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WORKING FLUID INVENTORY IN MINIATURE HEAT PIPE

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ABSTRACT

The heat pipe though has a wide application; the information available towards the development of an efficient heat pipe is seldom seen in the open literature. In the present study, investigations are carried out for optimizing the fluid inventory in a typical heat pipe. A “flooded” (with exceedingly large amount of working fluid) heat pipe has slow response and has limited lower range of operation in terms of operating temperature. On the other hand, “starving” (with too little amount of working fluid) heat pipe although exhibits fast response to heat loads, shows severe limit at high temperature conditions. In the present study, 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 response time for surge loads, 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 shall serve as a good database for the researchers in this field.

Keywords: Heat Pipe, Fluid Inventory, Miniature Heat Pipe.

1. INTRODUCTION

A heat pipe is a simple device that can quickly transfer heat from one point to another. They are often referred to as the superconductors of heat as they possess an extraordinary heat transfer capacity and rate with almost no heat loss [1]. The heat transfer takes place by repeated cycles of condensation and evaporation of the working fluid within a sealed system. Hence the heat pipe transfers higher amount of heat compared to normal conductors, with a less temperature difference and can be operated over a wide range of temperature (100K–1000K) according to type of working fluid. They are found particularly useful as cooling means for modern electronic devices, which are manufactured for high performance and high degree of integration. In a heat pipe, working fluid is evaporated at the evaporator section and condensed at the condenser section. Sufficient capillary pressure is needed to balance the gravitational pressure and the pressure losses in both vapor and liquid phases [2].

The three basic components of a heat pipe are the container, the working fluid and the wick or capillary structure. The function of the container is to isolate the working fluid from the outside environment. Therefore it should be leak-proof, should be able to maintain the pressure difference across its walls and enable transfer of heat to take place from and into the working fluid. The first consideration in the manufacturing of heat pipes is the identification of suitable working fluid and wick material. Based on the availability of operating temperature and pressure one has to determine the most acceptable working fluid and wick structure for the application considered. The heat pipes instead of gravity or mechanical work, utilizes capillary induced fluid flow for their operation.

A wickless network heat pipe for high heat flux spreading applications was developed by Cao, Y. and Gao, M. [3]. In this study the concept of the network heat pipe employing the boiling heat-transfer mechanism in a narrow space has been described. Two flat-plate wickless network heat pipes (or thermal spreaders) were designed, fabricated, and tested based on this concept by the authors. The fabricated thermal spreaders, which were made of Copper or Aluminum, were 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 could be dissipated through a finned surface by air cooling. The network heat pipes were tested under different working conditions and orientations relative to the gravity vector, with water and methanol as the working fluids. The maximum heat flux is achieved was about 40 W/cm^2 for methanol and 110W/cm^2 for water with a total heat input of 393W.

A heat transfer analysis of an inclined two-phase closed thermosyphon was developed by Zuo, Z. J. and Gunnerson, F.S. [4]. 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. Minimum working fluid inventory remained almost constant for a large range of inclination angles (0-70 deg) and then significantly increased for further increase of inclination angle. At a certain inclination angle, the mean heat transfer coefficient of the thermosyphon reached a maximum value, which was related to the heat transfer behavior in both condenser and evaporator. The highest flooding limit was at inclination angle ranging from 30 to 45 deg, which corresponded to the best balance of the two opposing effects: secondary circumferential flow and gravity reduction.

Zhang, J. and Wong, H.[5] studied heat transfer and fluid flow in an idealized micro heat pipe with the support of NASA and LaSPACE. They made an analysis for four different values of length to width ratio of an idealized micro heat pipe, viz. 20, 50, 100, and 200. From the liquid temperature distribution along the length of the micro heat pipe, they found that the temperature profile is relatively flat except the region near the evaporator, and for a micro heat pipe with larger length to width ratio, the length of the evaporator is shorter. From the vapor pressure distribution, they found that the pressure goes approximately linearly and is not strongly affected by the length to width ratio. On evaluating the effective thermal conductivity of a micro heat pipe increases with increase in the evaporation area at the evaporator, and length or width of the micro heat pipe. They also added that a fluid with larger latent heat would produce larger effective thermal conductivity.

