In the present chapter, firstly, the thermophysical properties of nanofluids are presented and compared the same with base fluid (DI water). Due to the fact that nanofluids possess desirable thermophysical properties especially the significant enhancement in the thermal conductivity, thermal performance of heat pipes with nanofluid as a working fluid are studied. The reason for performance enhancement while using the nanofluid is also discussed. It is found that the deposition of nanoparticles present in the nanofluid has a significant role in performance enhancement of heat pipes. Hence, to confirm the key effects of nanoparticle deposition, thermal performance of heat pipe with and without nanoparticle coated wick is studied.

Though the heat pipe with nanoparticle coated wick performs better than the traditional wicked heat pipe, it is felt that the preparation and coating of nanoparticles in the wick or evaporator wall is costly and time consuming process. Hence a cost effective method to prepare a uniform coating at the inner wall is proposed. An anodizing process is carried out to prepare a coating at the inner wall of the tube. Also, an extensive parametric study is performed to identify the suitable anodizing conditions. After development of coating, performance of both anodized and non anodized thermosyphons are studied. Finally, the effect of nanofluids on the performance of the heat pipe is analyzed numerically. The temperature, operating pressure, vapor and liquid velocities, pressure drop at the liquid/vapor interface and pore size are predicted for both heat pipes with water and nanofluids. Experimental and predicted temperature data are compared and found to be in reasonable agreement.
5.1 Thermophysical properties of working fluids

Measurement techniques and equipments used to measure the thermophysical properties of working fluids are presented in the chapter 2. Thermophysical properties such as viscosity, thermal conductivity, surface tension and density of working fluids are presented as follows in Figures 5.1-5.4:

Figure 5.1 shows the viscosities of Cu-water nanofluids and DI water. There is no significant effect on concentration of nanoparticles on the viscosity. However, a slight increase in the viscosity is observed for nanofluids compared to the base fluid. Also it showed temperature dependent characteristics for both DI water and nanofluid. Liquid viscosity tends to decrease as its temperature increases. For both wt\% (0.01 and 0.1) of copper nanoparticles the prepared nanofluid exhibits almost Arrhenius behavior.

![Figure 5.1 Viscosities of nanofluids](image-url)
Figure 5.2 Thermal conductivity of nanofluids

Figure 5.3 Surface tension of nanofluids
Thermal conductivity of Cu-water nanofluid is determined for different weight percentages of nanofluid. The average particle size of copper particle is 95 nm. Figure 5.2 reveals that the thermal conductivity of the nanofluid increases with the increase in the concentration of Cu nanoparticles in the base fluid. The thermal conductivity is enhanced by 3.79% for 0.01 wt% of Cu and 30% enhancement for 0.1 wt% of Cu particle. Similar trend in the thermal conductivity enhancement was observed in a study of Wang (1999) where CuO particles were dispersed in water and ethylene glycol. Also it was found that the effective thermal conductivity of ethylene glycol increases 26% when approximately 5 volume% of alumina powder was added, and it increases 40% when approximately 8 volume% of alumina powder was added.
The measured surface tension profiles are presented in Figure 5.3. There is no considerable variation in surface tension when metal particles are added to base fluids. Figure 5.4 presents the density variation of nanofluids at different temperatures. The density of the nanofluid is slightly higher than the base fluids. Also, as the temperature increases the density of both nanofluids and base fluid decrease.

5.2 Performance of heat pipe with nanofluids

The wall temperature of heat pipes filled with nanofluid and DI water at different heat inputs are presented in Figure 5.5 (a-d). It is seen that as the heat input increases, the wall temperature of both heat pipes charged with DI water and nanofluids increases. Also it is clearly seen that nanofluid decreases the wall temperature. The wall temperature reduces when the percentage of the copper increases. It is observed that reductions in the wall temperature of the evaporator, adiabatic and condenser are in the range of 3 to 23 °C, 4 to 27 °C and 3 to 6 °C respectively. This wall temperature reductions lead to reduction in resistance or increase in the thermal conductivity of the heat pipes. The temperature is maintained uniformly in the adiabatic region of the heat pipe in all the cases since the diffusion is predominant in the liquid-vapor region of the condenser.

The main reason for reduction in temperature is the presence of a large number of bubble nucleation sites in the evaporator section. Larger sized nucleation sites create larger vapor bubble and offer a high thermal resistance that prevents the transfer of heat from solid surface to liquid. The suspended nanoparticles tend to bombard the vapor bubbles during the bubble formation. Therefore, it is expected that the nucleation size of vapor bubble is much smaller for a fluid with suspended nanoparticles than that without them (Tsai, 2004). In order to analyze the nanoparticles deposition on the wick surface, wick samples from the evaporator
section is obtained by cutting and open the heat pipe after the experiments. The surface condition of the wick surface is then observed by SEM (JEOL make). The observed SEM image (Figure 5.6) shows the nanoparticles deposition on the wick surface forming a microporous structure/layer on the evaporator wall. The presence of the microporous layer can help to delay the critical heat flux in two ways. First, it increases the wettability and promotes rewetting upon bubble departure. Second, the layer may assist in dissipating the heat from hot spots by enhancing radial conduction on the surface. Of course, the magnitude of the radial conduction is small in the thin layer of the microporous structure. This reduces the wall temperature of the heat pipe.

