Experimental Investigation of Thermal Performance of Variable Conductance Cylindrical Heat Pipe Using Nanofluid

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Abstract — An experimental investigation is carried out to study the effects of nanofluid and mass of non-condensable gas on the thermal performance of variable conductance heat pipe by testing circular screen mesh wick heat pipe. The nanofluid used is water-based CuO-nanofluid with the volume fraction of 1, 3 and 5 Vol.%. The performance of the heat pipe is investigated at three different amounts of both heat input and mass of non-condensable gas (air). The wall temperatures distribution along the heat pipe using the water-based CuO nanofluid is lower than that of the heat pipe using DI water. Additionally, it increases with the increase of mass of non-condensable gas. The overall thermal resistance of the heat pipe using the water-based CuO nanofluid is lower than that of the heat pipe using DI water. Also, it decreases with the increasing of heat input and increases with the increasing of mass of non-condensable gas. The condenser inactive length increases with the increasing of mass of non-condensable gas and decreases with the increasing of nanoparticles concentration within the working fluid. The thermal resistance improvement of the heat pipe using the water-based CuO-nanofluid with the volume fraction of 5 Vol.% reaches 9.5% at coolant temperature of 18.3 oC and mass of non-condensable gas of 0.5mg.

Keywords — Cylindrical heat pipe, Nanofluid, Variable conductance, Thermal performance.

I. INTRODUCTION

Heat pipes have gradually recognized as a highly-effective heat transfer element in almost all industrial fields. Originally, the conventional heat pipe was invented by Gaugler of the General Motors Corporation in 1944, but remains without significant attention within the heat transfer community until the concept resurrected by the space program in the early 1960's. Since then, a great deal of literature has been published concerning experimental, numerical, and analytical work involving heat pipes.

For conventional heat pipes the fixed thermal conductance represents the major problem. Whereas, any change in the system, such as a change in heat load or coolant temperature will result in a change in the operating temperature of the device being cooled.

Some early means of operating temperature control include louvers, cold biasing and variable conductance heat pipe (VCHP). VCHP contain a non-condensable gas in the condensing section; see fig. 1, in order to achieve temperature control of the evaporation section under variations of thermal heat loads and changing cooling conditions, by altering the amount of condenser area available. The properties of high thermal conductivity and excellent active or passive control make VCHP an appropriate element for thermal-control and heat transfer problems of a wide range of practical applications.

Mezaache [1], performed a numerical and experimental investigations for steady-state thermal performance of a gas-loaded heat pipe. His physical modeling is based on the simple conduction model for the vapour region and a two dimensional heat and mass transfer model for the gas region. The heat pipe system is designed, with 1.05m long and 0.0889m diameter, to be experimented with variable heat input less than 800 W. The experimental evaluation of the thermal performance is made with water as working fluid and 0.1588 moles of helium as non-condensable gas. Prediction of the wall temperature along the heat pipe, and the radial and axial field of temperature and gas concentration in the vapour-gas condenser region are obtained from the analysis. The measured results agree well with numerical predictions. The results show that the mass diffusion decreases the thermal performances of the VCHP by reducing its isothermal region due to the accumulation of the non-condensable gas at vapour-liquid interface.

An analytical and experimental study introduced by Leriche et al. [2] for a Variable Conductance Heat Pipe (VCHP) applied to vehicle thermal management for the reduction of engine energy consumption after a cold start. The performance of copper/water VCHP using nitrogen as a non-condensable gas was theoretically modeled based on a nodal method and an experimental test bench was developed for this study. VCHP operate as a thermal switch, with a start-up temperature of 80 C. The present study comprises also, the effect of the air mass flow rate on the condenser and the effect of the inclination angle on the performance of the VCHP. The results show that the VCHP is an interesting solution for the vehicle thermal management in order to reduce the engine energy consumption after a cold start by controlling heating-cooling cycle of oil.

Saad et al. [3] presented a numerical and experimental investigation to evaluate the effect of non-condensable gases and axial conduction on the transient performance of copper-water wicked heat pipes. The heat pipe had an outer diameter of 19.05 mm, wall thickness of 1.65 mm and a length of 355.6 mm. while, the wick structure was 4 layers of a woven copper wire screen mesh with wire diameter of 0.109 mm and 3937 strands per meter (100 strands/inch). The effect of non-condensable gases and axial conduction was incorporated to an existing transient wicked heat pipe network model. the results show that the transient response effected by the presence of non-condensable gases. Whereas, they causes the wall to be heated along the length of the heat pipe over time as the non-condensable gases was compressed. Also, the axial conduction lengthen the region over which this occurs, but haven't a great influence on the time response during this phase.

