CHAPTER 1 INTRODUCTION

1.1 Motivation

Electronic equipments are being used in every aspect of modern life; from toys and home appliances to high power computers; from space systems to process industries. The reliability of the electronic components of a system is a major factor in the overall reliability of the system. Heat generation is an inescapable factor in all these systems. Continued miniaturization of the electronic circuits and the resulting high packing density has resulted in a dramatic increase in heat generation. A high rate of heat generation leads to corresponding high operating temperature of the electronic components which compromises its safety and reliability resulting in the higher failure rate of the systems. Hence the thermal control is an important factor in the operation of electronic equipment.

<table>
<thead>
<tr>
<th>S.No</th>
<th>Mode of heat transfer</th>
<th>Cooling medium</th>
<th>Heat transfer coefficient (W/m²-K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Free convection</td>
<td>Air</td>
<td>5-25</td>
</tr>
<tr>
<td>2</td>
<td>Forced convection</td>
<td>Air</td>
<td>10-200</td>
</tr>
<tr>
<td>3</td>
<td>Free convection</td>
<td>Water</td>
<td>20-100</td>
</tr>
<tr>
<td>4</td>
<td>Forced convection</td>
<td>Water</td>
<td>50-10000</td>
</tr>
<tr>
<td>5</td>
<td>Boiling (phase change)</td>
<td>Water</td>
<td>3000-100000</td>
</tr>
<tr>
<td>6</td>
<td>Condensing(phase change)</td>
<td>Water vapor</td>
<td>5000-100000</td>
</tr>
</tbody>
</table>

There are several cooling techniques such as conduction, natural convection and radiation, forced –air cooling, liquid cooling, and immersion cooling which are
used to effect the thermal control. Efficiency of the cooling can be denoted by the heat transfer coefficients realizable on the cooling surfaces. Table 1.1 presents the heat transfer coefficients achieved in various cooling methods. Based on the convective heat transfer coefficient, electronic cooling methods can be categorized into the following three major types:-

(i) Natural convection with single phase heat transfer
(ii) Forced convection with single phase heat transfer
(iii) Phase change heat transfer

Generally, the heat transfer coefficient of an air cooled system (single phase) is lower than the liquid cooled systems (single phase) due to the latter’s superior thermal properties. Similarly, the heat transfer coefficient of single phase forced convective heat transfer is higher than the single phase free convection heat transfer. Compared to the above single phase heat transfer processes, the phase change heat transfer process offers much higher heat transfer coefficient as shown in the Table 1.1. Hence the two phase cooling process is preferred wherever possible. However, the application of two phase cooling method is limited due to the space constraints and compatibility of the working fluids with the electronic component materials compared to those working fluids used in the immersed cooling method. In the immersed cooling method, the electronic components are directly immersed in a dielectric fluid and the boiling occurs at the surface of the electronic components. The main advantage of using the dielectric fluid is that these fluids do not have a high electrical conductivity compared to the other fluids such as water. In such situations with spatial constraints, the thermoelectric coolers and heat pipes are the alternatives for high heat transfer. The major disadvantages of using the thermoelectric cooler are low heat flux handling, high material cost and additional power consumption. Heat pipe is a passive heat transfer device, which works based on the phase change process of the working fluid. In this device, working fluids of high thermal conductivity and heat capacity can be used for transporting the heat from sources to sinks.
1.2 Historical background

The heat pipe concept was originally proposed by King and Perkins during mid 18\textsuperscript{th} century in the United Kingdom. Perkins tubes differ from present day heat pipe in the sense that they do not have a wick structure and are gravity assisted. The heat transfer takes place by absorption and rejection of latent heat during the phase change of working fluids as shown in Figure 1.1(a). This wickless heat pipe is also called as “Thermosyphon” (Faghri, 1995). Thereafter, Gaugler (1944) developed a device which would work without the assistance of gravity. In this device, liquid evaporates at a point above the place where condensation would occur and the condensed liquid is moved to higher elevation without requiring any additional work. The principle of the heat pipe is illustrated in Figure 1.1(b). It is a heat-transfer device consisting of a sealed metal tube with an inner lining of a wick-like capillary material and a small amount of fluid in partial vacuum. Heat is absorbed at one end of the enclosure by vaporization of the fluid and is released at the other end by condensation of the vapor. After the early development, many prototypes of heat pipes were developed and used in various applications - mainly in space programs. Now heat pipes are used worldwide in varying applications including heat recovery (Chaudhry, 2012), electronic cooling (Kawakara, 2012) laser diode cooling (Chien, 2003) and a number of terrestrial applications.

1.3 Operational limits

The rate of heat transport through a heat pipe is subject to a number of operating limits. The physical phenomena that might limit heat transport in heat pipes are due to a number of technical factors such as capillary, sonic, entrainment, boiling, frozen startup, continuum vapor flow as well as vapor pressure and condensation effects. These limitations depend on the size and shape of the pipe,
working fluid, wick structure, and operating temperature (Faghri, 1995). A
description of the process involved is shown in Figure 1.1

![Figure 1.1 (a) Thermosyphon (b) Heat pipe](image)