In a study of micro and miniature heat pipes, developed by A.R. Anand [6], attempts have been made to develop a one dimensional numerical model of micro heat pipes, taking into account the effect of liquid-vapor interfacial shear stress. Also governing equations for conservation of mass, momentum, and energy have been developed, based on control volume to study the performance characteristics and validate the experimental results.

To identify and understand better the phenomena, which governs the performance limitations and operating characteristics of micro heat pipes, Babin et al. [7] conducted a combined experimental and analytical investigation on two micro heat pipes, one Copper and one Silver, of length 57mm and cross section 1 mm^2 approximately with water as the working fluid. The steady-state experimental results obtained were compared with an analytical model and were found to predict accurately the experimentally determined maximum heat transport capacity for an operating temperature range of 40 °C to 60 °C. The results indicated that the steady state model could be used to predict accurately the level of performance.

In 1991, Wu and Peterson [8] developed a transient numerical model capable of predicting the thermal behavior of micro heat pipes during start up or vibration in the thermal load of evaporator. The numerical model was used to identify, evaluate, and better understand the phenomena, which governs the transient behavior of micro heat pipes as a function of physical shape, the properties of the working fluid, and the principal dimensions. The results were compared with the steady state results obtained by Babin et al. in 1990 and were shown to accurately predict the steady state dry out limit also. The wetting angle was also found to be one of the most important factors contributing to the heat transport capacity. But no experimental data were obtained on the transient operational characteristics.

In 1994, Faghri et al.[9] developed their mathematical model to examine the heat and mass transfer processes in a micro heat pipe, taking into account the variation of the curvature of the free liquid surface and the interfacial shear stress due to liquid–vapor interaction. The model described the distribution of the liquid chart in a micro heat pipe and its thermal characteristics depending on the charge of the working fluid and the heat load. It was observed from the modeling that for the same heat pipe, the charge required when interfacial shear stress is considered, is greater than the charge required if no shear is considered. Further for the same operating temperature the maximum heat transfer, when interfacial shear stress is considered, is less than the maximum heat transfer if no shear is considered.

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 develop a miniature heat pipe by following conventional basic design procedure and to optimize the working fluid inventory. The heat pipe is tested with and without working fluid for different input powers and for different fill ratios. Using the transient and steady state experimental results, different parameters such as thermal resistance, convective heat transfer co-efficient and time constant, are evaluated and different characteristic curves are plotted and analysed.

2. EXPERIMENTATION

The heat pipe is fabricated using a copper tube of 150 mm length and 5mm inner dia and 8mm outer (Fig1). A Mica Band heater having inner diameter 8 mm and length 50 mm and 230V, 50W capacity is used for providing the required heat source at the evaporator. The evaporator and adiabatic sections of the heat pipe are insulated using glass wool to minimize the heat loss through these portions. Dimmerstat and wattmeter were provided to control and measure the power input respectively. Calibrated J- Type thermocouples (1 mm dia) wires were used as temperature sensors. A simple 12- 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.

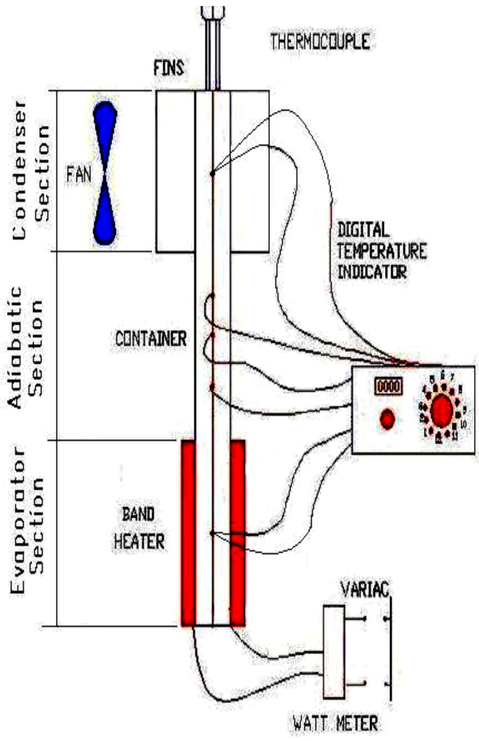

Fig 1. Experimental Test Rig

Experiments were conducted with dry run (i.e. without working fluid in the tube) and wet run (with working fluid inside). The heat pipe with out 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 various plots were drawn to study the performance of miniature heat pipe to optimize the fluid inventory.