Masaru et al. [4] experimentally described titanium heat pipe design, manufacture and tests. The heat pipe developed is titanium – ammonia –gas loaded variable heat pipe which has a heated reservoir (VCHP). Three grams as initial mass of pure nitrogen 99.99% was charged to the VCHP as non- condensable gas. This study has clarified that titanium is very desirable as heat pipe material for use in space.

For a high power photonic devices, thermoelectric modules (TEMs) are used for precision temperature control. TEMs consume a large amount of power, particularly when subjected to a wide range of ambient temperatures. Thus, the use of variable conductance heat pipes (VCHPs) as a lower power alternative to TEMs is investigated by Cleary et al. [5]. They characterize the performance of a methanol-argon VCHP with a non-wicked reservoir for both passive and active control. Moreover, they introduce the concept of an "ideal" working fluid for a gas-loaded VCHP. An experimental prototype was constructed and the obtained measurements are compared with the predictions of the flat front model.

The design and test of a pressure controlled heat pipe (PCHP) for spacecraft thermal management was discussed by Sarraf et al. [6]. The PCHP combines a conventional grooved aluminum–ammonia heat pipe with variable–volume non-condensable gas reservoir to create a heat pipe whose conductance can be precisely controlled. A prototype PCHP with variable volume control was successfully demonstrated and it was capable of maintaining a stable evaporator temperature within 0.1 ^{o}C over changes in heat sink temperature from -70 to -40 ^{o}C for input power ranging from 50 to 250W while a similarly-sized conventional variable conductance heat pipe yielded temperature swings of over 3.5 ^{o}C for the same variation of heat load and sink temperature. Therefore, PCHP approached, all of its design goals, and it is a significant advance over other means of temperature control even in its current non-optimized state.

Recently, a novel idea is considered by applying nanofluids, in which nanoparticles are uniformly and stably dispersed, as working fluids in heat pipes. This idea enhance the maximum heat transport rate and the effective thermal resistance which represent the main parameters that characterize the thermal performance of a wicked heat pipes

According to my best knowledge, there no papers presented the study on the heat transfer and flow characteristics of the variable conductance heat pipe with nanofluids. The objective of this paper is to study the thermal performance enhancement of variable conductance heat pipe with CuO-nanofluid. Effects of nanoparticles volume concentration and mass of non-condensable gas on the thermal performance of heat pipe are considered. The obtained results of the nanofluid are compared with those of the based fluid.

II. EXPERIMENTAL DETAILS

A. Preparation of Water-Based CuO-Nanofluid

A two steps method is used to prepare water-based CuO-nanofluids. The nanofluid was prepared by directly dispersing CuO nanoparticles, supplied by Nanostructured & Amorphous Materials Inc., into the distilled water in a Pyrex flask which fixed inside the ultrasonic water bath (type Elmasonic P180H and supply by Elma, Germany). The basin of ultrasonic device filled with distilled water over the mixture level in the flask by 2 cm. Then, the degas mode is switched on to remove air from the mixture.

After this, the flask sealed by the cap and oscillated continuously from 10 – 12 h in the ultrasonic water bath with a working frequency of 37 kHz and power efficiency of 100% at 60–70 °C so that the nanoparticles can be uniformly distributed. No surfactants were added into the nanofluid because they have considerable effects on the thermophysical properties of nanofluid. The volume fraction, which was the ratio of the volume of nanoparticles to that of the base fluid, was used to describe the nanoparticle concentration. In the experiments, the volume fractions of the nanofluid are 1, 3 and 5 Vol.% that corresponding to the mass concentrations w which can be estimated by the following correlation [7]:

$$\frac{1-w}{w} * \frac{\rho_{np}}{\rho_{bf}} = \frac{1-\varphi}{\varphi} \tag{1}$$

B. Experimental Set up

A schematic of the experimental test rig is shown in fig. 2(a). The tested copper heat pipe was built up locally. The length, the outer diameter and the wall thickness of the heat pipe were 555 mm, 19.05 mm and 0.85 mm respectively. There were six layers of copper meshes (mesh number 145) inside the tube. A close contact between the meshes and the inner wall can be guaranteed due to the internal tension of the meshes. The evaporator section, the adiabatic section and the condenser section of the heat pipe were 150 mm, 308 mm and 97 mm in length respectively. The evaporator section was heated uniformly by 60 W tape electrical heater and the condenser section was cooled by a constant-temperature bath.