1.3.1 Capillary limit

When the capillary forces, developed throughout the wick, are insufficient to
drive back enough liquid from the condenser to the evaporator, dry-out may occur in
the evaporator section and the heat pipe will fail to operate. Thus, the capillary limit
is defined as the power level at which capillary action first fails to supply enough
fluid to wet the entire evaporator. Under normal steady-state operations, the wick
structure of a heat pipe has to develop sufficient capillary-driven pressure to
overcome the total pressure drop. At any time, the following condition on the
capillary pressure ($\Delta p_{cap}$) must be satisfied
\[
\Delta p_{\text{cap}} \geq \Delta p_{\text{tot}}
\]  
(1.1)

where the \(\Delta p_{\text{tot}}\) is the sum of frictional pressure drops along the liquid path \(\Delta p_l\) and vapor path \(\Delta p_v\), pressure drops in the liquid due to body forces \(\Delta p_g\) (gravity, electromagnetic etc…), and pressure drops due to phase change at the liquid-vapor interface. For heat pipes operated at room-temperature, the pressure drops related to liquid-vapor phase change are generally negligible since the evaporation or condensation rates are relatively low.

Hence, the balance of pressure drops may be reduced to:

\[
\Delta p_{\text{tot}} = \Delta p_l + \Delta p_v + \Delta p_g
\]  
(1.2)

By substituting the pressure drops into equation 1.2 gives

\[
\frac{2\sigma \cos \theta}{r_c} = \frac{16\mu_v \rho_{\text{eff}} Q}{2\pi r_v^2 A_\nu \rho \nu \lambda} + \frac{\mu_l \rho_{\text{eff}} Q}{K A_\omega \rho \nu \lambda} + \rho_l g d_v
\]  
(1.3)

Simplifying for the capillary limit \((Q)\),

\[
Q = \left(\frac{2\sigma \cos \theta}{r_c} - \rho_l g d_v\right)\left[\frac{16\mu_v \rho_{\text{eff}}}{2\pi r_v^2 A_\nu \rho \nu \lambda} + \frac{\mu_l \rho_{\text{eff}}}{K A_\omega \rho \nu \lambda}\right]^{-1}
\]  
(1.4)

### 1.3.2 Boiling limit

If the radial heat flux in the evaporator section becomes too high, the liquid in the evaporator wick boils and the wall temperature becomes excessively high. The vapor bubbles that form in the wick prevent the liquid from wetting the pipe wall, causing hot spots. If this boiling is severe it dries out the wick in the evaporator, which is defined as the boiling limit (Faghri, 1995). This limit can be determined by the following equation:-

\[
q_{b,e} = \left(\frac{2\pi \rho \lambda f_{\text{eff}} T_v}{\lambda \rho_v \ln\left(\frac{T_e}{T_v}\right)}\right)\left(\frac{2\sigma}{r_{\text{cap}}} - \Delta p_{c,m}\right)
\]  
(1.5)
1.3.3 Sonic limit

The evaporator and condenser sections of a heat pipe represent a vapor flow channel with mass addition and extraction due to the evaporation and condensation, respectively. The vapor velocity increases along the evaporator and reaches a maximum at the end of the evaporator section. The limitation of such a flow system is similar to that in a converging – diverging nozzle with a constant mass flow rate, where the evaporator exit corresponds to the throat of the nozzle. Hence, the vapor velocity at this point cannot exceed the local speed of sound. This choked flow condition is called sonic limit. This limit can be calculated by the following equation

\[ q_s = A_v \rho_v \lambda \left[ \frac{y_v R_v T_v}{2(y_v + 1)} \right]^{1/2} \]  

(1.6)

1.3.4 Entrainment limit

At high relative velocities of liquid and vapor at the interface, droplets of liquid can be torn off from the wick surface and entrained into the vapor region that is flowing towards the condenser section. This entrainment of liquid leads to the dry-out in the evaporator due to the shortage of liquid inventory. This limit is called entrainment limit and calculated as

\[ q_t = A_v \lambda \left( \frac{\sigma \rho_v}{2 r_{h,w}} \right)^{1/2} \]  

(1.7)

1.3.5 Viscous limit

At low temperature, the vapor pressure difference between the closed end of the evaporator and closed end of the condenser may be very small. In such situation the viscous forces can be dominant and limit the operation of heat pipe. The maximum rate of heat transfer under this restricted vapor pressure drop limit is given by:

\[ q_v = \frac{A_v \rho_v^2 \lambda \rho_v R_v}{16 \mu_v L_e} \]  

(1.8)
1.4 Literature review

The heat transfer characteristics and limitations of heat pipes have been studied extensively and presented in the literature (Faghri 1995). This literature presents the research works that are closely related to the present experimental and numerical studies. It includes the heat transfer characteristics of heat pipes as well as thermosyphons using nanofluids and the details relating to mathematical modeling of heat pipes.

1.4.1 Heat pipe with nanofluids

It is well known that the thermophysical properties of heating or cooling fluids play a major role in the development of energy-efficient heat transfer equipment. However, the conventional heat transfer fluids such as ammonia, water, methanol, and glycol as well as engine oils have, in general, poorer heat transfer properties than most metallic solids (Shukla 2008, 2009). There have been some attempts to improve the heat transport capability of the working fluids by adding ultrafine metal particles to the base fluids to produce nanofluids. Choi (1995) from Argonne National Laboratory, U.S.A proposed a new class of fluids called “nanofluids” which is a suspension of solid nano sized metal particles with traditional fluids.