Figure 5.5 Thermal performance of the heat pipe (a) 100 W, (b) 150 W, (c) 200 W and (d) 250 W
Figure 5.7 shows the thermal efficiency of the heat pipe charged with nanofluid and DI water. It is interesting to note that the efficiency is improved by increasing the concentration of Cu in a Cu-water nanofluid. The efficiency is also increased by 10-14% with the copper nanofluid filled heat pipe as compared with the base fluid filled heat pipe. There is also 5% increase in the efficiency of the heat pipe charged with the Cu nanofluid when the wt% of the Cu is increased from 0.01 to 0.1

The variation of thermal resistances of the heat pipes filled with DI water and the Cu nanoparticles suspensions are presented in Figure 5.8 for different heat inputs. The thermal resistance of heat pipes decreases with the increase of heat input. For 0.1 wt% of Cu nanoparticles, the thermal resistance is reduced by 33%. The present results are supported by Riehl (2012). The reduction in thermal resistances is due to the activation of larger number of nucleation sites in the evaporator section which extends the regime of nucleate boiling to very high heat fluxes.

Figure 5.6 SEM view of the particle deposition
Figure 5.7 Thermal efficiency of the heat pipe

Figure 5.8 Thermal resistance of heat pipe
In order to show the effect of thermophysical properties of nanofluids and coating layer on the performance of heat pipe, the overall heat transfer coefficient is calculated. The overall heat transfer coefficient can be calculated by Equation 5.1 with the thermal resistances

$$U_p = \left[ R_{p,e} + R_{w,e} + R_v + R_{w,c} + R_{p,c} \right]^{-1}$$  \hspace{1cm} (5.1)

where the thermal resistances are determined using expressions given in Table 5.1. The thermal properties of the base fluid with nanoparticles suspension are taken at saturation temperatures and the overall heat transfer coefficient is calculated. The saturation temperature of the working fluid is assumed to be the temperature at the adiabatic section.

Table 5.1 Expressions for thermal resistances

<table>
<thead>
<tr>
<th>Expression</th>
<th>Formula</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal resistance at the evaporator wall</td>
<td>$R_{p,e} = \frac{r_0 t_w}{2L_e k_w}$</td>
</tr>
<tr>
<td>Thermal resistance at the liquid-wick combination in the evaporator</td>
<td>$R_{w,e} = \frac{r_0^2 t_\omega}{2L_e k_{eff} l_i}$</td>
</tr>
<tr>
<td>Effective thermal conductivity of the liquid-wick combination</td>
<td>$k_{eff} = \frac{k_1[(k_l + k_\omega) - (1 - \varepsilon)(k_l - k_\omega)]}{[(k_l + k_\omega) + (1 - \varepsilon)(k_l - k_\omega)]}$</td>
</tr>
<tr>
<td>Thermal resistance due to the vapor flow</td>
<td>$R_v = \frac{\pi r_o^2 T_v F_v}{p_v k} \left( \frac{1}{6} L_e + L_a + \frac{1}{6} L_e \right)$</td>
</tr>
<tr>
<td>Thermal resistance at the condenser wall</td>
<td>$R_{p,c} = \frac{r_0 t_w}{2L_c k_w}$</td>
</tr>
<tr>
<td>Thermal resistance at the liquid-wick combination in the condenser</td>
<td>$R_{w,c} = \frac{r_0^2 t_\omega}{2L_c k_{eff} l_i}$</td>
</tr>
</tbody>
</table>

The overall heat transfer coefficient of the heat pipe is calculated based on the lumped thermal resistance network shown in Figure 5.9. Figure 5.10 shows the
enhancement of overall heat transfer coefficient of the heat pipe. The overall heat transfer coefficient of the heat pipe filled with nanofluid is increased to a maximum of 6300 W/m\(^2\)-K. The overall heat transfer coefficient of the heat pipe increases with the increase in the saturation temperature or the critical heat flux. This enhancement is mainly due to the reduction in the thermal resistance of the liquid layer. There are three phenomena which may be responsible for their enhancement: an increase in the effective thermal conductivity, the decrease in solid-liquid contact between the nanofluid and the heated surface, and the random motion and collision of nanoparticles in the base fluid (Brownian motion). Kabelac and Kuhnke (2006) observed that the Brownian motion of the nanoparticles is not a dominant mechanism of the enhanced thermal conductivities. The variation of thermal resistance may occur not only by variation of thermophysical properties but also by surface modification caused during the boiling process. It is attributed to the coating layer formed on the heated surface during the heating process. The surface structure formed by nanoparticles ensures good capillary structure. A similar observation is found in a study of Liu (2007). Hence, it is understood from Figure 5.8 and Figure 5.10 that the surface structure formed by nanoparticles has more influence than the thermophysical properties on the performance of heat pipe.

![Lumped thermal resistance network](image)

Figure 5.9 Lumped thermal resistance network
5.3 Performance of heat pipe with nanoparticle coated wick

To show the effect of coating on the wick on performance of the heat pipe, the thermal resistance at the evaporator is calculated for coated and uncoated wicks using Equation 5.2.

$$R_e = \frac{\ln\left(d_{o,w}/d_{i,w}\right)}{2\pi l_e k_{wall}} + \frac{\ln\left(d_{o,\omega}/d_{i,\omega}\right)}{2\pi l_e k_{wick}} \quad (5.2)$$

The effective thermal conductivity of liquid-wick combination is calculated as

$$k_{wick} = \frac{k_i[(k_i+k_\infty)-(1-\varepsilon_n)(k_i-k_\infty)]}{[(k_i+k_\infty)+(1-\varepsilon_n)(k_i-k_\infty)]} \quad (5.3)$$

The porosity of the multiple layer of screen mesh wick ($\varepsilon_n$) is calculated using Equation 5.4

$$\varepsilon_n = 1 - \frac{n\delta_1(1-\varepsilon_1)}{\delta_n} \quad (5.4)$$

The porosity ($\varepsilon_1$) of a single layer of wick is calculated by

Figure 5.10 Overall heat transfer coefficient of heat pipes charged with DI water and nanofluids
In the case of coated wick, the increase in wire diameter of the screen mesh is measured using SEM and included in the Equation 5.5.