The heat pipe is charged with 34.7 ml of DI water or the water-based CuO-nanofluids, which are about 240% of the amount required to completely saturating the wick based on the experimental testes for optimal mass inventory selection. Also, the heat pipe is charged with different amounts of non-condensable gas (air) to occupy a part of the condenser section and make the condenser inactive length.

The heat pipe is insulated by glass wool (0.038 W/m.K) to minimize heat loss to surrounding. Seven T-type thermocouples (Applent Instruments, Inc., AT4564) are fixed onto the outer surface of the test section and distribution of them is indicated in fig. 2(b).

The experiments are performed with the heat pipe in the horizontal orientation. The heat input to the evaporator is increased in steps of 5 W while the temperature of the coolant is maintained constant at 18.3 ^{o}C. The temperature distribution along the heat pipe is measured and recorded at the steady-state condition. Based on the temperature measurement results, the thermal resistance of heat pipe is considered and is defined as

$$R_{th} = \frac{\bar{T}_e - \bar{T}_c}{(Q_{in} - Q_{out})/2} \tag{2}$$

Where $\overline{T}_e$ and $\overline{T}_c$ are averaged wall temperatures at the evaporator and condenser sections, Q_{in} and Q_{out} are input and output powers. The output power Q_{out} was calculated by

$$Q_{out} = \dot{m}\, C_p (T_{in} - T_{out}) \quad (3)$$

Where T_{in} and T_{out} are inlet and outlet temperatures to the condenser sections

III. RESULTS AND DISCUSSION

A. Effect of input heat on heat pipe thermal behavior

Fig. 3 and 4 show the axial temperature distribution for pure water and 3 Vol.% of CuO- nanofluid respectively, for coolant temperature of 18.3 ^{o}C and 1.5mg of air as non-condensable gas. These figures show that with increasing the input heat flux the wall temperature along the heat pipe increases. This is because of increasing the condenser active length and the working fluid evaporation rate so that the inside pressure will increase. While, fig. 5 shows the overall thermal resistance of the heat pipe at the different input heat flux for water and 3 Vol.% of CuO-nanofluid. This figure shows that the thermal resistance decreases with increasing the heat flux because of the increasing in the condenser active length which led to reduce the evaporator-condenser temperature difference.

B. Effect of mass of non-condensable gas on heat pipe thermal behavior

Fig.s 6 and 7 respectively show the axial wall temperature distribution for pure water and 1 Vol.% CuO-nanofluid for input heat of 25 W and coolant temperature of 18.3 ^{o}C. As shown in these figures, the axial wall temperature increases as the air mass increases. This is due to increasing the pressure inside the heat pipe which led to decreasing the condenser active length and increasing the active length temperature. Also, it can be noticed from fig. 8 that the heat pipe thermal resistance increases when the air mass increases. This is due to increasing the condenser inactive length which causes the increase in the temperature difference between the evaporator and condenser sections in the presence of more non-condensable gas.

C. Effect of nanofluid on heat pipe thermal behavior

Fig. 9 shows the effect of nanoparticles concentration (NPC) on axial wall temperature distribution of the heat pipe CuO-nanofluid for input heat of 25 W, coolant temperature of 18.3 ^{o}C and air mass of 0.5mg. It can be noticed from this figure that the wall temperature decreases as the NPC increases. This is due to increasing the thermal conductivity of the working fluid and thus the operating temperature decreases in the presence of more NPC. Although, increasing NPC led to increasing the condenser inactive length which increases the operation temperature. Fig. 10 shows the effect of nanoparticles concentration (NPC) on the heat pipe thermal resistance for CuO-nanofluid. This fig. shows that the thermal resistance decreases as the NPC increases. This is due to increasing the thermal conductivity of the working fluid, even with the reverse effect of increasing the condenser inactive length in the presence of more NPC.

