Comparative analysis of an environmentally friendly refrigerant with DI water when used in a thermosiphon heat pipe

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Abstract. The focus on building better and efficient cooling techniques for electronic gadgets are gaining importance day-by-day. Among the various techniques being studied passive cooling techniques using heat pipes are gaining momentum; the challenge being finding a suitable working fluid which would enhance the performance of these devices. With the present scenario demanding eco-friendly alternatives, the researchers have tried using R600a, a hydrocarbon refrigerant with zero Ozone Depletion Potential. The optimum fill ratio of 50% was determined experimentally at an orientation of 90° from horizontal. Experiments were carried out to find the performance parameters of the thermosiphon heat pipe at the said fill ratio by varying the input heat from 20 W to 140 W with an increment of 20 W each. Next, the working fluid was replaced by de-ionized water and experiments were repeated. The results suggest that the resistance of the heat pipe thermosiphon reduced by 27% when R600a was used. Also, heat transfer coefficients at the evaporator and condenser section increased by a significant margin of 19% and 30% respectively when R600a was used in comparison to DI water.

1. Introduction

Cooling has become an integral part of our lives, be it for human comfort or for increasing the working life of devices. A scaling down in terms of size of devices has contributed to an exponential increase in heat flux, which has a detrimental effect on its life and performance. Advanced, effective and economic methods that are being researched on to expel the heat that is generated, has led to the invention of heat (carrying) pipes. They are hollow evacuated tubes made of metals and partially filled with a fluid. This fluid is continuously made to undergo evaporation and condensation process to take heat away from a source and remove it to a sink without any need for any external power input [1].

Heat pipes may be classified as thermosiphon heat pipes and heat pipes with wicks. Among the two types, the complexities and difficulties in operation of wick used in heat pipes have been overcome by the use of thermosiphon heat pipes [2]. This simple system comes with a configurational disadvantage wherein the condenser always has to be on top during the operation, and not the evaporator.

Ever since its invention, attempts have been made to vary the geometrical parameters, working fluids, materials etc. to suit various applications. Researchers have come up with different working fluids which would positively influence the working of heat pipes [2-14]. Among other working fluids, good thermodynamic properties of refrigerants, especially R134a, has prompted researchers to use it as heat transfer fluid[15]. The downside of using R134a is its high Global Warming Potential [16]. In a work
that was reported, the performance of a thermosiphon heat pipe with porous copper coating charged with R134a showed reduction in resistance and increase in evaporator and condenser heat transfer coefficients by 11% and 25% respectively at 45° inclination of the thermosiphon heat pipe. In another work conducted with HFE7000 and nano-refrigerant HFE7000/Al2O3 on a thermosiphon, the heat transfer coefficient in evaporator was increased by 40.48% for 0.05% concentration of nano-refrigerant [17]. Eco-friendly nature characterized by low GWP and zero Ozone Depletion Potential have led researchers to turn their attention to R600a [18]. The authors have tried to compare the performance of R600a with DI water in thermosiphon heat pipe under different operating parameters.

2. Experimental section
The experimental layout of the test rig has been represented in Figure 1. The setup consists of a 500 W capacity wattmeter, Voltmeter, Ammeter, electric heater element, variable transformer, DAQ Agilent 34972A – data acquisition system, temperature sensors, fibre glass insulation and flow meter. The heat input to the evaporator section was given using the electric heating element. The heat supplied at this end would be absorbed by the working fluid and would be transported to the condenser end of the thermosiphon heat pipe while passing through the adiabatic section in the middle. A fibre glass insulation helped reduce the heat loss around the adiabatic section. The amount of heat transported to the condenser section was determined by passing cooling water around the section and noting the inlet and outlet temperatures. A 300 mm long thermosiphon with 100 mm long evaporator section, 80 mm adiabatic section and 120 mm long condenser section was used for the experimentation purpose. The thermosiphon had a thickness of 1.6 mm and an outer diameter of 12.7 mm. Data Acquisition system was used to record the surface temperatures of the thermosiphon and the inlet and outlet temperature of cooling water, all measured using ‘K’ type thermocouples.

The best fill ratio of a thermosiphon heat pipe is experimentally determined when charged with R600a. The performance of the thermosiphon charged at this best fill ratio is found out under varying operating conditions of heat load and has been compared to the performance of the thermosiphon charged to the same fill ratio with DI water under similar set of conditions. The experiments were run at different input values from 20 W to 140 W. Surface temperatures of various sections and inlet water temperature and outlet water temperatures of the condenser section were recorded. In order to attain steady state, the system was allowed to run for half an hour before readings were noted.