The calculated thermal resistance at the evaporator section of the heat pipe at different heat inputs is shown in Figure 5.11 for coated and uncoated wicks. It is seen that the resistance at the evaporator section of the heat pipe with coated wick is lower than that of the same with uncoated wick.

Figure 5.11 Resistance at the evaporator

Figure 5.12 (a) – (c) shows the variation of wall temperature along the length of the heat pipe at different heat inputs. It is observed that the wall temperature increases with increasing heat input in the heat pipe operated with both coated and uncoated wicks. The average wall temperatures of the evaporator are 69.5, 81.6 and
90.5 °C at 100, 150 and 200 W respectively in the case of heat pipe operated with uncoated wick whereas in the heat pipe operated with coated wick, the same are 64.9, 76.4 and 89.0 °C respectively at 100, 150 and 200 W. The wall temperature at the evaporator section of heat pipe operated with coated wick is lower than that of heat pipe operated with uncoated wick. Since deposited nanoparticles improve surface areas of the wick, the heat transfer from the wall of the evaporator to the working fluid is increased. Hence, temperature at the evaporator section of the heat pipe is decreased.

The average wall temperatures of the condenser are 39.6, 44.4 and 48.4 °C respectively at 100, 150 and 200 W in the case of heat pipe operated with uncoated wick whereas in the heat pipe operated with coated wick, the same are 36.7, 35.2 and 46.1 °C respectively at 100, 150 and 200 W. As expected, similar to the observation at the evaporator section, the wall temperature at the condenser section of heat pipe operated with coated wick is lower than that of heat pipe operated with uncoated wick. Unlike the temperature variation at evaporator and condenser sections, the same at the adiabatic section is higher in heat pipe operated with coated wick. An increase in thickness of the wick due to the coating is likely to increase the heat transfer to outside and hence could increase the surface temperature of the adiabatic section marginally.

Figure 5.13 (a) shows the thermal resistance in the evaporator at different heat inputs. Thermal resistance \( R_e \) decreases with the increasing heat input in the heat pipe operated with uncoated wick whereas the effect of heat input on the thermal resistance \( R_e \) of the heat pipe operated with coated wick is negligible. The reduction in \( R_e \) of the heat pipe operated with coated wick is 40%, 37% and 32% at 100 W, 150 W and 200 W respectively when compared with \( R_e \) of the heat pipe operated with uncoated wick. The main reason for reduction in thermal resistance \( R_e \) is due to the large number of smaller nucleation sites created at the evaporator
of the heat pipe. Smaller nucleation sites produce smaller bubbles and increase the heat transfer between the evaporator wall to working fluid. A larger size bubble nucleation creates a higher thermal resistance that prevents the transfer of heat from the wall of the evaporator to the working fluid (Collier 1996).

Figure 5.13 (b) shows the thermal resistance in the condenser section at different heat inputs. Thermal resistance in the condenser section of heat pipe operated with coated wick is higher than that of the conventional heat pipe. Though it is higher than that of conventional one, the larger reduction in the thermal resistance at the evaporator of the same lowers the total resistance as shown in the Figure 5.14. It is interesting to note that the thermal resistance of the heat pipe operated with nanoparticle coated wick is comparable with that of heat pipe operated with Cu-water nanofluid prepared from the same sized particles at lower heat inputs. However, at the heat input of 200 W, thermal resistance of the heat pipe operated with coated wick is higher than that of heat pipe operated with nanofluid.

The calculated evaporator and condenser heat transfer coefficients are shown in Figure 5.15 (a) and (b) respectively. It is seen from Figure 5.15 (a) that the heat transfer coefficient at the evaporator of the heat pipe operated with coated wick is higher than that of conventional one and it decreases with increasing heat input. Though the heat transfer coefficient at the evaporator is higher in the heat pipe operated with coated wick, the same is lower at the condenser when compare to conventional one as seen in Figure 5.15 (b). It may be noted that heat transfer at evaporator takes place due to nucleate boiling which is the result of bubble formation at the nucleation sites. This is the result of low contact angle resulting from a large number of nucleation sites of a rough surface. On the other hand, it is well known that condensation is favored by a smoother surface with high contact angle. However, in the present case, the coating in the condenser surface is rough leading to reduced heat transfer coefficients.
Figure 5.12 Wall temperature along the length of the heat pipe (a) 100 W (b) 150 W and (c) 200W
A decrease in both thermal resistance in the evaporator, total resistance in the heat pipe and the enhancement in heat transfer coefficient at the evaporator clearly indicates that the thermal performance of the heat pipe is enhanced by deposition of nanoparticles over the wick.

