A COMPARATIVE INVESTIGATION OF HEAT TRANSFER CAPACITY LIMITS OF HEAT PIPES

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ABSTRACT

A COMPARATIVE INVESTIGATION OF HEAT TRANSFER CAPACITY LIMITS OF HEAT PIPES

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Heat pipe is a passive two phase device capable of transferring large rates of heat with a minimal temperature drop. It is a sealed tube with a wick structure lined in it and with a working fluid inside the tube. It consists of three parts: an evaporator, a condenser and an adiabatic section. The heat pipes are widely used in electronics cooling and spacecraft applications. Although they can transfer large rate of heat in a short range, they have operating limits, namely: the capillary limit, the viscous limit, the entrainment limit, the sonic limit and the boiling limit. These limits determine the heat transfer capacity of the heat pipe. The properties of the working fluid, the structure of the wick, the orientation of the pipe, the length and the diameter of the tube etc. are the parameters that affect the limits. In this study, an analytical 1-D heat pipe model is formed and a computer code is prepared in order to analyze the effects of the parameters on the heat transfer capacity of a heat pipe. Water, Ammonia and Mercury are investigated as working fluids for different operating temperature ranges. The software is tested for a typical application for each working fluid.

Key Words: Electronics cooling, heat pipe, heat transfer limit

ÖZ

ISI TÜPLERİNİN ISI AKTARIM KAPASİTE SINIRLARININ KARŞILAŞTIRMALI İNCELENMESİ

Küçük, Sinan
Yüksek Lisans, Makina Mühendisliği Bölümü
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Isı tüpü, çok düşük sıcaklık farkı ile çok yüksek ısı transferi sağlayabilen iki fazlı pasif bir araçtır. Isı tüpü içerisinde ağ yapısı bulunan ve içerisinden soğutucu akışkan geçen yalıtılmış bir tüptür. Isı tüpü üç bölümden oluşur; evaporatör, kondenser ve yalıtılmış bölüm. Isı tüpleri elektronik soğutmada ve uzay uygulamalarında yoğun olarak kullanılırlar. Isı tüpleri, kısa mesafede yüksek ısı transferi sağlayabilmelerine rağmen kılcal limit, yapışkan limit, sürükleme limiti, sonik hız limiti ve kaynama limiti olarak adlandırılan operasyonel limitlere sahiptir. Bu limitler ısı tüpünün ısı taşıyabilme kapasitesini belirler. Soğutucu akışkanın özellikleri, ağ yapısının özellikleri, tüpün konumu, tüpün uzunluğu ve çapı gibi değişkenler limitleri etkiler. Bu çalışmada, analitik olarak bir boyutlu bir ısı tüpü modeli oluşturulmuş ve belirtilen değişkenlerin ısı tüpünün ısı taşıyabilme kapasitesi üzerindeki etkilerini analiz etmek için bir bilgisayar kodu hazırlanmıştır. Soğutucu akışkan olarak Su, Amonyak ve Civa farklı çalışma sıcaklık aralıkları için incelenmiştir. Bilgisayar kodu her soğutucu akışkana özel uygulamalarla test edilmiştir.

Anahtar Kelimeler: Elektronik soğutma, ısı tüpleri, ısı taşıma limiti

To my family and my love …

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NOMENCLATURE

Latin Symbols

r	radius (m)
C	correlation coefficient
f	correlation coefficient for friction
Re	Reynolds number
L	length (m), layer number of wick
g	gravitational acceleration (m/s^2)
q	heat (W)
K	permeability of wick
d	diameter (m)
P	pressure (Pa)
R	gas constant (J/K.mol)
T	temperature (K)
q''	heat flux (W/m^2)
S	crimping factor of wick structure
N	number of meshes in wick structure
k	thermal conductivity (W/m.K)
ΔP	pressure difference (Pa)
ΔT	temperature difference (K)
$\frac{\partial P}{\partial x}$	pressure gradient in x direction (Pa/m)
dx	incremental change in x direction
M	Merit number

Greek Symbols

σ	surface tension (N/m)
μ	viscosity (Ns/m^2)
ρ	density (kg/m^3)
λ	latent heat of vaporization (kJ/kg.K)
ψ	tilt angle (°)
γ	ratio of specific heat
π	Pi
ε	porosity of the wick

Subscripts

ce	capillary effective
c	capillary, condenser
e	evaporator, entrainment
b	boiling
s	sonic
eff	effective
ph	phase transition
m	maximum
v	vapor, viscous
hv	hydraulic vapor
w	wick, wall
l	liquid
cr	critical
i	internal
n	critical nucleation site
sat	saturation
+	normal hydrostatic
ll	axial hydrostatic