The experiments show that the nanoparticles concentration (NPC) has significant effect on the heat pipe thermal performance. This represented by the heat pipe thermal resistance improvement which can be obtained by

$$Improvement\,\% = \frac{R_{th,water} - R_{th,nanofluid}}{R_{th,water}} * 100 \quad (4)$$

The highest thermal resistance improvement obtained for CuO-nanofluid reaches up to 9.5% which accrued at input heat of 25 W and NPC of 5 Vol.% as shown in Fig. 11.

This improvement is not attributed only to the thermophysical properties of the nanofluids but also due to the formation of a thin porous coating layer by nanoparticles on the wick surface in the evaporation region. This confirm experimentally by **[8,9]** and confirmed by **[10,11]**. They indicated that the coating layer formed by nanoparticles improves the surface wettability by reducing the contact angle and increasing the surface roughness which in turn increases the critical heat flux. This improves the maximum heat transport rate and reduces the thermal resistance of the heat pipe using nanofluid.

IV. CONCLUSION

This paper deals with the thermal enhancement of the variable conductance heat pipe performance, using pure water and 1, 3 and 5 Vol.% of CuO-nanofluid as the working fluids. Addition three different amounts (0.5, 1.1 and 1.5 mg) of air as non-condensable gas.

Conclusions may be drawn from the experimental results as follows:

1-Increasing the input heat flux increases the wall temperature and the condenser active length and decreases the thermal resistance.

2-Increasing the mass of non-condensable gas (air) decreases the condenser active and increases the wall temperature and the thermal resistance.

3-Increasing the nanoparticles concentration reduces the wall temperature, the condenser active and the thermal resistance. Whereas, the improving in the heat pipe thermal resistance reaches to 9.5% at (Q_{in} =25 W, Ts=18.3 ^{o}C and NPC=5 Vol.%).

4-Results indicate that the Copper Oxide nanofluid has remarkable potential as working fluid for horizontal heat pipe of higher thermal performances.

V. REFERENCES

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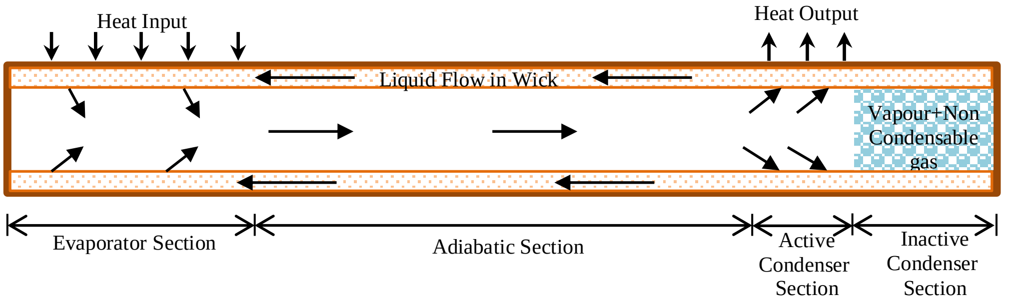
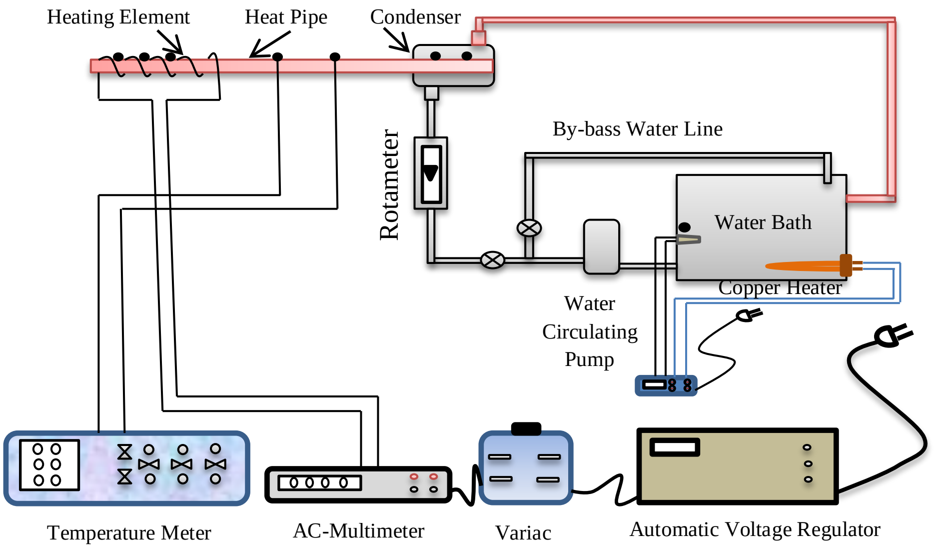
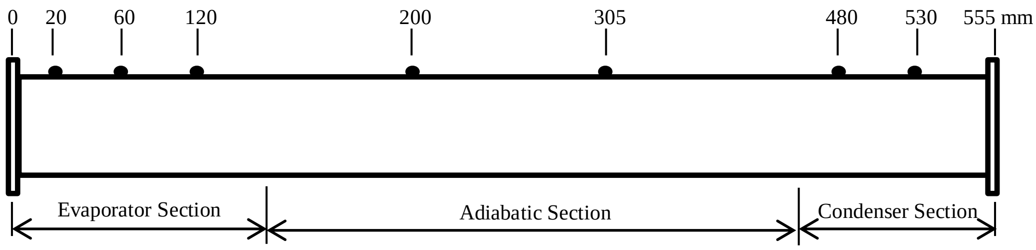


