser section. Thermocouples at each section were placed at an interval of 20 mm.

Experiments were conducted with dry run (without any working fluid in the tube) and wet run (with working fluid inside). The heat pipe without working fluid essentially represents metallic conductor. Its performance is considered as the base for the evaluation of the heat pipe (with working fluid in it). The transient tests were conducted on the heat pipe, in which heater was put on and the temperature rise was observed at regular intervals till the steady state was achieved. After achievement of steady state the temperatures at the six points were noted by changing the positions of the selector switch. This experiment was repeated for different heat inputs, different fill ratios and for different working fluids. Various plots were drawn to study the performance of the miniature heat pipe to optimize the fluid inventory. The different heat inputs were achieved by changing the out put voltage from the variac.

Fill ratio means the percentage of the evaporator section volume that is filled by the working fluids. The fill ratios used in this experiment were 35%, 55%, 85% and 100% of the evaporator volume for all three different working fluids.

All the temperature readings, at the six points on the heat pipe surface, were taken for all three working fluids for all the fill ratios after reaching steady state condition.

MATHEMATICAL EQUATIONS USED

Effectiveness of the heat pipe is indirectly brought in terms of thermal resistance –

$$R = \frac{T_e - T_c}{Q} \text{ } ^\circ\text{C/W} \text{ ----- (1)}$$

And the overall heat transfer co-efficient is given by-

$$h = \frac{Q}{A(T_e - T_c)} \text{ W/m}^2\text{-}^\circ\text{C} \text{ ----- (2)}$$

RESULTS AND DISCUSSIONS

Experiments were carried out in dry mode (without working fluid) and wet mode (with working fluid in it). The dry mode experiment represents the heat transfer characteristics in an ordinary conductor, while the wet mode depicts the live heat pipe characteristics. Three different working fluids namely distilled water, methanol and acetone which have varying useful working range of temperature are tested in this study. The heat pipe was filled with 35%, 55%, 85% and 100% of the evaporator volume tested for different heat input and working fluids.

Axial Temperature Profiles: Axial temperature profiles are drawn from the data of temperatures that is obtained at different axial distances on the heat pipe body.

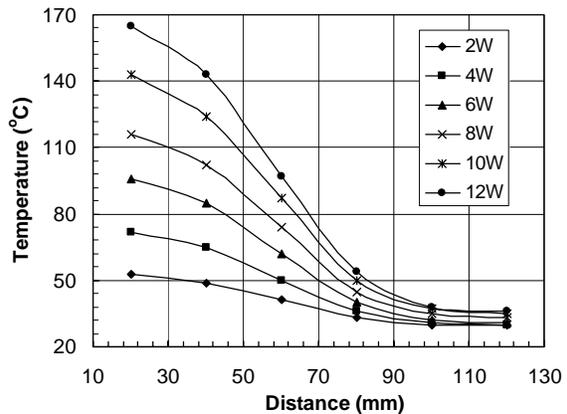


Fig. 2: Axial temperature profile for DRY RUN

The axial temperature distribution along the heat pipe for dry run and wet run (with 55% fill ratios) are shown in Figs. 2 to 5 respectively.

The evaporator, adiabatic section and condenser temperature variations with distance for dry run are shown in Fig. 2. It shows that the slope of axial temperature distribution increases with heat input and shows larger temperature differences across the condenser and evaporator section. The trend is obvious since greater temperature slope is