Though the thermal performance of heat pipe is improved by deposition of nanoparticles over the wick, it is important that the surface of the wick should sustain the built-in porous structure during the operation. Figure 5.16 (a) and (b) show SEM photographs of coated wick after twenty hours of operation. It can be noticed that the porous structure formed by deposition of nanoparticles is retained on the surface of the coated wick. It reveals that the use of nanoparticle coated wick in the heat pipe is feasible.
Figure 5.14 Total resistance of heat pipes

(a)
Figure 5.15 Heat transfer coefficient of (a) evaporator and (b) condenser

Figure 5.16 SEM photographs of coated wick after twenty hours of operation (a) 430 X and (b) 27000 X
5.4 Performance of anodized thermosyphon

5.4.1 Characterization of anodized aluminium tube for heat transfer application

The main process parameters involved in anodizing process are:-

- Electrolytic flow rate
- Acid concentration
- Final cell voltage
- Electrolysis time

A number of trials are carried out to fix these parameters to obtain a uniform oxide layer thickness with minimum contact angle. The details are shown in Table 5.2. An analysis on the effect of these parameters on the oxide layer thickness and contact angle is given below:-

5.4.1.1 Effect of electrolyte flow rate

Electrolyte flow rate is an important parameter from the point of view of maintaining uniform anodizing throughout the height of the tube and efficient removal of heat and gas generated during the anodization process. At very low electrolyte flow rates (Exp.1, 2 in Table 5.2), the oxide layer contains only pores of smaller size (Figure 5.17 (a)). Very small pores are noticed in the lower region (Figure 5.18 (a)) of the aluminium tube whereas much larger pits are noticed on the top region (Figure 5.18 (b)). Further, the oxide layer thickness observed at the bottom (Figure 5.18 (c)) and top (Figure 5.18 (d)) portions of the tube is found to be different. The thickness of the oxide layer is around 9 µm in the bottom portion and 14 µm in the top portion. Lower flow rates lead to much stronger dissolution at the bottom of the tube. This is because the fresh electrolyte entering this region contains very little dissolved aluminium due to anodic dissolution. The dissolved aluminium
being carried by the electrolyte from the bottom region ensures much less anodic dissolution and hence much thicker oxide layer in the top region. The same reason is also applicable to surface morphology of the oxide layer.

Table 5.2 Details of the anodization experiments

<table>
<thead>
<tr>
<th>Exp. number</th>
<th>Electrolyte flow rate (ml/min)</th>
<th>Final cell voltage ($V_{final}$)</th>
<th>Acid concentration (vol %)</th>
<th>Anodizing time (minutes)</th>
<th>Average coating thickness ($\mu$m)</th>
<th>Average pore size ($\mu$m)</th>
<th>Average contact angle ($\theta$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>uncoated</td>
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<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>110</td>
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<tr>
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<td>10</td>
<td>30</td>
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<td>63.7</td>
</tr>
<tr>
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<td>53.7</td>
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<td>9.0</td>
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<td>90</td>
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<tr>
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<td>10</td>
<td>120</td>
<td>35.3</td>
<td>11.0</td>
<td>&lt;10</td>
</tr>
</tbody>
</table>
At flow rates above the 380 ml min$^{-1}$ (Exp. 3, 4 in Table 5.2), it is found that both surface morphology and thickness of the oxide layer are almost uniform and same respectively (Figure 5.19 (a-d)). This is due to the uniform dissolution and oxide layer growth process throughout the height of the tube. The thickness of the
oxide layer throughout the tube is around 12 μm (Figure 5.19 (c, d)). At much higher flow rate (840 ml min\(^{-1}\)), the oxide layer thickness is decreased to 5.9 μm throughout the tube due to the higher dissolution. However a uniform morphology without pores was observed. Hence electrolyte flow rate is chosen as 380 ml min\(^{-1}\) for further experiments. The contact angle and oxide layer thickness obtained at different electrolyte flow rates are shown in Figure 5.20 (a). It is noticed that electrolytic flow rate has negligible effect on contact angle.

### 5.4.1.2 Effect of sulfuric acid concentration

The variations of contact angle and thickness of the oxide layer with sulfuric acid concentration are shown in Figure 5.20 (b). At lower sulfuric acid concentration (5%), the thickness of the oxide layer formed is around 10 μm. However the contact angle is much higher (101°), which leads to poor wettability. Thus 5% sulfuric acid concentration is not sufficient for the formation of thicker film with higher wettability. The results obtained with 10% sulfuric acid concentration (Exp. 3 in Table 5.2) is similar to that obtained at 15% (Exp.7 in Table 5.2) and better than that obtained at 5% (Exp. 6 in Table 5.2). Hence sulfuric acid concentration of 10% was selected for further studies.

### 5.4.1.3 Effect of anodizing time

Keeping the electrolyte flow rate of 380 ml min\(^{-1}\) and acid concentration of 10%, the electrolysis time is varied between 20 minutes to 50 minutes. With the increasing electrolysis time, the oxide layer thickness is found to increase approximately from 4.3 μm to 14 μm (Figure 5.20 (c)). However the contact angle of the oxide layer is around 63 ± 3°, which indicates that anodizing time does not improve the wettability significantly. The surface morphology, which shows both micro and nano pores, obtained at anodizing time of 50 minutes is shown in Figure
5.17(c). It is noticed that the pore size increases slightly with increasing anodization time.