Fig. 1: Schematic of a variable conductance heat pipe.



(a) Experimental test rig



(b) Test section

Fig. 2: Schematics of (a) experimental test rig and (b) test section.

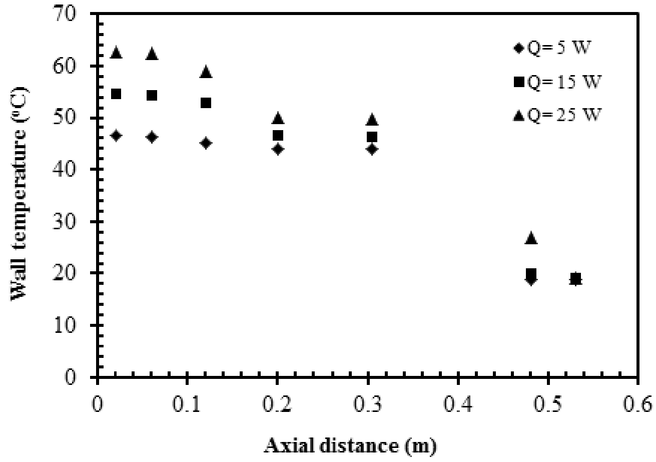


Fig. 3: Effect of input heat flux on heat pipe wall temperature distribution for pure water, coolant temperature of 18.3 °C and air mass of 1.5 mg.

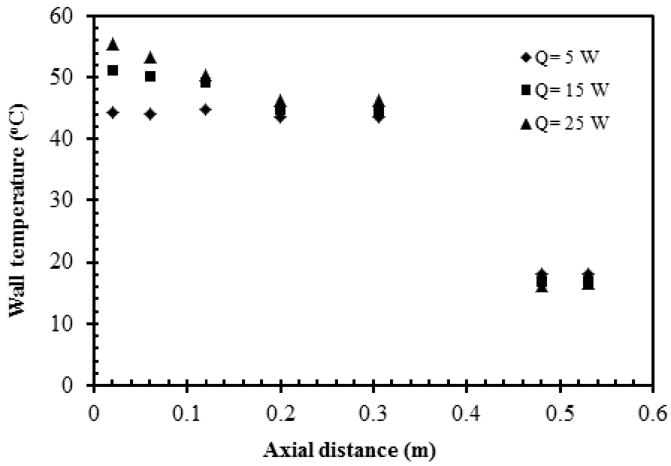


Fig. 4: Effect of input heat flux on heat pipe wall temperature distribution for 3 Vol.% CuO nanofluid, coolant temperature of 18.3 °C and air mass of 1.5 mg.