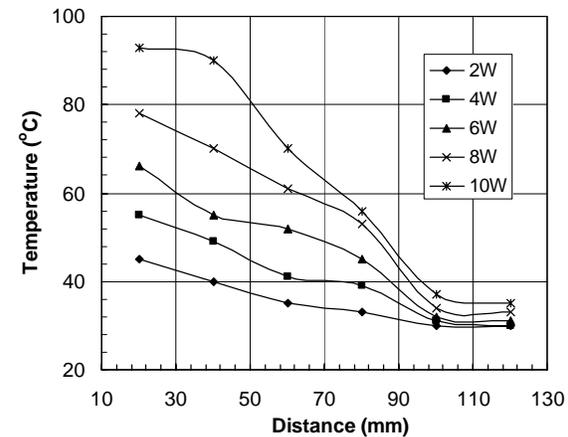


Fig. 3: Axial temperature profile for Water With 55% fill ratio

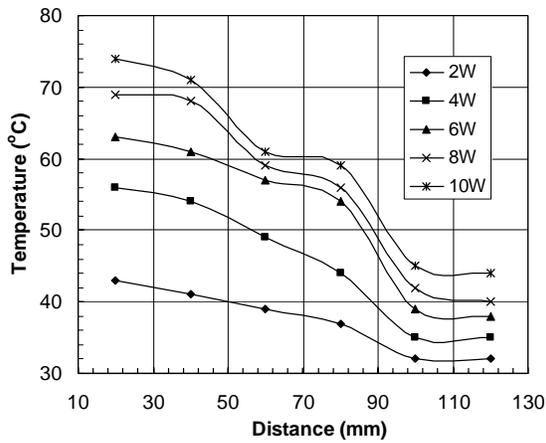


Fig. 4: Axial temperature profile for Methanol With 55% fill ratio

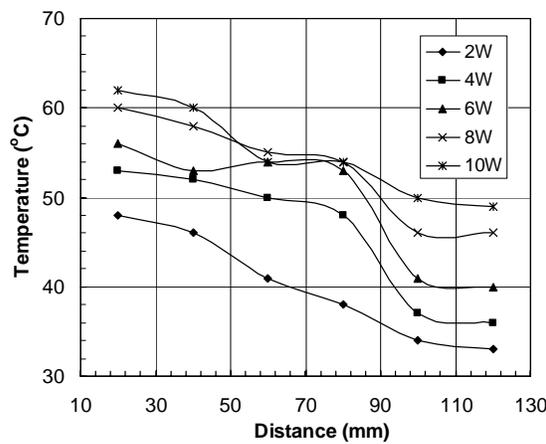


Fig. 5: Axial temperature profile for Acetone With 55% fill ratio

required for increased heat transfer in case of simple conduction heat transfer.

On the other hand, wet run shows reduced slopes of axial temperature distribution at similar heat inputs, indicating the effective augmentation of heat transfer at even reduced temperature slopes. The abrupt change in the slope of axial temperature distribution for water at 10W heat input (Fig. 3) indicates the seizure of heat pipe operation. At this stage, the rate of evaporation at evaporator is higher than condensation rate at condenser. Similar trends are found for all other working fluids.

In case of methanol the slopes of axial temperature distributions are lower than that of water and dry run and in case of acetone as working fluid the slopes are lower than that of water and methanol.

Variation of Thermal Resistances (R) with Heat Input: Figures 6-8 show the variations of thermal resistances that occur at different fill ratios for the three different working fluids at different heat input. These graphs are used for comparison of thermal resistances at different fill ratios of different working fluids.

The variations of thermal resistances with different

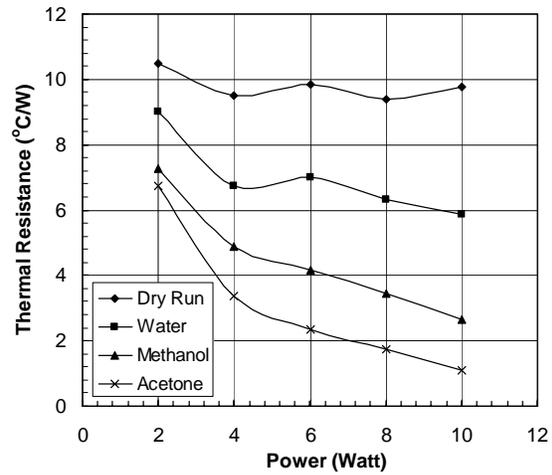


Fig. 6: variations of thermal resistance with different heat inputs for 35% fill ratio