Wang et al. (1999) measured the thermal conductivities of nanoparticles in fluid mixture. Xuan and Qiang (2000) presented a procedure of preparing nanoparticles suspensions. The thermal conductivities of Cu nanoparticle suspensions in transformer oil and water were measured. It was observed that the thermal conductivity of nanofluids depends on the stability of the nanofluids and was improved by adding laurite salt. Liu et al. (2006) presented the thermal conductivity enhancement of water in the presence of copper using chemical reduction method. It was observed that the thermal conductivity was enhanced by 23.8% for Cu-water
suspension with a volume fraction of 0.001 of Cu powder. Kim et al. (2007) studied the effect of morphology of carbon nanotubes on the enhancement of thermal conductivity of aqueous fluids. Wright et al. (2007) reported the enhancement of thermal conductivity of nanofluids containing Ni coated single wall carbon nanotubes by an applied magnetic field.

At the first scientific conference centered on the theme on “Nanofluids: Fundamentals and Applications” during September 16-20 in 2007 at Copper Mountain, Colorado, it was decided to begin an International Nanofluid Property Benchmark Exercise (INPBE) to decide the inconsistencies in the database and help advance research on nanofluid properties. Based on the reports on INPBE, Buongiorno et al. (2009) presented the benchmark data of the thermal conductivity for various nanofluids. However, the mechanism for the enhancement in thermal conductivity is still unclear and the subject under research. Eastman et al. (2004) proposed four possible mechanisms for the thermal conductivity enhancement as revealed in Figure 1.2. They are:

- the Brownian motion of the nanoparticles
- the molecular-level layering of the liquid at the interface between liquid and nanoparticle
- the nature of heat transfer in the nanoparticles
- the effects of nanoparticle clustering

Similarly, Xuan and Li (2000) also presented four possible mechanisms for the improved effective thermal conductivity of nanofluids: the increased surface area due to the dispersion nanoparticles, the interaction and collision among suspended particles, the intensified mixing fluctuation and turbulence of the fluid, followed by the dispersion of nanoparticles.
Xu et al. (2010) performed an experimental study of convective heat transfer of self-assembled ethanol in polyalphaolefin (PAO) nano-emulsion fluids and studied the thermophysical properties. It includes the thermal conductivity and viscosity of nano-emulsion at various temperatures. The enhancement in thermal conductivity was found to be moderate but increases rapidly with the increasing temperature (in the range of 35 - 75 °C). A significant enhancement in the convective heat transfer coefficient, by a factor of 2.2, obtained when the nano-emulsion fluids were used. It was proposed that an explosive vaporization of the ethanol nano-droplets was the possible reason for the heat transfer coefficient enhancement. Such an explosive liquid–vapor phase transition might augment the fluid heat transfer through the heat of vaporization and the fluid mixing induced by the sound waves. Development of these kinds of phase changeable nano-emulsions would open a new direction for the thermal fluids studies.

The heat transfer phenomenon of nanofluids was analyzed in various studies. Zhou (2004) investigated the heat transfer characteristics of Cu nanofluids subjected
to acoustic cavitations in an acetone based Cu nanofluid suspension. It was observed that the single phase heat transfer was enhanced due to the addition of a small amount of nanoparticles while boiling heat transfer was reduced. Nguyen et al. (2007) presented an experimental heat transfer enhancement using alumina-water mixture for cooling a microprocessor and other electronic components. It was found that the heat transfer coefficient was enhanced up to 40% as compared to the base fluid for a volume concentration of 6.8%. He et al. (2007) presented the flow behavior and the heat transfer regimes (both laminar and turbulent) for a vertical pipe. Convective heat transfer characteristics in the developing region of tube flow with constant heat flux using alumina–water nanofluids were studied by Anoop et al. (2009). The effect of particle size on heat transfer in laminar region was also investigated.

Nanofluids were considered as the working fluid for heat pipes in many electronic cooling applications. Tsai et al. (2004) have examined the effect of structural characteristics of nanoparticles on the thermal performance of heat pipe with nanofluids and observed that the thermal resistance of the heat pipes with nanofluid was lower than that of distilled water. Similar thermal resistance reduction was observed in the experimental investigation of grooved heat pipes charged with nanofluids by Kang et al. (2006). Park and Ma (2007) studied the effect of nanofluids on heat transport capability in a well-balanced oscillating heat pipe. It was observed that the maximum thermal resistance reduction was about 50% for 10 nm sized particles and the reduction was 80% for the particle size of 35 nm when compared with DI-water filled heat pipes. The performance of sintered heat pipe was studied by Kang et al. (2009) using 10 nm and 35 nm size silver nanofluids. The study concluded that the wall temperature was reduced by 0.56 °C to 0.65 °C compared with DI water filled heat pipe. Hajian et al. (2012) carried out an experiment to study the effects of silver nanofluids on the thermal performance of a
medium-sized (1m length and 1inch diameter) cylindrical meshed heat pipe, in both transient and steady states. It was found that by using nanofluids, the thermal resistance was decreased by 30% and response time of the heat pipe decreased by 20% compared to DI water.