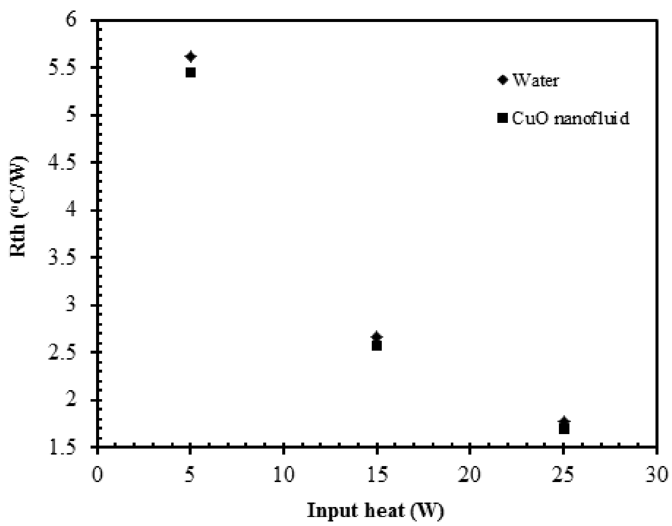


Fig. 5: Effect of input heat flux on heat pipe thermal resistance for pure water and 3 Vol.% CuO nanofluid for coolant temperature of 18.3 °C and air mass of 1.5 mg.

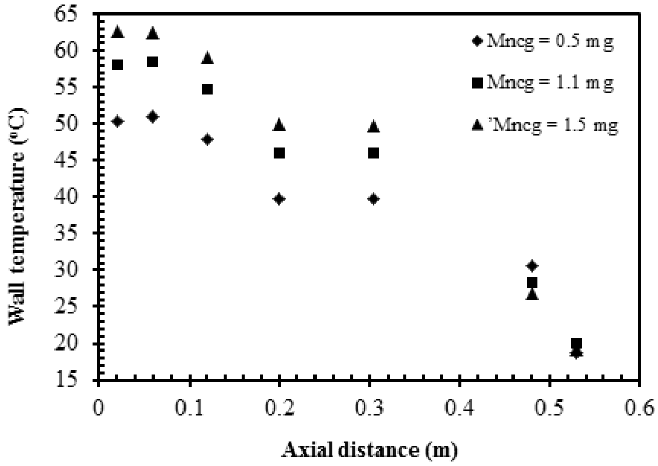


Fig. 6: Effect of mass of non-condensable gas on heat pipe wall temperature distribution for pure water, input heat of 25 W and coolant temperature of 18.3 °C.

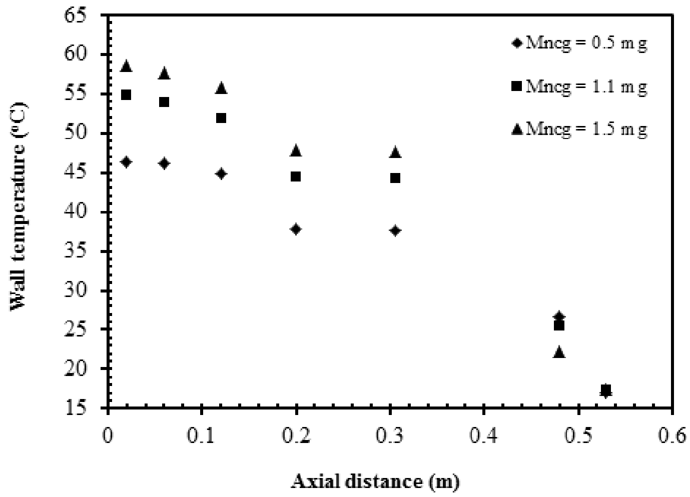


Fig. 7: Effect of mass of non-condensable gas on heat pipe wall temperature distribution for 1 Vol.% CuO- nanofluid, input heat of 25 W and coolant temperature of 18.3 °C.

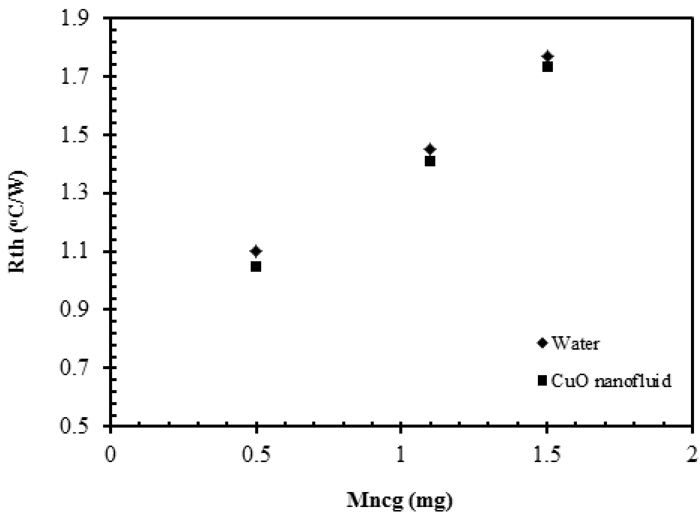


Fig. 8: Effect of mass of non-condensable gas on heat pipe thermal resistance for pure water and 1 Vol.% CuO-nanofluid for input heat of 25 W and coolant temperature of 18.3 °C.