![SEM view at the bottom](image1)

![SEM view at the top](image2)

Figure 5.18 Surface of the coating (a) at the bottom of the pipe and (b) at the top of the pipe, and cross sectional view (c) at the bottom of the pipe and (d) at the top of the pipe at constant electrolyte flow rate of 28 ml min\(^{-1}\)

5.4.1.4 Effect of final cell voltage

The final anodization cell voltage (\(V_{\text{final}}\)) is found to be an important parameter to determine the oxide layer thickness and contact angle. Hence a number of experiments are carried out by varying \(V_{\text{final}}\) from 10 V to 20 V (Exp. 10-15 in Table 5.2). With increasing cell voltage, the measured contact angle is found to
decrease from 70° to 23°. The film thickness increases with increasing final cell voltage as long as $V_{\text{final}}$ is less than or equal to 19V (Figure 5.20 (d)). At the final cell voltage of 20 V, the film thickness achieved at $V_{\text{final}}$ of 19 V decreases to 18.5 μm from 23 μm (Figure 5.20 (d) and Table 5.2). The pore depth at $V_{\text{final}}$ of 20 V (Figure 5.17 (d)) is also found to be lower than that achieved at the $V_{\text{final}}$ of 19 V (Figure 5.17 (e, f)). Hence it appears that under the present experimental conditions, the oxide layer dissolution rate at the film electrolyte interface exceeds the film growth rate at $V_{\text{final}}$ of 19 V.

Figure 5.19 Surface of the coating (a) at the bottom of the pipe and (b) at the top of the pipe and cross sectional view (c) at the bottom of the pipe and (d) at the top of the pipe at constant electrolyte flow rate of 560 ml min$^{-1}$.
Figure 5.20 Effects of (a) Electrolyte flow rate, (b) Electrolyte concentration, (c) Anodizing time and (d) Applied voltage on contact angle and thickness of the coating.

Hence further experiments are carried out by maintaining the $V_{\text{final}}$ at 19 V and increasing the anodization time. Though this approach provided maximum film thickness (>35 µm), the contact angle is found to be < 10°. SEM micrographs (Figure 5.17 (e, f)) show that the average pore size is greater than 11 µm (Exp. 16, 17 in Table 5.2). Further investigations are also made to understand the effects of $V_{\text{final}}$ on the anodization process.
The anodization current systematically increases with increasing cell voltage as shown in Figure 5.21. Once $V_{\text{final}}$ is reached, the anodization current density also attains a steady value as long as $V_{\text{final}}$ is below 17 V (Figure 5.21). The current density increases with time when $V_{\text{final}}$ is set at 18 V and 19 V. At 19 V the current increases sharply at higher anodizing time when compared to 18 V. It appears that above the $V_{\text{final}}$ of 18 V, the dissolution through pores of the oxide layer increases sharply and leads to the formation of broader pores with much thicker anodized layer. It is also found that at the $V_{\text{final}}$ of 20 V, the anodizing current fluctuates between wide limits. Hence reproducible current-voltage curve could not be recorded in this case.

Figure 5.21 Effect of anodization cell voltage on average anodization current density with time

87
Figure 5.22 Surface roughness of oxide layer prepared at (a) 18 V and (b) 19 V
AFM images obtained after anodization at 18 V and 19 V under otherwise identical experimental conditions (Exp. 13, 14 in Table 5.2) are presented in Figure 5.22. The AFM image (Figure 5.22 (b)) shows that the oxide layer obtained after anodizing at 19 V is found to be more rough with less number of pits when compared to the sample anodized at 18 V (Figure 5.22 (a)), suggesting pore dissolution and film growth in this region.

5.4.2 Thermal stability of the anodized oxide layer

To study the stability of the oxide layer formed on the inner wall of the heat pipe, two samples were tested in an oven at a temperature of 300 °C which is higher than the normal operating temperature of the heat pipes (150 – 200 °C). SEM micrographs (Figure 5.23 (a, b)) obtained after this thermal treatment do not show any significant change in film morphology. No cracks or pits generation is noticed on the oxide layer, which confirms the thermal stability of the oxide layer.

Figure 5.23 Surface of the oxide layer (a) before heating and (b) after 6 hours of heating at 300 °C and cross sectional view of the oxide layer (c) before heating and (d) after 6 hours of heating at 300 °C
5.4.3 Performance analysis

It is known that thermal resistance due to the evaporation/boiling at the evaporator is influenced by the nature of inner surface condition of the TPCT. Generally the thermal resistance due to the same at the evaporator can be expressed as

\[ R_e = \frac{1}{hA_e} \]  \hspace{1cm} (5.6)

Considering that nucleate boiling prevails at the evaporator section of the TPCT (Imura 1983), the thermal resistance due to the boiling at the evaporator can also be expressed by Mikic–Rohsenow correlation (1969):

\[ R_e = \frac{1}{hA_e} = \frac{1}{\frac{1}{2}NaD_b^2 A_e \sqrt{f} \sqrt{n} \sqrt{\frac{\rho l c}{k_l c_l}}} \]  \hspace{1cm} (5.7)

According to the Equation 5.7, nucleation site density, bubble diameter, evaporator area and the bubble frequency are the important parameters that influence evaporator resistance. Figure 5.24 shows the scanning electron microscope (SEM) images of the non anodized and anodized inner wall surfaces of the TPCT. The density of the micro cavity of non anodized and the anodized surfaces are estimated by counting the number of pores present per micrometer area and deriving the total evaporator area. It is observed that the total micro cavities present in the evaporator of the anodized pipe is around 78000 whereas the same in the non anodized evaporator is around 15000. So, the nucleation sites present in the anodized surface is at least 2-3 times higher than that of the non anodized surface. It is also known that activation of the nucleation sites increases with increasing heat flux. This will dramatically reduce the incipient boiling wall superheat of the anodized surface and enhance the heat transfer (Lee, 2010). Also the heat transfer enhancement in the anodized (nano-porous) layer is due to the combined effects of
the interaction between active nucleation sites, the increase of bubble generation frequency, the intensification of bubble interaction and the thin-film evaporation effect (Li, 2007). Table 5.2 (Exp. Uncoated and 9) shows the contact angles for anodized (62.9°) and non-anodized (110°) surfaces. Fritz (1935) showed that bubble size increases with increasing contact angle. Though bubbles with smaller size generated from the anodized surface leads to reduction of thermal resistance according to the Equation 5.7, the large increase in active nucleation sites as well as an enhancement in bubble release frequency and area of the evaporator overcome the negative effect of bubble size.