3. RESULTS AND DISCUSSION

Experiments were carried out in "dry mode (without working fluid in it)" and "wet mode (with working fluid in it). The dry mode experiments represent the heat transport characteristics in an ordinary conductor, while the wet mode depicts the live heat pipe characteristics. Three different working fluids namely Acetone, Methanol and Distilled water which have varying useful working range of temperature are tested in this study. The heat pipe initially was filled with 35%, 55%, and 85% and100% of the evaporator volume tested for different heat input and working fluids.

The results are derived from the temperature profile measured using thermocouples. The measurement system has its own inherent uncertainties. The thermocouple – temperature display system has an uncertainty of ± 2% of full scale. The manual reading of temperature with varying time contains is estimated to posses an error of about 1% of reading. The results obtained have to be viewed with this uncertainty limit.

3.1 Transient Plots

The transient temperature profiles obtained from dry run at 2W heat input is shown in Fig 2. This represents the combined thermal inertia of heater and the dry heat pipe. The evaporator section gets heated up and transfers the heat to the condenser section by conduction and finally, the heat is dissipated to atmosphere through the fins. The temperatures reach steady state at about 1000 s from the start of the heater in dry condition and the temperature difference between the evaporator and condenser region is noticeably large.

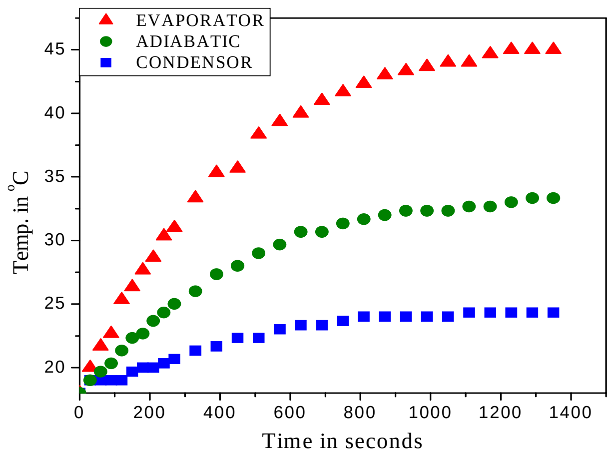

Fig 2. Transient plots for heat input of 2W for dry run (without working fluid in the heat pipe)

The transient plots for the water as working fluid with 55% of fill ratio to the evaporator volume at 2W heat input is shown in Fig. 3. The reduced temperature levels in all the three sections when compared to dry run (Fig. 2) are noticeable.

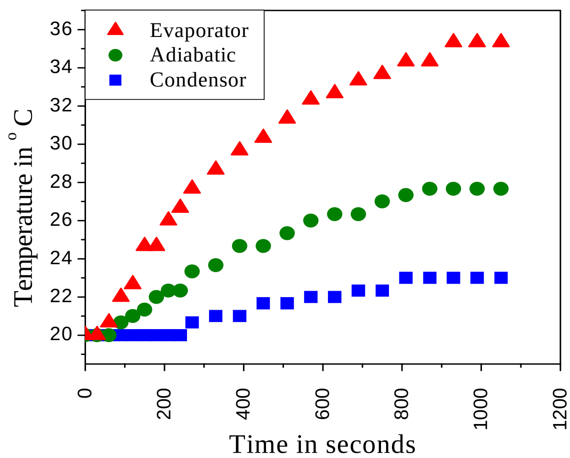

Fig 3. Transient plots for heat input of 2W for water with 55% fill ratio.

The combined time constant plots are shown in Fig 4, wherein, the time constant derived from Figs. 2 and 3 are shown for varying power inputs for dry run and wet run with water at 55% fill ratio of volume of evaporator. Obviously, the wet-run show not only the reduced temperature levels as depicted in Fig. 3 but also shows a faster response with reduced time constants (Fig. 4). In General, the time constant reduces with increase in heat input in all the cases indicating augmented energy transfer at higher heat loads.