CHAPTER 1

INTRODUCTION

1.1 Heat Pipes

In recent years, developments in electronics field brought high-speed electronic equipments to our lives. The effects of these developments may be commonly seen in computers. The speed of computers increased tremendously, in the last 10 years. The clock speed of CPUs increased so much that we can use today's desktop computers for almost every demanding computational task of the past. Everything with a benefit comes with a cost. With the developments, the thermal management of these electronic equipments becomes harder and harder. The heat dissipation from functional elements can no longer be done with the conventional approaches like passive heat sinks.

This situation may be better observed in notebook computers. The most important equipment in a computer is the central processing unit (CPU). It is also the most heat dissipating equipment in a computer. Today, an Intel Pentium M CPU dissipates 60 W of heat. If new dual core CPUs are considered, this value becomes 70 W. The heat dissipation problem may be solved more easily in desktop computers. A big size heat sink may be used in a desktop computer coupled with a bigger fan. But this is not the case in notebook computers. The space is limited and the weight impact should be considered. Today, the demand in the notebook computer field is to have notebook computers that are thinner, lighter in weight and as fast as desktop counterparts. Also to be able to increase the battery life, notebooks should consume as less energy as possible. Another problem in notebooks is to transfer the heat generated by the CPU to outside of the notebook.

Transferring heat from CPU surface at the middle of the notebook computer to outside is difficult because of the thin casing structure. The difficulties in thermal management of high performance electronic components or CPU’s force the industry to find a new way of accomplishing sufficient cooling performance from a small area. The simplest way to achieve a higher cooling rate is to move from single phase heat transfer to two phase one using heat pipes.

“A heat pipe is a passive two phase device capable of transferring large amount of heat with minimal temperature drop.” says Peterson [1]. The idea of the heat pipe was first suggested by Gaugler in 1942 [2]. But that device did not take much attention. “Later, in 1964, Grover and his colleagues at Los Alamos National Laboratories published the results of an independent investigation and first applied the term heat pipe. Since that time, heat pipes have been employed in numerous applications ranging from temperature control of the permafrost layer under Alaska pipeline to the thermal control of optical surfaces in spacecraft.” [1].

A heat pipe is simply a sealed tube. The structure is shown in Figure 1.1. There is a wick structure adherent to pipe wall. And the tube is filled with the working fluid which is kept at saturation conditions.