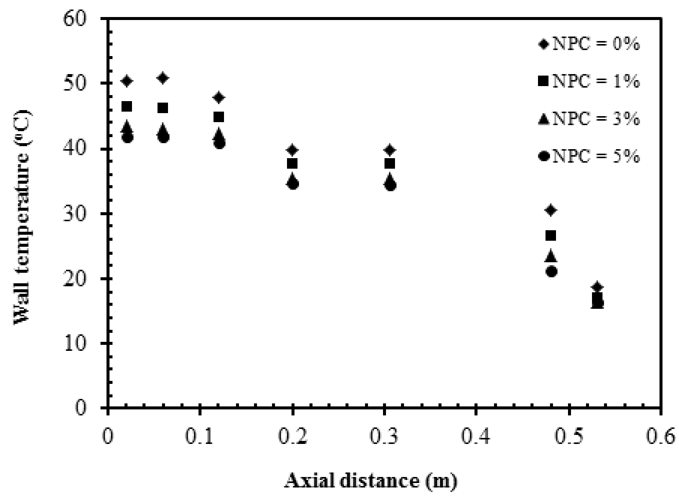


Fig. 9: Effect of nanoparticles concentration (NPC) on heat pipe wall temperature distribution for CuO-nanofluid, input heat of 25 W, coolant temperature of 18.3 ^oC and air mass of 0.5mg.

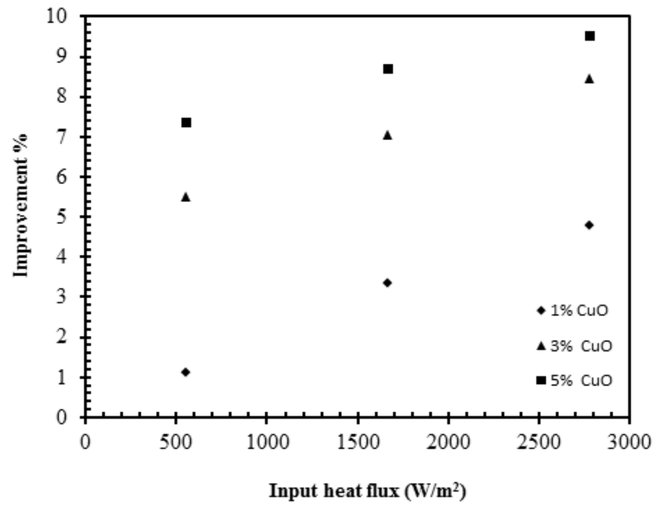


Fig. 11: Improvement of heat pipe thermal resistance, with CuO – nanofluid, for coolant temperature of 18.3 ^oC.

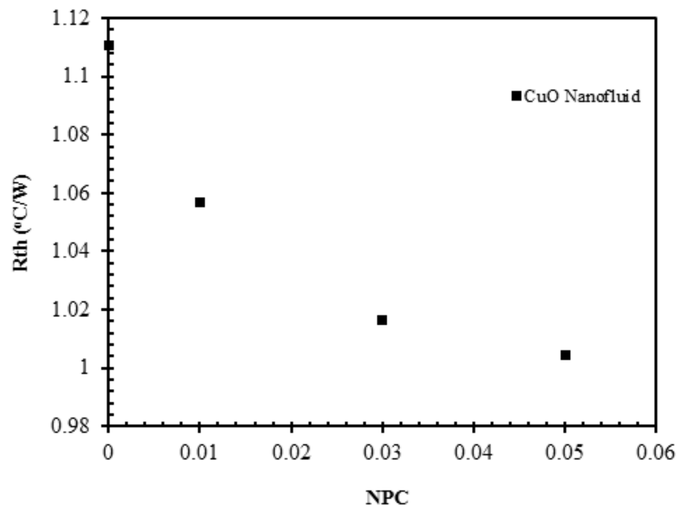


Fig. 10: Effect of nanoparticles concentration (NPC) on heat pipe thermal resistance for CuO-nanofluid for input heat flux of 25 W, coolant temperature of 18.3 ^oC and air mass of 0.5mg.