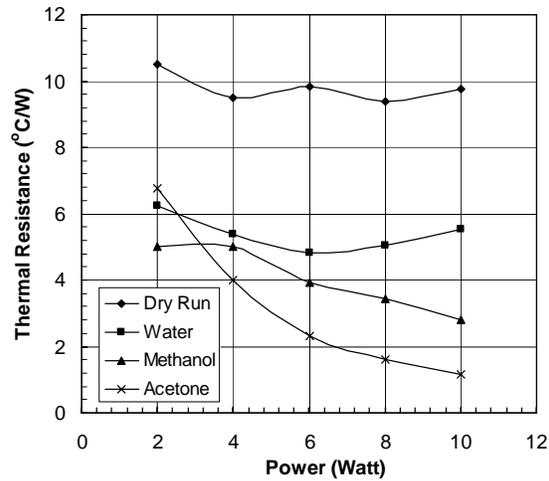


Fig. 7: Variations of thermal resistance with different heat inputs for 55% fill ratio

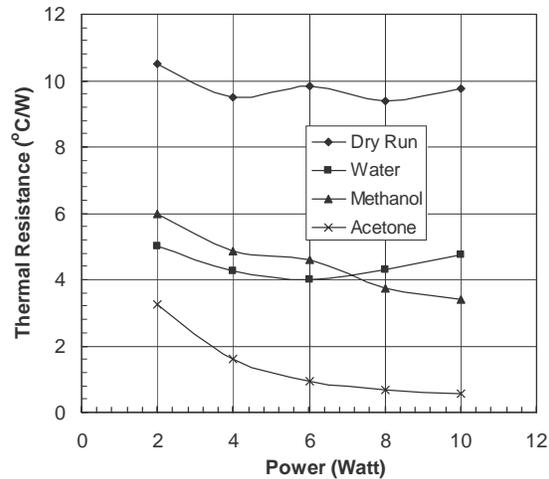


Fig. 8: Variations of thermal resistance with different heat inputs for 100% fill ratio
heat inputs for dry run and wet run (for 35%, 55% and 100%) are shown in above three figures (Figs. 6 to 8). In general wet run shows the reduced thermal resistances for all levels of heat input and all types of

working fluids. The dry run shows the largest values of thermal resistances and it is almost constant for varying heat loads. Acetone shows the minimum thermal resistances at all heat inputs for all fill ratios.

For 35% fill ratio, the graph pattern (Fig. 6) indicates that lower thermal resistances can be obtained for higher heat loads as same for other fill ratios.

For 55% fill ratio(Fig. 7), values of thermal resistances decreases for methanol and acetone but after 6W the value increases for water . So, at 55% water can perform better at 6W.

For 100% fill ratio(Fig. 8), water and methanol shows almost similar values of thermal resistances, but acetone shows minimum values for higher heat inputs.

Variation of Overall Heat Transfer Co-efficient (h) with Heat Input: Figures 9-11 show the variations of overall heat transfer co-efficient that

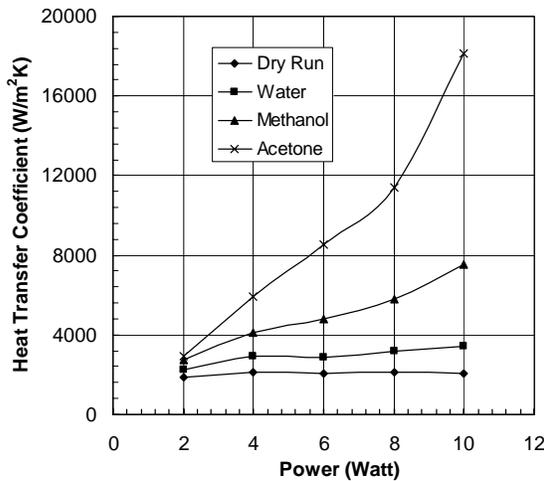


Fig. 9: Overall heat transfer co-efficients with different heat inputs for 35% fill ratio