Liu et al. (2007) tested a flat heat pipe with CuO-water nanofluid. This experiment confirmed that the boiling heat transfer characteristics of the miniature flat heat pipe (MFHP) evaporator can be strengthened by using CuO-water nanofluids. It was also observed that the heat transfer coefficient and critical heat flux increase with the increase of the concentration when the mass concentration was less than 1%. In another study, Liu et al. (2011) found that the inclination angle had a strong effect on the heat transfer of grooved heat pipes charged with both DI water and the CuO nanofluid. A higher thermal performance was obtained at the inclination angle of 45° for heat pipes charged with both water and the nanofluid. In a similar study, Liu et al. (2010) investigated a horizontal meshed heat pipe charged with CuO nanofluid and water at different operating pressures. A significant enhancement in the heat transfer coefficients of both evaporator and condenser was observed. It was also found that the maximum heat flux of heat pipe was increased when the CuO particles were added into the DI water. Further, it was observed that the operating pressure had a noticeable impact on both the evaporating and condensing heat transfer enhancements. Yang et al (2008) found that the optimum concentration to achieve the maximum heat transfer using CuO-water nanofluid was about 1 wt%. Also it was observed that the operating pressure had an apparent influence on both the heat transfer coefficient and the Critical Heat Flux (CHF) of nanofluids. Wang et al. (2010) found that the total thermal resistance was reduced by 50% and the heat removal capacity enhanced by 40% when DI water was replaced with CuO nanofluids. Also it was found that the thermal performance of
miniature grooved heat pipe can be improved after substituting water by CuO nanofluid. In a comprehensive study, Liu et al. (2011) investigated the effect of nanoparticle material, size as well as mass concentration and operating pressure on the evaporation and condensation heat transfer coefficients, the maximum heat flux and the total thermal resistance of the heat pipe. Five kinds of nanoparticles were taken for the studies which were Cu with two mean diameters of 40 nm and 20 nm, CuO with two mean diameters of 50 nm and 20 nm and SiO with a mean diameter of 30 nm. The results showed that adding Cu and CuO nanoparticles into the base fluid can apparently improve the thermal performance of the heat pipe and there was an optimal nanoparticle mass concentration to achieve the maximum heat transfer enhancement. However, addition of SiO nanoparticles into the base fluid deteriorates the heat transfer performance. It was found that the coating layer formed on the heated surface was responsible for the differences in the heat transfer performance.

Naphon et al. (2008) studied the effect of titanium nanofluids on heat pipe performance for different percentages of nanoparticles, tilt angles and fluid charges. The results showed that the heat pipe efficiency was 10.6% higher than that with the base fluid for 0.1% volume concentration. In another study, Naphon et al. (2009) presented an enhancement of heat pipe efficiency with refrigerant–nanoparticles mixtures. The refrigerant-R11 was used as a base working fluid and the size of the titanium nanoparticle was 21 nm. It was found that the efficiency of the heat pipe was enhanced by 1.4 times as that with pure refrigerant for 0.1% nanoparticles concentration.

Liu et al. (2009) studied the heat transfer performance of an axially micro-grooved heat pipe using water-based carbon nanotube suspensions as the working
fluid. The effects of mass concentration of the carbon nanotube and the operating pressure on the evaporation and condensation heat transfer coefficients were investigated. The experimental results showed that carbon nanotube suspensions apparently improve the thermal performance of the heat pipe and there was an optimal mass concentration of the carbon nanotube (about 2.0%) to achieve the maximum heat transfer enhancement. Also, it was found that the operating pressure had a significant influence on the enhancement of heat transfer coefficients and minor influences on the enhancement of the maximum heat flux.

Mousa (2011) experimentally studied the behavior of nanofluids to improve the performance of a circular heat pipe using pure water and Al$_2$O$_3$-water based nanofluid as working fluids. The effect of filling ratio, volume fraction of nanoparticles in the base fluid and heat input on the thermal resistance was studied. The results showed that the thermal resistance decreases with the increasing of Al$_2$O$_3$ nanoparticle content in the base fluid compared to that of pure water. Similar results have also been observed by Moraveji et al (2012) for ‘L’ shaped heat pipe. Hung et al. (2012) studied the thermal performance of heat pipe using Al$_2$O$_3$-water nanofluid. The effect of fill ratio, tilt angle, length of the heat pipe, heating power, and weight fraction of nanoparticle were considered in this study. The experimental results showed that all the above said parameters affect the thermal performance of the heat pipe. At a heating power of 40 W, the optimal thermal performance of heat pipes with lengths 0.3 m, 0.45 m, and 0.6 m using Al$_2$O$_3$-water nanofluid was 22.7%, 56.3%, and 35.1%, respectively observed when DI water used as the working fluid. Also it was found that the higher concentrations of nanofluids affects the thermal performance of heat pipe due to high water adsorption, which in turn facilitates forming a coating layer through the sedimentation of nanoparticles on the surface of the evaporation section.
Teng et al. (2010) studied the effects of filling quantity, inclination angle of heat pipe and weight fraction of nanoparticles on the thermal efficiency of straight copper heat pipe. It was found that the thermal efficiency was enhanced by 16.8% higher than that of heat pipe charged with DI water. It was also found that 1 wt% was the optimum concentration for the smooth operation of heat pipe. Do et al. (2010) studied the effect of nanofluids on the thermal performance of heat pipes by testing circular screen mesh wick heat pipes using Al$_2$O$_3$-water nanofluids with the volume fraction of 1.0 and 3.0. A reduction in evaporator wall temperature as well as thermal resistance and enhancement in the maximum heat transport rate were found by introducing the Al$_2$O$_3$ nanofluids instead of water. Also it was stated that these improvements were not because of the enhancement in the thermal conductivity of the nanofluids but because of the thin porous coating layer formed by nanoparticles at wick structures.