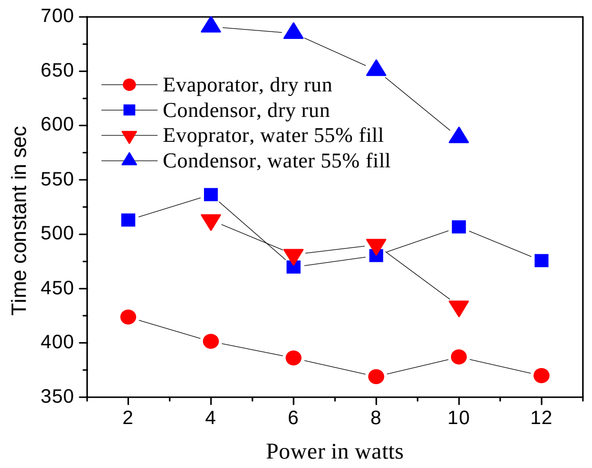

Fig 4. Time constant variation for varying input in dry run (without working fluid) and with water as working fluid (55% fill ratio)

3.2 Axial Temperature Distribution

The axial temperature distribution along the heat pipe for dry run and wet run (water, 55% fill ratio) are shown in Fig. 5 and Fig. 6 respectively, wherein the evaporator, adiabatic and condenser temperature variations are shown. In case of dry run (Fig. 5) the slope of axial temperature distribution increases with the heat input and shows larger temperature difference across the evaporator and condenser section. The trend is obvious, since greater temperature slope is required for increased heat transfer in case of simple conduction heat transfer.

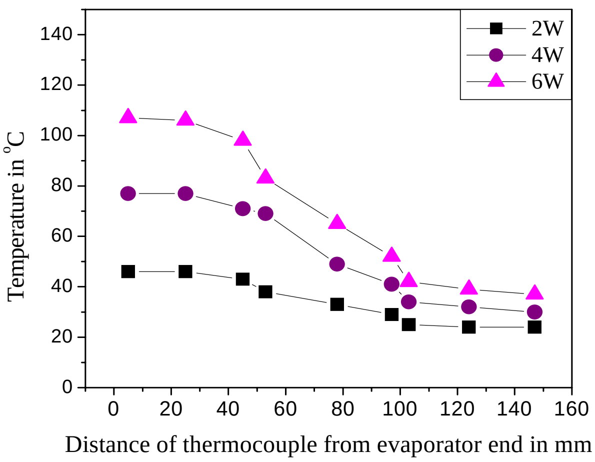

Fig 5. Axial temperature profile for dry run

On the other hand the wet run (Water, 55% fill ratio), shows reduced slopes of axial temperature distribution (Fig. 6) at similar heat inputs, indicating the effective augmentation of heat transfer at even reduced temperature slopes. The abrupt change in the slope of the axial temperature distribution for 6W heat input (Fig. 6) indicate the seizure of heat pipe operation. At this stage, the rate of evaporation at the evaporator is higher than the condensation rate at the condenser resulting in dry out. Similar trends in transient plots and axial temperature distributions are observed for all sorts of working fluids tested and varying heat inputs (plots not shown).

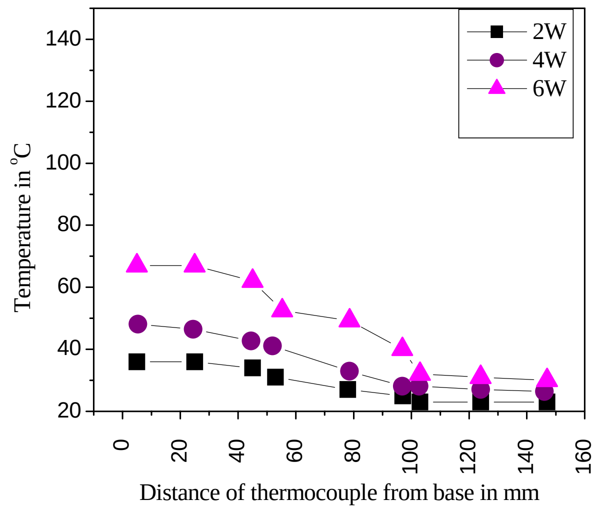

Fig 6. Axial temperature profile with water as working fluid (55% fill ratio)