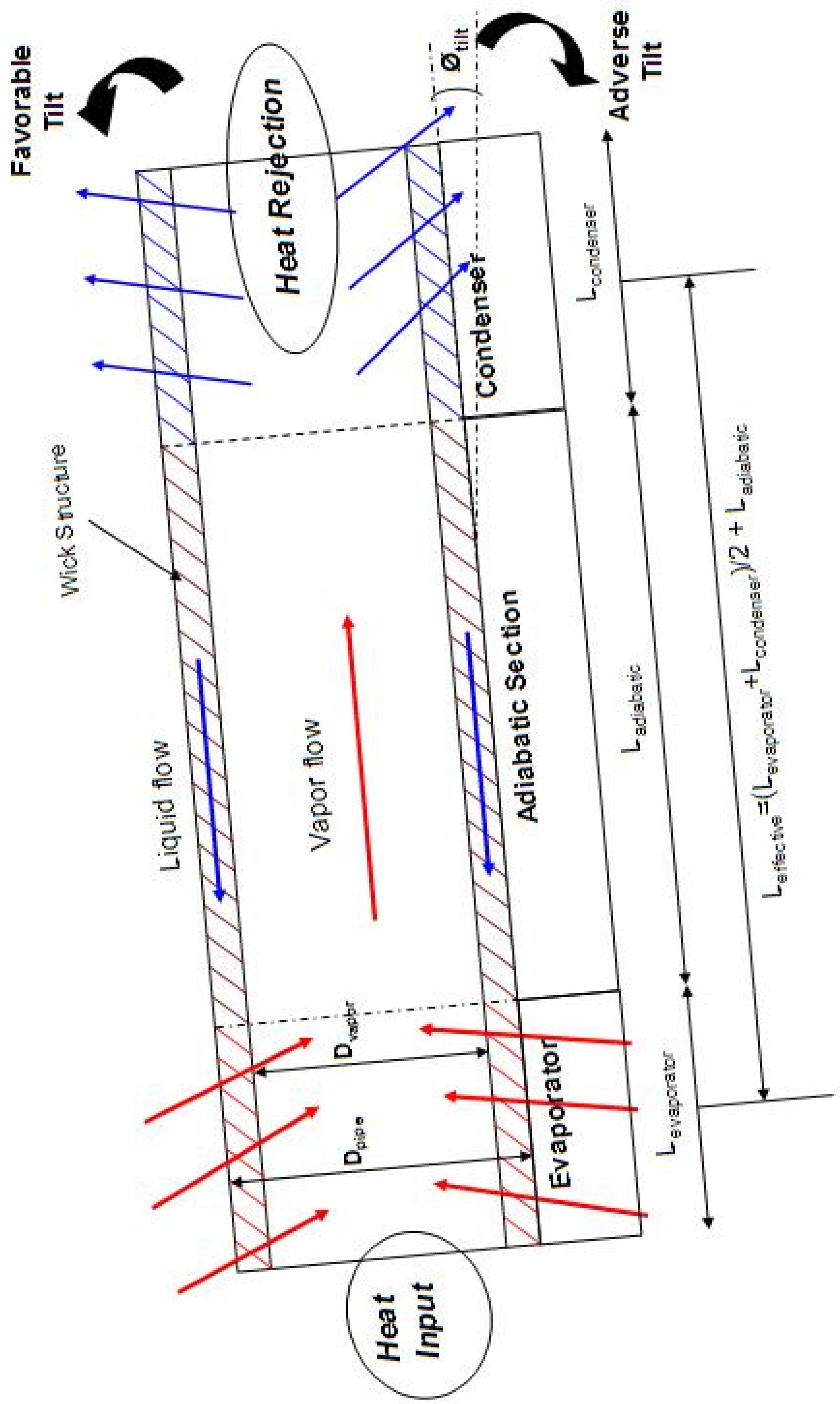


Figure 1.1 Structure of a Heat Pipe

As seen in Figure 1.1, the heat pipe consists of three parts: evaporator, adiabatic section and condenser.

a) Evaporator:

This is the part of the heat pipe where heat is added. Evaporation occurs inside this section.

b) Adiabatic section:

This is the section of the heat pipe in which the coolant exists in two-phase at the same time. The evaporated liquid flows to condenser, and at the same time condensed liquid flows back to evaporator inside wick structure.

c) Condenser:

This is the section of the heat pipe where heat rejection occurs. The heat rejection may be done by natural convection, forced convection or by conduction to another heat sink. The heat rejection causes the working fluid to condense. The condensed fluid enters into wick structure and by capillary forces, flows back to evaporator.

Working principle of the heat pipe is simple. As heat is accumulated into the heat pipe, the working fluid in the wick structure evaporates. Due to evaporation, a pressure difference occurs between two ends of the pipe. The working fluid is driven by this pressure gradient and it flows through the vapor space to the other end of the pipe, which is the condenser. At the condenser end, heat is removed from the heat pipe, causing the working fluid to condense. The condensed liquid goes into the wick structure. Capillary forces cause the liquid to flow back to evaporator and the cycle is completed. There is no activating equipment like a pump in the system. It is simple and consists of only a closed pipe, a wick structure and a working fluid. Therefore it is called passive. Since the coolant exists in two-phase at the same time, it is called two-phase device.

The cross section of a heat pipe can be seen in Figure 1.2 [15]. As seen in the figure, the wick structure is adherent to pipe wall. Various types of wick structure can be used in heat pipes. The wick is important for a heat pipe, since it provides capillary force and this force causes the fluid flow from condenser to evaporator. Some common types of wick structures are shown in Figure 1.3 [15].

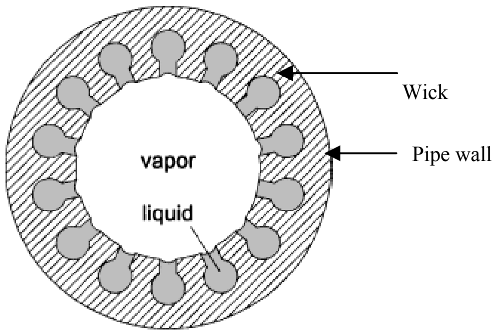


Figure 1.2 – Cross Sectional View of a Heat Pipe with a Groove Wick Structure [Adapted from [15])