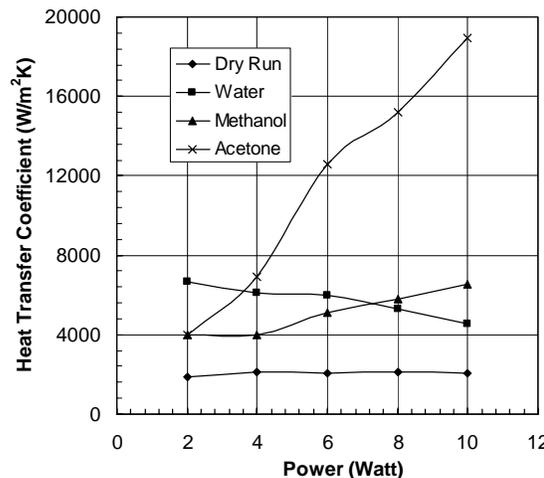


Fig. 10: Overall heat transfer co-efficients with different heat inputs for 85% fill ratio

occur at different fill ratios for the three different working fluids at different heat input. These graphs are used for comparison of heat transfer co-efficients at different fill ratios of different working fluids.

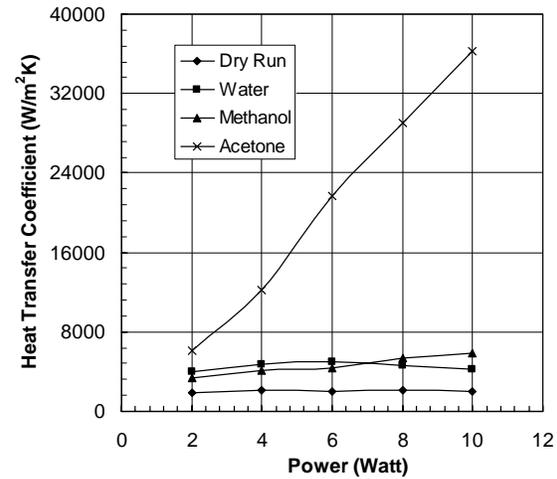


Fig. 11: Overall heat transfer co-efficients with different heat inputs for 100% fill ratio

The dry run shows an overall heat transfer co-efficient of around 2000 W/m²·°C corresponding to the forced convective heat transfer at the fin end. When the heat pipe is charged with working fluids, there are remarkable increase in heat transfer co-efficient owing to the augmentation of heat transfer rate by the evaporation and condensation process in side the heat pipe.

For 35% fill ratio (Fig. 9), water shows nearly constant value of heat transfer co-efficients, values for methanol, as working fluid, increases slightly with the increment of heat input. In case of acetone the heat transfer co-efficient increases very rapidly with input heat.

For 85% fill ratio (Fig. 10), the value of heat transfer co-efficient falls very slowly with the increment in heat input for water as working fluid, increases slowly for methanol and increases very rapidly when acetone is acting as working fluid.

For 100% fill ratio (Fig. 11), water and methanol shows almost similar pattern of heat transfer co-efficient, but acetone shows greater values for higher heat inputs.

Heat pipe with acetone shows very high heat transfer co-efficient. However, this monotonous trend of increasing value of heat transfer co-efficient with increased load is limited by the burn out at highest heat input. As explained earlier, at this state the rate of condensate return will be lesser than the rate of evaporation, leading to “starving” at the evaporator section.

Identifying The Optimum Fluid Fill Ratio:

Comparative plot of temperature difference between the evaporator and condenser section at varying fill ratio of working fluid as a percentage of evaporator volume for all the three working fluids with the heat

loads of 6 W and 10 W are shown in the Figs. 12 and 13. In all the cases acetone shows minimum temperature differences at all fill ratios. Hence it can be stated that for the temperature ranges tested in this study, acetone forms the best working fluid.

In the case of methanol and water, the fill ratio have minimum effect on the temperature difference between evaporator and condenser. On the other hand, acetone shows reduced temperature difference at higher fill ratios. With acetone as working fluid, 100% fill ratio of evaporator volume shows the best result with minimum temperature difference across the evaporator and condenser.