3.3 Thermal Performance

Effectiveness of the heat pipe is indirectly brought in terms of thermal resistance and convective heat transfer co-efficient. The thermal resistance is computed as

$$R = \frac{T_e - T_C}{Q} \quad {}^0C/W \tag{1}$$

The overall heat transfer co-efficient is given by

$$h = \frac{Q}{A\,(T_e - T_C)} \quad W/m^2 - {}^OC \tag{2}$$

The variation of thermal resistance and overall heat transfer coefficient for dry run and wet run (with working fluids of water, acetone and methanol) for varying heat input is shown in Fig. 7 and Fig. 8 respectively. In general, the wet run shows reduced thermal resistance and enhaunced heat transfer coefficient at all levels of heat input for all type of working fluids. The thermal resistance of the heat pipe without working fluid in it is about 10.5 ^{0}C/W for all the heat inputs. Heat pipe with water as working fluid exhibit a value of 6 ^{0}C/W for all the heat inputs, while acetone and methanol show reducing thermal resistance at increased heat inputs (Fig. 7).

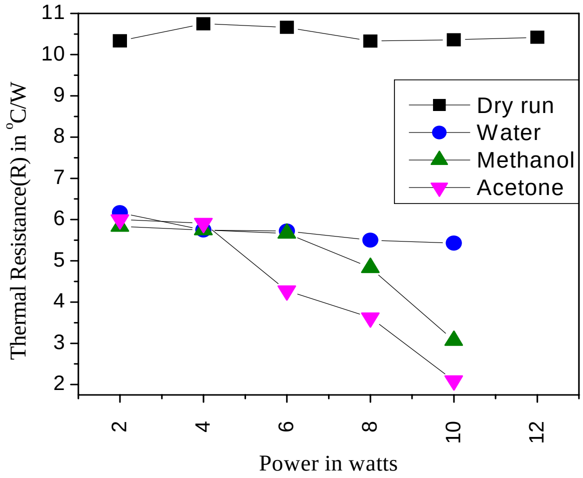

Fig 7. Variation of thermal resistance with heat input power for 55% fill ratio for different working fluids at varying heat load.

The dry heat pipe shows an overall heat transfer coefficient of 2000 W/m^2-^{0}C, (Fig. 8) corresponding to the forced convective heat transfer at the fin end. The heat pipe charged with the working fluid show a remarkable increase in the heat transfer coefficient owing to the augmentation of heat transfer rate by the evaporation and condensation process inside the heat pipe. Water as working fluid (55% fill ratio), exhibit a nearly constant heat transfer coefficient of about 3500 W/m^2-^{0}C, while acetone and Methanol show steady increase in the heat transfer coefficients at increased heat loads. Heat pipe with acetone shows a record high of heat transfer coefficients of around 9,500 W/m^2-^{0}C. However, this monotonous increasing trend of heat transfer coefficient with increased load is limited by the burnout 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.

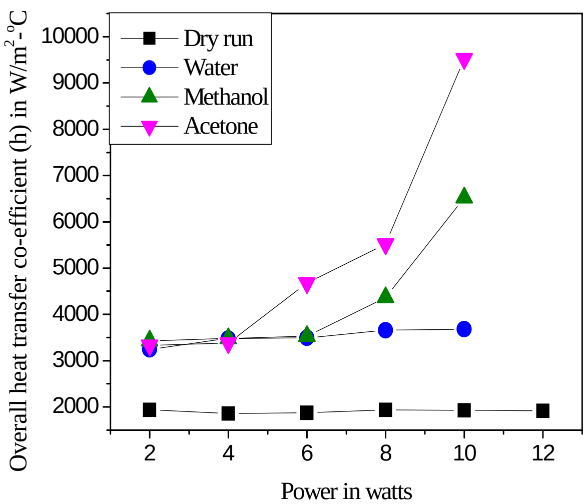

Fig 8. Variation of overall heat transfer coefficient with heat input power for 55% Fill ratio for different working fluids at varying heat load.

3.4 Fluid Inventory

Further, 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 with2 W of heat load is shown in Fig. 9. In all the cases, acetone shows minimum temperature difference at all fill ratios. Hence it is concluded that for the temperature range tested in this studies, the acetone forms the best working fluid.