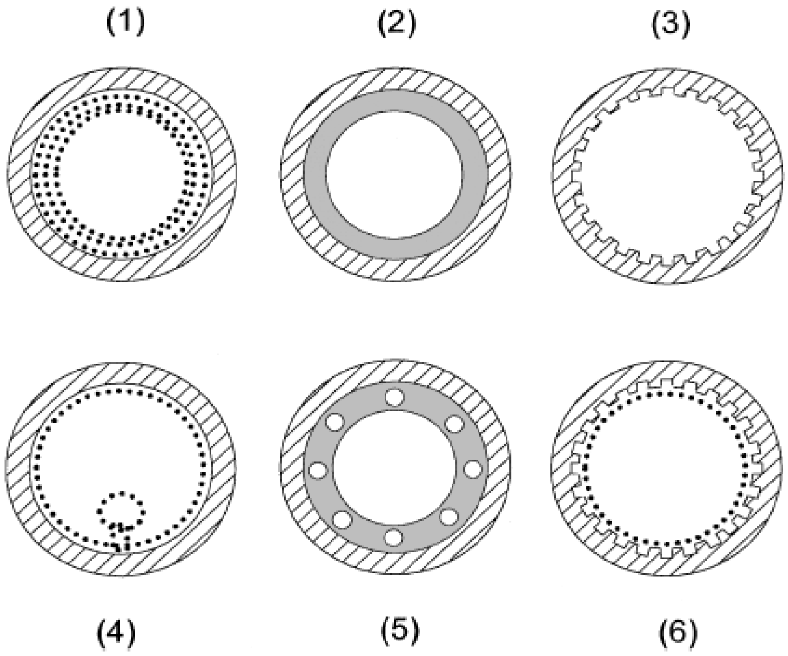


Figure 1.3 – Common Types of Wick Structure (Adapted from [15])
1) Wrapped screen 2) Sintered Metal 3) Open Grooves
4) Artery 5) & 6) Combined Structure

“Heat pipes can be classified in several different ways; by operating temperature range (cryogenic, ambient, or liquid metal), by wick structure (arterial or composite), or by function (rotating/revolving heat pipes, micro heat pipes, variable conductance heat pipes, or thermal diodes).” [1].

1.2 Literature Search and Past Studies

First heat pipe concept was introduced by R.S. Gaugler. Later, at the beginning of 1960s, the concept is introduced by an independent study that is done by G.M. Grover. This study was the first study where the term “heat pipe” was used. Later, in years, the heat pipe concept took much more attention and various studies were done to investigate heat pipes.

Babin, et al. [3] studied the steady-state modeling and testing of a micro heat pipe. A combined experimental and analytical investigation was conducted to identify and understand better the phenomena that govern the performance limitations and operating characteristics of micro heat pipes. Analytical model was developed for five different limits governing the maximum heat transport capacity of a heat pipe; the sonic limit, the entrainment limit, the boiling limit, the viscous limit and the capillary limit. In the experimental portion of the investigation, two micro heat pipes, one copper and one silver, $1mm^2$ cross-sectional areas and 57 mm. in length, were evaluated experimentally to determine the accuracy of the steady-state model [3]. The study is concluded that by comparing the steady-state experimental results, analytical model could predict accurately the experimentally determined maximum heat transport capacity for the operating temperature range of 40^0C to 60^0C. Therefore, it was concluded that the steady-state analytical model can be used to predict accurately the level of performance.

Babin and Peterson [4] experimentally investigated the flexible bellows heat pipe for cooling discrete heat sources. In their study, a computer model was developed to design flexible bellows heat pipe in order to cool small discrete sources or arrays of small heat sources. This model was used to evaluate the operational characteristics and performance limitations of these heat pipes. For experimental procedure, three flexible bellows heat pipes about 40 mm. in length and 6 mm. in diameter were tested using three different wick configurations and different tilt angles. The test pipes were found to be boiling limited over the most of operating temperature range tested. Finally, it was concluded that flexible bellows heat pipe did in fact operate as intended and that the operation could be accurately predicted.

Khrustalev and Faghri [5] studied the thermal analysis of a micro heat pipe. In their study, a mathematical model was developed and the heat and the mass transfer processes in a micro heat pipe were examined. The model in the study was describing the distribution of the liquid in a micro heat pipe and its thermal characteristics depending upon the liquid charge and the applied heat load. The importance of the working fluid fill, minimum wetting contact angle, and the shear stresses at the liquid-vapor interface in predicting the maximum heat transfer capacity and thermal resistance of the micro heat pipe was demonstrated. The predicted results obtained from the model were compared to existing experimental data. It was concluded that shear stresses in the liquid at liquid-vapor interface significantly influenced the maximum heat transfer capacity. Moreover, the dynamic component of the pressure gradient in the liquid had no important effect on the performance and for nearly maximum heat loads, the largest portion of the liquid pressure drop occurred in the evaporator and the beginning of the adiabatic section. Finally, the amount of working fluid and the minimum wetting contact angle strongly influenced the performance characteristics of micro heat pipe.