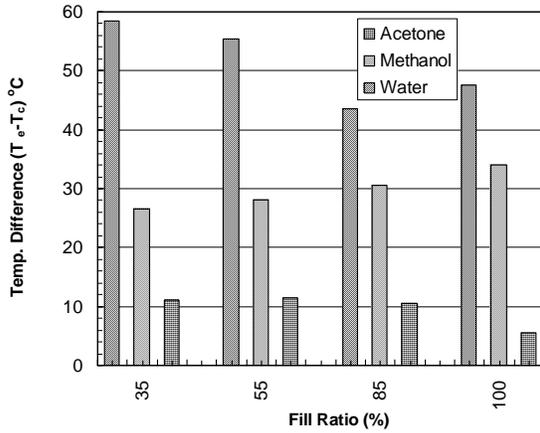


Fig.12: Temperature vs fill ratio for different working fluids for input heat of 6W

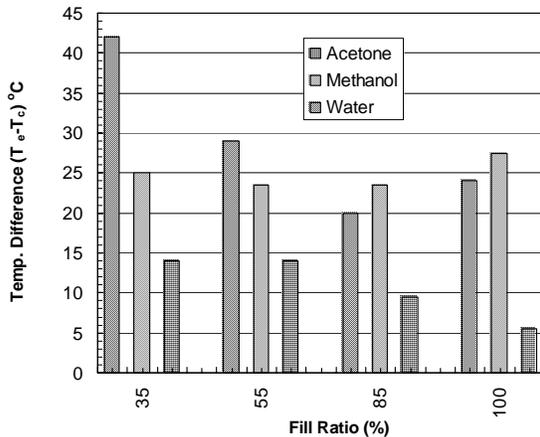


Fig.13: Temperature vs fill ratio for different working fluids for input heat of 10 W

Variations of Heat Transfer Co-Efficients and Thermal Resistances with Varying Heat Loads: The effect of fill ratio of working fluid on heat transfer co-efficients and thermal resistances are shown in Figs. 14 to 17.

In case of water as working fluid (Fig. 14 and Fig. 16), it is observed that it shows maximum value of heat transfer co-efficient and minimum value of thermal resistance at 85% fill ratio. Lower and higher than 85% fill ratio results in lower values of heat transfer co-efficients and higher values of

thermal resistances than that of 85%. So, it can easily be stated that for water as the working fluid, a miniature heat pipe will perform its best at 85% fill ratio.

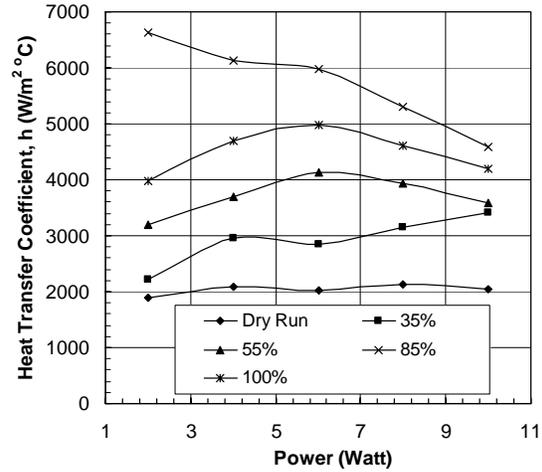


Fig. 14: Variations of heat transfer co-efficients with varying heat loads for water

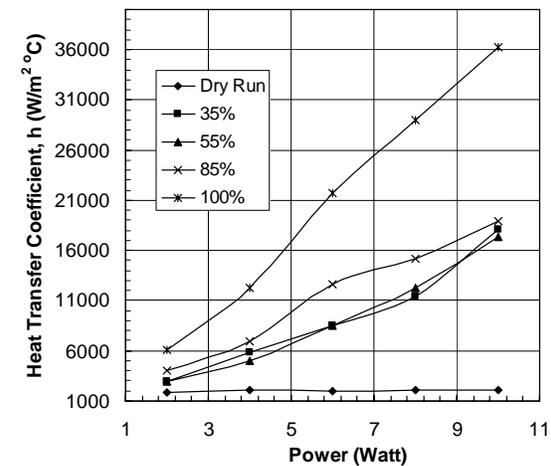


Fig. 15: Variations of heat transfer co-efficients with varying heat loads for Acetone