Saleh et al. (2012) prepared ZnO powders using a chemical precipitation method and tested a heat pipe and measured the temperature distribution and thermal resistance of the heat pipe. In this study, ZnO nanoparticles were dispersed in ethylene glycol and used as the working fluid. The results showed that the temperature distribution and thermal resistances were decreased as the concentration and the crystalline size of the nanoparticles were increased. Kole and Dey (2012) performed an experimental investigation using copper nanofluids in screen meshed heat pipes. The wall temperature distributions and the thermal resistances between the evaporator and the condenser sections of a heat pipe containing nanofluids were investigated for three different angular positions. It was found that the wall temperature of the heat pipes decreased along the test section from evaporator to the condenser and increased with input power. The average wall temperatures at the evaporator of the heat pipe with nanofluids were much lower than those of the heat pipe with distilled water.
1.4.2 Thermosyphon with nanofluids

Xue et al. (2006) carried out an experiment to study the effect of carbon nanotube suspension on the thermal performance of a two-phase closed thermosyphon. In this study, an aqueous solution of carbon nanotubes was utilized as the working fluid. In comparison with the thermosyphon filled with distilled water, the one filled with carbon nanotube suspension had a high evaporation section wall temperature and higher thermal resistance. Also the nanofluid with carbon nanotubes deteriorated the performance of the thermosyphon.

Liu et al. (2007) performed an experiment to investigate the effect of concentration of nanoparticles in the nanofluid on the thermal performance of a miniature thermosyphon. The results showed that the CuO-water nanofluids can greatly enhance the boiling heat transfer performance of thermosyphon compared with that of same charged with water. Also it was found that the optimal mass concentration of nanoparticles to achieve the maximum heat transfer performance was 1.0 wt%. Liu et al. (2010) studied the heat transfer performance of a miniature thermosyphon with water based carbon nanotube (CNT) suspensions as the working fluid. The suspensions consisted of DI water and multi-wall carbon nanotubes with an average diameter of 15 nm and a length ranges from 5-15 µm. The experiments were performed under three operating pressures of 7.4 kPa, 13.2 kPa and 20 kPa. The effects of the mass concentration of CNT and the operating pressure on the average evaporation and condensation heat transfer coefficients, the critical heat flux and the total thermal resistance of the thermosyphon were investigated. The results showed that CNT suspensions can apparently improve the thermal performance of the thermosyphon and there was an optimal CNT mass concentration of nanofluid (about 2.0%) to achieve the maximum heat transfer enhancement. Also it was found that the operating pressure had a significant influence on the enhancement of the
evaporation heat transfer coefficient, and a slight influence on the enhancement of the critical heat flux. Further, the enhanced heat transfer effect was weak at low heat fluxes while it increased gradually with increasing the heat flux.

Khandekar et al. (2008) studied the overall thermal resistance of closed two-phase thermosyphon using pure water and various water-based nanofluids (of Al$_2$O$_3$, CuO and laponite clay) as working fluids. It was observed that all these nanofluids showed inferior thermal performance than pure water. Further it was found that the wettability of all nanofluids on copper substrate, having the same average roughness as that of the thermosyphon container pipe, was better than that of pure water. Their scaling analysis showed that the increase in wettability and entrapment of nanoparticles in the grooves decreased the evaporator side Peclet number which led to poor thermal performance. Noie et al. (2009) studied the efficiency of a thermosyphon with Al$_2$O$_3$ nanoparticles suspensions as the working fluids. The experimental results showed that the efficiency of the thermosyphon increased up to 14.7% when Al$_2$O$_3$-water nanofluid was used instead of pure water. Also it was found that the wall temperature profile of the thermosyphon with nanofluid was lower than that of the one filled with pure water. Parametthanuwat et al. (2010) studied the thermal characteristics of a two-phase closed thermosyphon with silver-water nanofluids. The thermosyphons were made of copper tubes with an inner diameter of 7.5, 11.1 and 25.4 mm. Filling ratios of 30%, 50% and 80% with respect to evaporator length and aspect ratios of 5, 10, and 20 with an inclination angle of 90° were used. The operating temperatures were 40 °C, 50 °C and 60 °C. It was found that the filling ratio had no effect on the heat transfer characteristics of the thermosyphon in the vertical position. In another study (Parametthanuwat, 2010) they found that the maximum heat transfer rate of 750.8 W was observed with aspect ratio of 20 (ID of 25.4) and working temperature of 60 °C.
Huminic et al. (2011) presented an experimental investigation on the use of iron oxide nanoparticles in the thermosyphon. Iron oxide particles were added to water and used as the working fluid. The temperature distribution of the thermosyphon at various concentrations of the nanofluid was measured. Their results showed that the thermal performance of thermosyphon was improved by the nanofluid compared with DI water. In another case, Huminic et al. (2011) performed a series of experiments to study the effects of inclination angle, operating temperature and nanoparticles concentration on the heat transfer characteristics of thermosyphon. It was found that the nanoparticles had a significant effect on the enhancement of heat transfer characteristics of the thermosyphon.

Yang et al. (2012) carried out an experimental study to understand the flow boiling heat transfer of water based CuO nanofluids in the evaporator of a thermosyphon loop under steady sub-atmospheric pressures. The experimental results showed that both the heat transfer coefficient and the critical heat flux of flow boiling in the evaporator of the thermosyphon loop can be enhanced by substituting nanofluids for water. The operating pressure had a significant impact on the heat transfer coefficient of nanofluids. Also it was found that the operating pressure had no effect on the critical heat flux enhancement.