Figure 5.25 (a) – (c) shows the variation of wall temperature along the height of the TPCT at different heat inputs. It is observed that the wall temperature increases with the increasing heat input in the TPCT with and without anodized surface. It is interesting to note that the evaporator temperature of the TPCT with anodized surface is lower than that of TPCT with non-anodized surface in all heat inputs. Further, the difference between wall temperatures of the TPCT with and without anodized surfaces increases with increasing heat input. Also it is seen that the evaporator temperature profile for the non-anodized case is not uniform which indicates changing boiling dynamics which are not as effective as the anodized case. The anodized surface has a uniform temperature distribution indicating uniform boiling (probably because of the smaller bubbles) and a lower temperature which indicates that this mode of boiling is more effective (probably due to the increased site density). The present results are qualitatively supported by previous results (Rahimi 2010). Though a significant change in the wall temperature at the evaporator is noticed in the anodized TPCT, the adiabatic and condenser wall temperatures of both anodized and non-anodized TPCTs show negligible differences in all heat inputs.
Figure 5.24 SEM images of (a) non anodized and (b) anodized surfaces
Figure 5.25 Wall temperatures along the height of the TPCT at (a) 100 W (b) 150 W and (c) 200 W
Figure 5.26 (a) shows the thermal resistance of the evaporator of both anodized and non anodized TPCT at different heat inputs. Thermal resistance of the evaporator \( R_e \) decreases almost linearly with increasing heat input in both TPCTs. The noticed reduction in \( R_e \) of the anodized TPCT is from 1.5% to 15% for the heat input range from 50 W to 250 W compared with \( R_e \) of non anodized TPCT. The main reason for reduction in thermal resistance \( (R_e) \) is probably due to the large number of smaller nucleation sites created at the evaporator of the heat pipe.

Figure 5.26 (b) shows the thermal resistance of the condenser section of the anodized and non anodized TPCT at different heat inputs. The difference between condenser resistances of both TPCTs is negligible. In general, the condenser is considered longer than the evaporator for better heat transport. This makes the surface area of the condenser larger than that of the evaporator. Hence the heat flux at the condenser is lower than that of evaporator for a given heat input. Figure 5.27 (a) shows the heat transfer coefficient at the evaporator section of the anodized and non anodized TPCT at different heat inputs. The heat transfer coefficients of the evaporator of both TPCTs increase almost linearly with the increasing heat input. However the heat transfer coefficient of the evaporator of the anodized TPCT is higher than that of the non anodized TPCT and the difference between the same of evaporator of both TPCTs increases with increasing heat input. This may be due to the change in boiling dynamics and the mode of heat transfer which takes place at the evaporator. The mode of heat transfer in the evaporator depends on the vapor pressure and heat flux in the TPCTs (Kempers, 2008). The heat transfer in the evaporator is dominated by the conduction at the lower heat fluxes since less number of nucleation sites are activated. It is dominated by nucleate boiling at higher heat fluxes due to the large number of activated nucleation sites. This can be explained by using Mikic–Rohsenow correlation (1969) given as
\[ N_a = \frac{q}{\pi k \rho c_l} \sqrt{\frac{\Delta T}{\frac{1}{2} \sqrt{D_b^2 \Delta T / \pi k \rho c_l}}} \quad (5.8) \]

The enhancement in heat transfer coefficient of the anodized TPCT is from 1.5% to 15.7% for the heat input range from 50 W to 250 W compared with the same non anodized TPCT. Though the heat transfer coefficient of the evaporator is enhanced in the anodized TPCT, the same for the condenser of both TPCTs show no significant difference between them (Figure 5.27 (b)). It is noticed that the heat transfer coefficient of the condenser increases with the increasing the heat flux. This is because the coolant flow rate at the condenser is adjusted to maintain the same \( \Delta T \) for all heat inputs.

The effect of inclination angle on the performance of anodized and non anodized TPCTs is shown in Figure 5.28. The variations of temperature difference (\( \Delta T \)) between the evaporator wall and vapor at various heat inputs for different inclination angles (90\(^\circ\), 45\(^\circ\), 0\(^\circ\)) are presented. As expected, \( \Delta T \) increases linearly with increasing heat input both in anodized and non anodized TPCT. The \( \Delta T \), both in anodized and non anodized TPCT, decreases with increasing inclination angle from 0 to 90\(^\circ\). This is because the condensate inventory in the evaporator is dominated by gravity. In all cases, \( \Delta T \)s in the anodized TPCT are lower than that of the non anodized TPCT. This is due to the enhancement in the number of nucleation sites, which support nucleate boiling at the evaporator, on the anodized TPCT as explained earlier. The heat transfer characteristics of the anodized and non anodized TPCT are shown in Figure 5.29. In all cases, heat loss increases with increasing heat input. Among the selected inclination angles, TPCT operated under vertical mode shows better heat transport both in anodized and non anodized cases. It is seen that the anodized TPCTs transport more heat than the non anodized TPCTs at all inclination angles and at higher heat inputs.
Figure 5.26 Thermal resistances of the (a) evaporator and (b) condenser of the anodized and non anodized TPCT
Figure 5.27 Heat transfer coefficients of (a) evaporator and (b) condenser sections of the anodized and non-anodized TPCT
Figure 5.30 shows the total thermal resistance of the anodized and non-anodized TPCT. Total thermal resistance decreases exponentially with increasing the heat input in both TPCTs. However, the same for anodized TPCT is lower than that of the non-anodized TPCT, and the difference between the resistances of both TPCTs increases with increasing heat input. Though the condenser resistance of both TPCTs is almost the same, the larger reduction in the thermal resistance of the evaporator lowers the total resistance of the anodized TPCT.