In case of water and methanol, the fill ratio is shown to 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 the working fluid, 100% fill ratio of evaporator volume shows the best result with minimum temperature difference across the evaporator and condenser

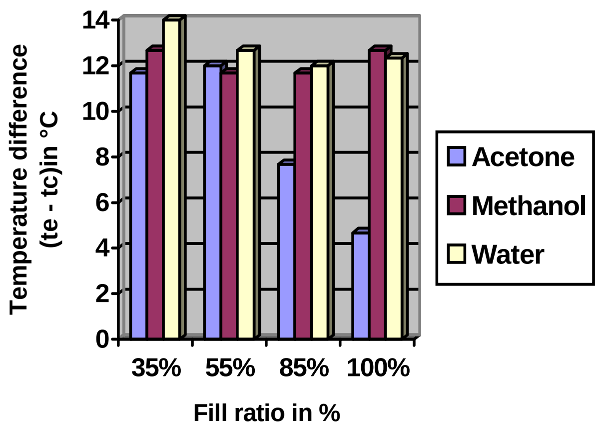

Fig 9. Temperature vs. Fill ratio for different working fluids at 2 Watts of heat input

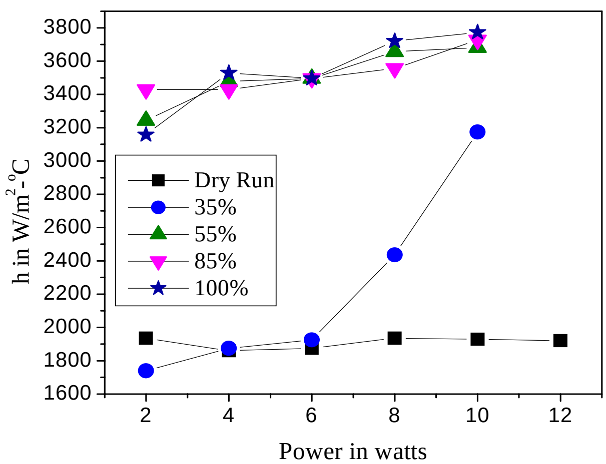

Fig 10. Variation of heat transfer coefficient with varying heat load for water at varied fill ratio of liquid. Percentage fill ratio shown is with respect to evaporator volume.

The effect of fill ratio of working fluid on the heat transfer coefficient and the thermal resistance are shown in Fig. 10 and Fig. 11 respectively with water as working fluid. Similar trends are observed in case of other fluids also (plots not shown). Both the plots indicate better performance at fill ratios greater than 35% of evaporator volume. At higher fill ratios, the heat transfer coefficient is elevated and thermal resistance is reduced as seen in these plots (Figs. 10 and 11).

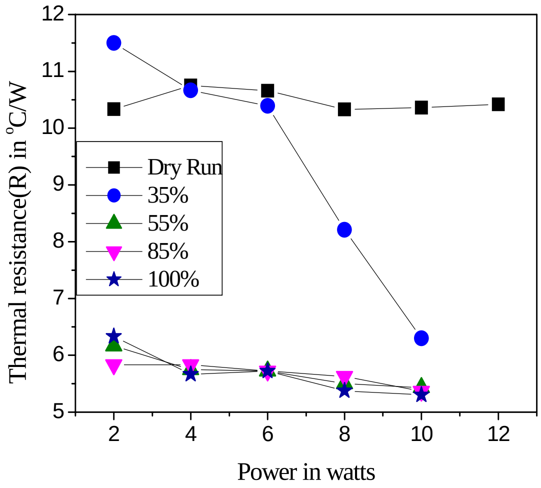

Fig. 11. Variation of thermal resistance with varying heat load for water at varied fill ratio of liquid. Percentage fill ratio shown is with respect to evaporator volume.

4. CONCLUSIONS

- A miniature 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, 10w.
- The system reaches steady state early in case of wet run when compared to dry run.
- 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 resistance when compared to dry run. For a 2W heat input capacity, the thermal resistance observed in the dry run was 10 ^{0}C/W and that in wet run was 6 ^{0}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 heat pipe shows a nearly constant.
- 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. However, 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 35% 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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6. NOMENCLATURE

Symbol	Meaning	Unit
T_e	Average Evaporator Temperature	(°C)
T_c	Average Condenser Temperature	(°C)
R	Thermal Resistance	(°C/W)
h	Overall heat transfer coefficient	(W/m^2-^{0}C)
Q	Heat Input	(W)
A	Heat transfer surface area at the evaporator	(m^2)