Lu et al. (2011) carried out an experiment to investigate the thermal performance of the open thermosyphon using DI water and water-based CuO nanofluids as the working fluids. The effects of filling rate, type of the base fluid, mass concentration of nanoparticle and the operating temperature on the evaporating heat transfer characteristics were studied. It was found that the optimal filling ratio to the evaporator is 60% and the thermal performance of the open thermosyphon increases generally with the increase of the operating temperature. Yang et al. (2011) carried out an experiment to understand the effect of water-based surface
functionalized (fluids with surface modified nanoparticles) special nanofluids on the thermal performance of the thermosyphon. Another experiment was conducted with traditional nanofluid (fluids without surface modified nanoparticles) as the working fluid. The results confirmed that an existence of porous deposition layer on the heated surface of the evaporator during the operation of thermosyphon using traditional nanofluid. However, there was no coating layer formed in the evaporator while using the functionalized nanofluid. Also it was observed that functionalized nanofluid enhanced the evaporator heat transfer coefficient, though it had no effect on the maximum heat flux. Finally, Liu et al. (2012) summarized the research work done on heat pipes using nanofluids as working fluids in recent years. The effect of characteristics and mass concentrations of nanoparticles on the thermal performance in various kinds of heat pipes with different base fluids under various operating conditions was discussed. The mechanism of enhancement or degradation of heat transfer utilizing nanofluids in the heat pipe was explained.

1.4.3 Computational studies on heat pipes and thermosyphon

Efforts have been made to simulate the working of heat pipes over the past few decades. These studies related the flow and heat transfer in cylindrical heat pipes, flat heat pipes, micro heat pipes, thermosyphons and vapor chambers. Relevant literatures related to the present study are provided in this section.

Tien and Rohani (1974) studied the effects of vapor pressure drop on the heat pipe performance. This study includes the effects of vapor pressure variation on vapor temperature distribution and the overall performance of the heat pipe. In this analysis, mass, momentum and energy conservation equations in conjunction with the thermodynamic equilibrium relations and appropriate boundary conditions were solved numerically for a cylindrical heat pipe. This study demonstrated that the
approximate solution based on the parabolic boundary-layer equations does not provide an accurate picture of vapor pressure variations at relatively high evaporation and condensation rates. Heat conduction in the wall, liquid-wick regions and compressibility effect of the vapor inside the heat pipe were considered in a study of Chen and Faghri (1990). In this numerical study, the two-dimensional governing equations in conjunction with the thermodynamic equilibrium relations and appropriate boundary conditions were solved. Huckaby et al. (1994) used a collocation-spectral method to compute the dynamic behavior of the vapor flow in a heat pipe. In this method, one-dimensional heat pipe was considered as sample problem with evaporation at one boundary and condensation on the other.

Two-dimensional, HPTAM (Heat pipe Transient Analysis Model) was developed and benchmarked by Tournier et al. (1994) using transient experimental data for cumulating operations of fully-thawed heat pipes. Steady state vapor and axial wall temperature profiles and the transient power throughput were presented. The axial temperature profiles of liquid and vapor pressures of heat pipes and effective radius of curvature of the liquid meniscus at the liquid-vapor interface was also presented. In another study, Tournier et al. (1996) developed a free-molecular continuum vapor flow model and incorporated in HPTAM to analyze the startup of a radiatively-cooled sodium heat pipe from a frozen state. Vapor flow and wall temperatures at different timings during the startup were obtained in this study.

Legierski et al. (2006) conducted a study on modeling and measurements of the heat and mass transfer in heat pipes using commercial software. In this study, the physical model included the effects of liquid evaporation and condensation inside the heat pipe. The internal vapor flow was completely simulated and the transient temperature profiles were compared with experimental results. In a rare attempt, Alizadehdakhel et al. (2010) studied a gas/liquid two-phase flow and the
simultaneous evaporation and condensation phenomena in a thermosyphon using commercial code. The volume of fluid model (VOF) technique was used to model the interaction between these phases.

Flat heat pipes were used to remove the heat or flatten the temperature gradient over high density electronic packages. Vadakkan et al (2003) developed a stable numerical procedure to analyze the transient performance of flat heat pipes for large input heat fluxes and high wick conductivity. A structured collocated finite volume scheme was used in conjunction with the SIMPLE algorithm to solve the continuity, energy and momentum equations. In addition, system pressurization was computed using overall mass balance. The improved numerical scheme was used to analyze the flat two dimensional heat pipes. In another study, Vadakkan et al. (2004) developed a three-dimensional model to analyze the transient and steady state performance of the flat heat pipe with multiple heating sources. Three-dimensional heat conduction in the heat pipe wall, flow and energy equations in the vapor and liquid region were solved. In the wick region, they used an equilibrium model for heat transfer and a Brinkman-Forchheimer extended Darcy model for fluid flow. The change in density due to the pressurization of the vapor was incorporated in the continuity equation. The temperature, vapor flow and hydrodynamic pressure distributions were calculated with respect to time from coupled continuity, momentum and energy equations in the wick and vapor regions. The mass flow rate at the liquid-vapor interface was obtained from the kinetic theory.