Though thermal performance of heat pipe is improved by anodization of the inner wall of the TPCT, it is important that the surface of the wall should sustain the porous structure during the operation. Figure 5.31 shows SEM photographs of anodized wall after 35 hours of operation. It can be noticed that the surface of the TPCT wall retains the porous structure formed by anodization. From this study it can be concluded that the use of anodized porous coating in the TPCT is feasible and advantageous.

![Figure 5.28 Temperature difference between the evaporator and the vapor at various heat inputs for TPCTs operated under different inclination angles](image)

Figure 5.28 Temperature difference between the evaporator and the vapor at various heat inputs for TPCTs operated under different inclination angles
Figure 5.29 Heat transfer characteristics of anodized and non-anodized TPCTs at different inclination angles and heat inputs.

Figure 5.30 Total thermal resistance of the anodized and non-anodized TPCT.
5.5 Numerical results

In order to validate the present model, a flat heat pipe (Vadakkan, 2004) is simulated and compared with the results of present and previous models. The transient evaporator wall temperature, transient system pressure and the steady state wall temperature of the flat heat pipe predicted using present model are shown in Figures 5.32-5.34. The predicted results are comparable with the results presented in the literature (Vadakkan, 2004).

A heat pipe shown in Figure 4.1 is simulated for input powers of 250 W and 350 W. The measured heat transfer coefficients applied in the condenser section are 1070 and 1872 W/m²·K for 250 and 350 W respectively. The initial temperature and initial pressure of the system is assumed as 297.9 K and 3778 Pa respectively. The $k_{\text{eff}}$ of the wick is calculated by the Equation 4.5 and the same is 1.11 W/m·K for DI water. The $k_{\text{eff}}$ of the wick is 1.69 W/m·K when the nanofluid is used without considering the deposition of nanoparticle. $k_{\text{eff}}$ value of 1.75 is used for the nanofluid when particle deposition on the wick is considered.
Figure 5.32 Transient wall temperature at the center of the evaporator section of flat heat pipe at the power input of 30 W

Figure 5.33 Transient variation of system pressure in the vapor core for the 30W heat input
To establish a suitable time step for unsteady simulation, varieties of tests are performed with five different time steps varying between 0.2 and 4 seconds. As a sample, the dependence of the evaporator temperature on the time step is shown in Figure 5.35. Though, the variation in temperatures is negligible for the time steps between 0.2 and 4 seconds, a time step of 1 second is considered in the present study. A typical temperature contour of the heat pipe simulated at 250 and 350 W for water and nanofluids are shown in Figure 5.36 It is seen that the evaporator temperature of the heat pipe is found to be higher than that of the adiabatic and condenser sections for the same heat input. It is also observed that the temperature gradient decreases along the axial distance of the heat pipe leading towards the condenser section. Further it is noted that the temperature distribution of the heat pipe filled with nanofluid (Figure 5.36 (b, d)) is lower than that of the heat pipe filled with DI water (Figure 5.36 (a, c). Further, a strong temperature gradient in the wick region is observed. Due to the addition of nanoparticles, the temperature
gradient is flattened throughout the length of the heat pipe; and thereby enhancing the heat transfer capability of the heat pipe when a nanofluid is used as a working fluid.

![Figure 5.35 Influence of time step on the evaporator wall temperature](image)

The predicted temperature of the heat pipe charged with DI water, nanofluid with and without considering the particle deposition as well as measured temperature of the heat pipe charged with DI water is shown in Figure 5.37 and 5.38 for 250 and 350 W respectively. From these results, it is clearly seen that the predicted results agree with the experimental results. A maximum temperature deviation between the predicted and measured is found to be ± 0.4% and ± 1.5% at the evaporator and condenser sections respectively. The higher deviation in the condenser section is caused due to the non uniform cooling effect present in the condenser section. The reason for this behavior is attributed to a constant heat
transfer coefficient which is applied at condenser section in the model. However in experiment, the heat transfer coefficient along the condenser section will vary. After validating the present model, the thermophysical properties of nanofluids are incorporated into model and the effect nanofluids on the performance of the heat pipe is analyzed. The Figures 5.37 and 5.38 show the wall temperatures of heat pipe charged with Cu-water nanofluids. The results show that the wall temperature of the heat pipe charged with Cu-water nanofluids is lower than that of heat pipe charged with the DI water. It is also observed that the average evaporator wall temperature of the heat pipe is decreased by 5% and 8% for 250 and 350 W respectively when compared with that of the heat pipe with considering the coating effect.

The thermal resistance of the heat pipe is calculated as \( R = \frac{\Delta T}{Q} \), where, \( \Delta T \) is the temperature difference between evaporator and condenser. \( Q \) is the heat transfer rate of the heat pipe. Thermal resistances of heat pipe charged with DI water and charged with nanofluid are compared for two different heat inputs. The thermal resistance of 0.22 and 0.21 °C/W respectively observed for 250 and 350 W for DI water. For nanofluid, reduction in thermal resistance of 35% has been observed when compared with that of the heat pipe with DI water for the same heat input. A similar resistance reduction of 33% was observed in the experimental study (Figure 5.8).