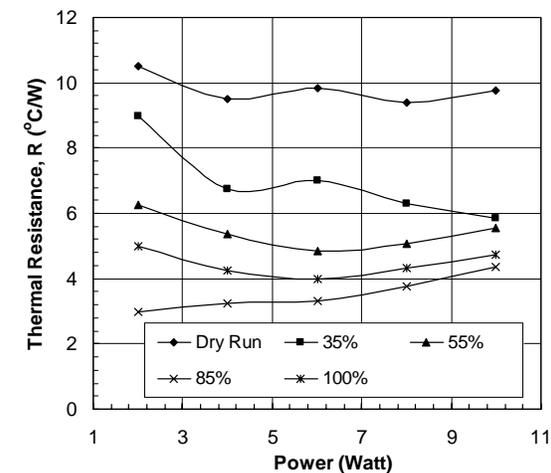


Fig. 16: Variations of heat thermal resistances with varying heat loads for Water

In case of acetone as working fluid (Fig 15 and Fig 17), it is clear that at higher fill ratios of acetone a miniature heat pipe will perform good as it shows higher values heat transfer co-efficients and lower values of thermal resistances at higher fill ratios. For lower fill ratios the performance of miniature heat pipe degrades, if acetone is working fluid.

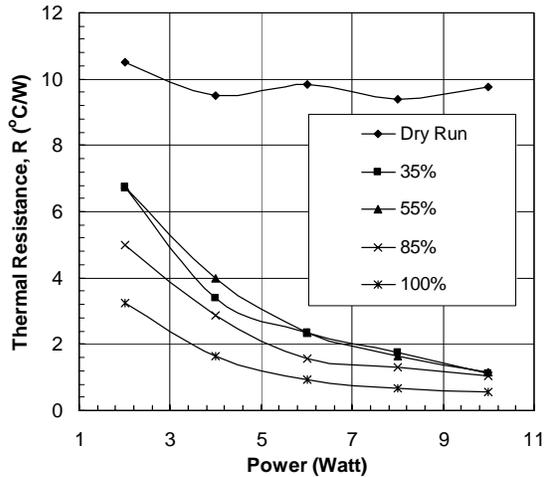


Fig. 17: Variations of thermal resistances with varying heat loads for Acetone

CONCLUSIONS

A miniature heat pipe of a 10 W capacity has been successfully developed, fabricated and tested. Different operating characteristics are presented at different heat inputs viz, 2W, 4W, 6W, 8W; 10W. The system reaches steady state early in case of wet run when compared to dry run. From the investigation, the following findings are obtained:

- The steady state temperature increases with increased heat loads. Slope of axial temperature distribution in dry run increases with the heat input, on the other hand the wet run shows an averaged constant temperature slopes.
- The operating heat pipe with wet run has lesser overall thermal resistance when compared to dry run. For a 2W heat input capacity, the thermal resistance observed in the dry run was 10.5 °C/W and that in wet run was 7.25 °C/W.
- The overall heat transfer coefficient of heat pipe increases with increase in heat input, in the range of inputs tested for acetone and methanol, while water filled heat pipe shows a nearly constant value.
- The fill ratio of working fluid as a percentage of evaporator volume is shown to have minimum effect on the performance of heat pipe with respect to the temperature difference when water and methanol are used as working fluids.

- In case of acetone, the temperature difference across evaporator and condenser continues to drop down with an increase in the fill ratio.
- With acetone as the working fluid, 100% fill ratio of evaporator volume shows the best result with minimum temperature difference across the evaporator and condenser.
- In general, fill ratios of working fluid greater than 85% of volume of evaporator show better results in terms of increased heat transfer coefficient, decreased thermal resistance and reduced temperature difference across the evaporator and condenser.

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