Ranjan et al. (2011) developed a microscale evaporation model for thin-film evaporation in capillary wick structures. This model considered the evaporating liquid meniscus in wick microstructures under saturated vapor conditions. Thermo-capillary convection due to non-isothermal conditions at the liquid/vapor interface was also included in this model. This microscale evaporation model was coupled
with the general (macro) model to include the microstructure effects in the liquid/vapor interface (Rajan 2011). Meniscus curvature at every location along the wick was calculated as a result of this coupling. This coupled model was used to predict the thermal transport in heat pipes and vapor chambers.

Carbajal et al. (2007) developed a quasi-3D numerical model to analyze the flat heat pipe. This model was capable of predicting the temperature distributions and flow field variables. In this study, explicit finite volume method with small step size was used for computations in the vapor phase. The physical problem consisted of an evaporator surface. This surface was subjected to transient non-uniform heating for a short period of time and the heat source was removed. Then the system was allowed to cool by natural convection and radiative heat transfer at the condenser region. The velocity distribution in the vapor core and the transient temperature distributions of the heat spreader were obtained.

Rice and Faghri (2007) performed a complete numerical analysis of heat pipes including flow in the wick structure. Single and multiple heat sources were used in this analysis. Also constant convective and radiative heat sinks were used. This study did not fix the internal pressure references by a point, but allowed it to rise and fall based on the physics of the problem.

Wang et al. (2007) simulated the two-phase flow in vapor chamber and compared the simulation results with experimental test data. In this analysis, they studied the effects of film thickness, stagnant liquid phase that exists in the form of mist/droplet near the wall, filling ratio, micro-gravity, minimum power required and effective length on the performance of vapor chamber.
Xiao and Faghri (2008) developed a three-dimensional model to analyze the thermo-hydrodynamic behaviors of flat heat pipes without empirical correlations. In this model, heat conduction in the wall, fluid flow in the vapor chambers and porous wicks and the coupled heat and mass transfer at the liquid/vapor interface were considered. This model was used to analyze the flat heat pipes with and without vertical wick columns in the vapor channel. Parametric effects, including evaporative heat input and size on the thermal and hydrodynamic behavior in the heat pipes were also investigated in this study.

Chen et al. (2009) performed a numerical investigation of a thermal module which includes a plate-fin heat sink embedded with a vapor chamber. A concentrated heat source was used in this study. The internal vapor was assumed as a common heat-transfer interface between the wicks. In addition, the isotropic and orthotropic approaches were proposed to calculate the effective thermal conductivities of the vapor chamber.

Aghvami et al. (2011) developed a simplified analytical thermal-fluid model including the wall and both liquid and vapor flows for flat heat pipes or vapor chambers with different heating and cooling configurations. A two dimensional numerical model which includes the wall, wick and vapor regions was developed and compared the results of numerical model with that of analytical models.

The finite volume method has been widely used to simulate working of the heat pipes. Very few attempts have been made to simulate the heat pipes using finite element and finite difference methods. Jang et al. (NASA REPORT) developed a transient compressible one-dimensional model to study the vapor flow dynamics in a heat pipe. The numerical results were obtained by using the implicit non-iterative Beam-Warming finite difference method. Using this model, the transient behavior of
the vapor flow was successfully predicted under subsonic, sonic, and supersonic speeds. This model also described the vapor flow dynamics in cylindrical heat pipes at high temperatures. Two-dimensional heat transfer and fluid flow in a heat pipe at steady state were numerically simulated using the Finite Element Method (Thuchayapong 2012). Vapor core, wick, wall of container and water jacket were considered in the physical model. To investigate the effect of capillary pressure on performance of a heat pipe, capillary pressure model was used in the liquid-vapor interface as a simple linear function. Kaya et al. (2007) performed a CFD simulation to study the steady state performance characteristics of heat pipes. In this study, a three-dimensional finite-element method was used to solve the mass, momentum and energy equations. The liquid and vapor flow in the entire heat pipe domain was considered for simulation.

Zhu and Vafai (1998) performed an analytical and numerical study for the steady incompressible vapor and liquid flows in an asymmetrical flat plate heat pipe. The three-dimensional analytical model was used with the boundary layer approximation to describe the vapor flow under the strong flow reversal. Also the Non-Darcian effects for the liquid flow through the porous wick were also considered. In the numerical study, a finite element scheme based on the Galerkin method of weighted residuals was used to solve the full set of nonlinear differential elliptical equations of motion and the continuity equation for the three-dimensional vapor flow. The three-dimensional effects were discussed and the results showed that a three-dimensional analysis was necessary if the vapor channel width to height ratio was less than 2.5. Two-dimensional finite element simulation was performed under natural and forced convection modes, by using commercial code (Mohamed 2012). In this simulation the wall temperatures at the evaporator section, adiabatic and condenser sections were analyzed. The velocity and pressure distributions of
vapor and liquid were also analyzed. This work is mainly focused on the characterization of working fluid in a vertically placed twin U-shape heat pipe.