Figure 5.39 shows the x-directional velocity profile devoted as \( u \) velocity at the center line of heat pipe at different heat inputs for both DI water and nanofluid. It is seen that, the \( u \) velocity is zero at the beginning of evaporation section and gradually increases towards the adiabatic section. The maximum velocity is at the beginning of adiabatic sections and the velocity remains constant along the adiabatic section. In the condenser section it decreases and becomes zero at the end. It is also observed that the velocity increases with the increase in the heat input. An increase in the \( u \) velocity for 250 and 350 W respectively is 0.6 and 1.05 m/s with the heat pipe that uses nanofluid.
Figure 5.36 Temperature contours of heat pipe at (a) 250 W with water (b) 250 W with nanofluid (c) 350 W with water and (d) 350 W with nanofluid.

Figure 5.37 Temperature profile of heat pipe charged with water and nanofluid at 250 W.
Figures 5.40 and 5.41 show the vapor and liquid velocity profiles respectively along the y direction at x=0.1 m. It is seen that the vapor velocity (Figure 5.40) is maximum at the center of the vapor core and decreases towards the wick/vapor interfaces. The vapor and liquid velocities are increased when the nanofluid is replaced with DI water at the same power level. Also it is observed that the order of magnitude of the vapor velocity is three times higher than that of the liquid velocity.

![Temperature profile graph](image)

Figure 5.38 Temperature profile of heat pipe charged with water and nanofluid at 350 W

The variation of operating pressure with respect to time is presented in Figure 5.42. The evaporator wall temperature at x = 0.5 m is obtained over the time and presented in Figure 5.42. Similarly the vapor temperature at the midpoint of the adiabatic section (x = 0.15 m, y = 0.0095) is presented in Figure 5.44. From the Figure 5.42, it is observed that as the time increases, the operating pressure of both heat pipes with DI water and nanofluid is increasing for a given heat input. The increase in operating pressure leads to an increase in the evaporator temperature.
(Figure 5.43) as well as the vapor core temperature (Figure 5.44) of a heat pipe. However the use of nanofluids in the heat pipe significantly reduces the operating pressure, evaporator wall temperature and vapor temperature of the heat pipe when compared with that of the heat pipe with DI water at the same heat input. Interestingly, it is noted that the heat pipe charged with nanofluids attains the steady state faster than the heat pipe with DI water.

The transient and steady state liquid pressure ($\Delta P_l$) distribution along the wick-vapor interface and along the axial direction is shown in Figure 5.45 and 5.46 respectively for heat pipe with DI water and nanofluid. The pressure drop at each location of the heat pipe is calculated by subtracting the absolute pressure at the right end of the condenser from the actual pressure at the desired location. If the pressures in the liquid and vapor are assumed to be approximately the same in the condenser section, the pressure difference at the evaporator section must be supported by the capillary head. This pressure drop is used to identify the location of dry-out in the evaporator section. Dry-out occurs or initiates at the position where the pressure drop is maximum. From Figures 5.45 and 5.46, it is clearly seen that the maximum pressure drop is found at the left end of evaporator during transient and steady state conditions. This leads to the occurring of dry-out at the left end of the evaporator section. The pressure drop at the wick-vapor interface is higher for heat pipe charged with nanofluids when compared to the one charged with DI water. The reason for this increase in pressure drop is due to the addition of metallic nanoparticles, which form a porous coating in the wick region. The addition of nanoparticles leads to an increase viscosity of working fluid, resulting high dynamic pressure losses. Also the deposition of nanoparticles in the wick structure offers an additional resistance to the fluid flow flowing from condenser to the evaporator.
Figure 5.39 Velocity at the center line of heat pipe

Figure 5.40 Velocity of vapor at the evaporator section (x = 0.1 m)
Figure 5.41 Velocity of liquid at the evaporator section (x = 0.1 m)

Figure 5.42 Variation of operating pressure in the heat pipe
Figure 5.43 Variations of temperature at the evaporator section of heat pipes

Figure 5.44 Variation of vapor temperature
Figure 5.45  Liquid pressure drop at the liquid-vapor interface of the heat pipe filled with DI water at 350 W

Figure 5.46  Liquid pressure drop at the liquid-vapor interface of the heat pipe filled with nanofluid at 350 W
The capillary pressure head ($\Delta P_c = \Delta P_l + \Delta P_v$) is calculated by adding the liquid and vapor pressure drops at the liquid/vapor interface. The vapor pressure drop at the liquid-vapor interface is about 0.7 Pa and 0.4 Pa respectively for DI water and nanofluid. The maximum capillary head determined from the Figure 5.47 is 2776 Pa and 5236 Pa respectively for water and nanofluid at 350W. The pore size of the wick structure is calculated by using the capillary head as $2\sigma/r_c = \Delta P_{cap}$. The pore radius required to support this capillary pressure head is $5.04 \times 10^{-5}$ and $2.63 \times 10^{-5}$ m respectively for DI water and nanofluid. Thus a pore radius can be calculated for nanofluids.

Figure 5.47 Comparisons of pressure drop at the liquid-vapor interface of heat pipe charged with DI-water and nanofluid at 350 W

![Graph showing pressure drop at liquid-vapor interface of heat pipe charged with DI-water and nanofluid at 350 W](image)