3. Results and discussion
Like heat pipes, thermosiphon heat pipes also helps transport heat through a considerable distance from a source to a desirable sink, even if the temperature difference is not significantly great. Effectiveness of the device may be computed if fraction of heat transported from evaporator end to condenser end can
be measured. This can be found out by recording the difference in temperatures of cooling water between the outlet and inlet. The resistance ($R$) offered by the heat pipe to the flow of heat can be found out using the below mentioned relation:

$$ R = \frac{T_e - T_c}{Q_{out}} $$

Here $R$, $T_e$, $T_c$, $Q_{out}$ corresponds to resistance of thermosiphon, average surface temperature of evaporator, average surface temperature of condenser and heat rejected from the condenser section respectively. Heat rejected may be calculated using the relation:

$$ Q_{out} = m \cdot C_p (T_o - T_i) $$

In the above equation, $C_p$, $m$, $T_o$ and $T_i$ represents to specific heat, mass flow rate, condenser cooling water outlet temperature and inlet temperature. The performance of the thermosiphon heat pipe was evaluated at different fill ratios of R600a starting from 40% to 70% with 10% increments while the thermosiphon heat pipe was aligned vertically. The best fill ratio was identified and other performance parameters were calculated at this fill ratio. Next, the refrigerant was replaced by de-ionized water and tests were repeated for computing performance of the thermosiphon at the best fill ratio corresponding to the refrigerant.

**Figure 2.** Variation of resistance under varying fill ratios

A 40% fill ratio resulted in dry-out condition, which suggests that the refrigerant was under filled. While comparing the resistance values at 20 W of heat input, the 60% fill ratio exhibited an increase in resistance by 9% when compared to 50% fill at the same heat input. Similarly, the resistance increased to 13% at 70% fill ratio at the same heat input. The results suggest that the 60% and 70% fill ratios might have flooded the evaporator with working fluid, which would have affected its working. The experimental results suggest that from among the various fill ratios, the thermosiphon heat pipe performed best when it was filled up to 50% of its volume. Hence, experiments were repeated with de-ionized water at 50% fill ratio. The graphical representation of the resistance at various fills have been plotted in Figure 2.
Figure 3. Resistance offered by the thermosiphon working with different fluids

Figure 3 shows the resistance offered to the flow of heat by the thermosiphon working with R600a as well as DI water at 50% fill ratio for various input heat values. From the general trend, it could be observed that the variation in resistance with the heat input exhibited by thermosiphon heat pipe working with DI water was in line with the trend exhibited by R600a. It was noted that the resistance value came down as heat input increased which complements to the findings of few other researchers [13,15]. The values suggest that the thermosiphon heat pipe exhibits better heat transfer capabilities at greater values of heat inputs.

Other parameters considered to analyse the heat transfer ability of the thermosiphon were condenser and evaporator heat transfer coefficients ($h_c$ and $h_e$), which can be computed by:

$$h_c = \frac{q_c}{T_{ad} - T_c} \quad (3)$$

$$h_e = \frac{q_e}{T_e - T_{ad}} \quad (4)$$

The heat flux at condenser and evaporator sections are denoted by $q_c$ and $q_e$ respectively. $T_e$, $T_{ad}$ and $T_c$ refers to evaporator, adiabatic and condenser sections’ surface temperature respectively.

Figure 4. Variation of condenser heat transfer coefficient
Figure 4 and Figure 5 represent the heat transfer coefficients at source as well as sink sections of thermosiphon for both R600a and de-ionized water. The observations suggest that heat transfer coefficients at sink or condenser section is more than that of the heat transfer coefficient at the evaporator section for both sets of working fluids. The smooth interior of the tube would have aided the easy recirculation of fluid back to the evaporator from condenser. Corrugations or surface roughness at the evaporator end would have enhanced the heat transfer coefficient at the evaporator end. It was also observed that as the heat input was increased, the evaporator heat transfer coefficient increased exponentially, more for R600a than de-ionized water. The shift from ‘drop-wise’ to ‘film-wise’ condensation at greater inputs would justify a gradual decrease of condenser heat transfer coefficient after an initial increase at 40 W. The graphs also suggest that the thermosiphon working with R600a exhibits superior performance in comparison to de-ionized water.

4. Conclusion
The optimum fill of R600a which delivered the best performance corresponded to 50% fill ratio. The mean value of resistance obtained at this fill was 0.352 W/m²K which was than 5% and 9%. The working of the heat pipe was compared when the fluid was replaced by DI water. The results show that the even though the geometric specifications and the experimental conditions remain unchanged, thermosiphon heat pipe performance is remarkably affected by the changes in working fluid. Owing to better heat transfer capabilities, the mean resistance value when refrigerant was used as working fluid was 27% lesser when compared to DI water. The average evaporator heat transfer coefficient was 805.74 W/m²K when R600a was used. This value decreased to 652 W/ m² K when DI water was used marking a reduction of 19%. Similarly, the heat transfer coefficient at the condenser side was 1009.56 W/m²K compared to 704.85 W/m²K when R600a and DI water were used respectively. The heat transfer coefficient also marked a decrease by 30% when DI water was used. This also focuses the strong need for using a working fluid with better thermos-physical properties in order to extract the best performance from the heat transfer device.

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