Shafahi et al. (2010) utilized an analytical model to investigate the thermal performance of rectangular and disk-shaped heat pipes using nanofluids. The liquid pressure, liquid velocity profile, and temperature distribution along the wall of heat pipe were analyzed. Also they analyzed the temperature gradient along the heat pipe, thermal resistance and maximum heat transfer capacity for the flat-shaped heat pipes using a nanofluid. It was found that the flat-shaped heat pipe’s thermal performance using a nanofluid was substantially enhanced compared with one used a regular fluid. The nanoparticles present within the working fluid result in a decrease in the thermal resistance and an increase in the maximum heat load capacity of the flat heat pipe. In another study, Shafahi et al. (2010) conducted a two-dimensional analysis to study the thermal performance of a cylindrical heat pipe using nanofluids. The nanoparticles, namely Al$_2$O$_3$, CuO and TiO$_2$ were considered as the working fluids. It was observed a substantial change in thermal resistance of the heat pipe, temperature distribution and maximum capillary heat transfer of the heat pipe by introducing a nanofluid. It was also found that the nanoparticles within the liquid could enhance the thermal performance of the heat pipe by reducing the thermal resistance while enhancing the maximum heat load it could carry. The existence of an optimum mass concentration for nanoparticles in maximizing the heat transfer limit was established. From this investigation, it was also found that smaller particles had a more pronounced effect on the temperature gradient along the heat pipe.

Alizad et al. (2012) analyzed the thermal performance of flat-shaped heat pipe using an analytical model. Using this model the transient behavior and operational start-up characteristics of heat pipes were analyzed. This model accounts
for the heat transfer characteristics within the heat pipe wall and the wick within the condensation and evaporation sections. Three different primary nanofluids namely, CuO, Al₂O₃, and TiO₂ were utilized in this analysis. The results illustrated that there was an enhancement in the heat pipe performance while achieving a reduction in the thermal resistance for both flat-plate and disk-shaped heat pipes throughout the transient process. Also it was observed that a higher concentration of nanoparticles increases the thermal performance of both the flat-plate and disk-shaped heat pipes. Do et al. (2010) conducted a one-dimensional conduction analysis considering the thermophysical properties of Al₂O₃ nanofluids and the surface characteristics formed by nanoparticles. It was found that thin porous coating present in the wall material was the key to enhance the heat transfer performance of a flat micro heat pipe.

Leong et al. (2012) conducted an analytical study on the thermal performance of a thermosyphon operated with water and nanofluids. This study found that nanofluid properties such as thermal conductivity play only a minor role in enhancing the thermal performance of the thermosyphon.

1.5 Inference from the literatures

Various kinds of nanofluid were effectively utilized as working fluid in heat pipes. Base fluids (water, methanol, acetone, refrigerants etc) containing metal (Cu, Ag, Au and Al etc) and metal oxide (CuO, Al₂O₃ and TiO₂ etc) nanoparticles were used as working fluids. A few studies (Kang 2006, Naphon 2008) showed that the thermophysical properties of the working fluids were the main factors influencing the performance enhancement in heat pipes. Recent studies (Tsai 2004, Qu 2010) showed that the thin porous coating formed over the evaporator surface of the heat pipe was the key to performance enhancement. The present work is to study the
effect of thin porous coating present on the wall and the wick materials of the heat pipe on the heat transfer performance.

Many attempts have been made in the past to simulate the cylindrical heat pipes, flat heat pipes and micro heat pipes. A complete review of modeling efforts in the heat pipe was presented in a study by Garimella and Sobhan (2001). In most of the studies, the interfacial resistance was not included in the mathematical formulation. Very few studies (Tournier 1994, Vadakkan 2004, Ranjan 2011) incorporate the interfacial resistance in the model. Tournier and El-Genk (1994) included the evaporation/condensation resistance at the interface in the formulation so that the interface pressure differs from the system pressure. Vadakkan (2004) included the interfacial resistance in to the model along with a change in temperature in the vapor core. Also, in this study, an incompressible formulation was incorporated while all other studies mentioned above used a compressible model. In this formulation system pressure build-up with time was accounted. In the present study, to accurately predict the temperature change in the vapor core, a similar model (Vadakkan, 2004) was used. In addition to this, the effect of nanofluid on the performance of heat pipe was studied by incorporating the thermophysical properties of nanofluids and the effect of nanoparticle deposition on the wick.

1.6 Objectives

The main objectives of this research are as follows:-

- To prepare a nanofluid suitable for heat pipe application and study the thermophysical properties of the nanofluids
- To study the performance of the heat pipe with different nanofluids
- To explore the mechanism of heat transfer enhancement while using nanofluid in heat pipes
• To study the performance of heat pipe using a nanoparticle coated wick
• To develop a porous coating on the inner surface of the heat pipe container wall
• To study the performance of the heat pipe with coated container wall
• To develop a CFD model to predict the operating pressure, velocity and temperature profiles

1.7 Organization of thesis

The present thesis is divided into 6 chapters as follows:-
Chapter 1 describes the basic information about heat pipes and their various limitations. It also describes the relevant published literature for the present study. It includes parametric studies of heat pipes, heat pipes using nanofluids as working fluids, thermosyphon using nanofluids and mathematical modeling of heat pipes.
Chapter 2 presents the preparation and characterization of nanofluids. It also describes the nanoparticle coated wick and tube with anodized inner surface.
Chapter 3 provides the details of experimental studies on heat pipes using DI water, nanofluids, and the work with coated wicks and anodized tubes.
Chapter 4 contains the details of mathematical modeling of the heat pipes with relevant information on governing equations and solution methods.
Chapter 5 presents the results and discussion on the relative performance of heat pipes charged with nanofluids and with plain fluids, heat pipes using coated and uncoated wicks, heat pipes with anodized and non anodized tubes.
Conclusion and recommended future work are presented in Chapter 6.