Nguyen, et al. [6] investigated cooling systems using miniature heat pipe for a mobile PC. Three different cooling systems with heat pipes for a notebook PC were examined in the study. Those were; heat pipe with heat spreader plate, heat pipe with heat sink and fan, hinged heat pipe systems. Experimental tests were done for those three different cooling systems. It was found that for a CPU having surface temperature of 95°C and 40°C ambient temperature, heat pipe with heat spreader plate could dissipate 6 W, whereas other two systems could dissipate 13W.

There are some studies in literature which are based on possible improvements that can be done. One of these studied belongs to Moon, et al. [7]. In the study, they tried to improve the thermal performance of a miniature heat pipe used for a notebook PC. A miniature heat pipe was used by reducing the cross-sectional area of the pipe about 30 % of original and the diameter was pressed from 4mm. to 2 mm. A test of miniature heat pipe was performed in order to review the thermal performance by varying pressed thickness, total length of pipe, wall thickness, heat flux and inclination angle. Moreover, new wick types were considered in the study. As a result of the study, the limitation of the pressed thickness of the heat pipe with woven wire wick was found to be 2 – 2.5 mm. It was also concluded that the thermal resistance and heat transfer limit of miniature heat pipe could be improved about 10 % by reducing wall thickness from 0.4 mm. to 0.25 mm. As final decision, it was mentioned that heat pipe with composite wick of woven/straight wire had the lower thermal resistance and larger heat transfer limit than those of the heat pipe with circular woven wire wick.

Another experimental study about micro heat pipes was done by Moon, et al. [8]. In the study, micro heat pipes with cross-section of polygon were manufactured and tested for operating characteristics and heat transfer limits. The micro heat pipes tested in the study had triangular cross-section with curved sides and rectangular cross-section with curved sides. The used heat pipe was a copper-water heat pipe. In the experiment, a vacuum chamber was used and the operating temperatures were from 60°C to 90°C. As a result, it was found that the heat transfer limit of triangular micro heat pipe was 1.6 times larger than rectangular heat pipe.

Heat pipes are widely used in cooling PC CPU’s. A study about heat pipe cooling technology for a desktop PC CPU was done by Kim, et al. [9]. They investigated the performances of conventional aluminum heat pipe and a remote heat exchanger with heat pipe. Three heat pipes with woven wire wick were used in the heat pipe cooling module. Water was the working fluid of heat pipes. Performance tests were done using two cooling systems. The study concluded that heat sink cooling module had most excellent thermal performance at high fan speed, whereas heat pipe cooling module had better thermal performance at lower fan speeds. Therefore, heat pipe cooling module could be applied as a system of low acoustic noise and high performance, solving the problem of acoustic noise of conventional heat sink modules.

Berre, et al. [10] studied the silicon micro heat pipes for electronics cooling experimentally. In their study, an experimental investigation was conducted to determine the thermal behavior of micro heat pipe arrays made of silicon. Two different types of heat pipes are used in experiments, one with triangular channels 230 µm. wide, 170 µm. deep, and other with triangular channels 500 µm. wide and 340 µm. deep, coupled with arteries. Two working fluids were tested, ethanol and methanol. Different fill charges between 0% and 66% were used. At the end of the study, it was shown that with liquid arteries, maximum 300% improvement in effective thermal conductivity at high heat flux was reached.

Another study on micro heat pipe was done by Suman, et al. [11]. A model of the capillary limit of a micro heat pipe was formed and the dry-out length was predicted in the study. A model for fluid flow and heat transfer in a micro heat pipe of any polygonal shape was presented. Two different heat pipes, rectangular and triangular were considered as case studies. The performance and capillary limitations of such a device were analyzed in detail. The profile of radius of curvature was used to predict the onset of dry-out point and dry-out length. Moreover, a method was developed to estimate the dry-out length as a function of system geometry and variables like heat input and inclination. It was found that the dry-out length increased with increase in heat input and inclination. Finally, the results predicted by model were compared with experimental results and it was shown that the model was validated with experimental results.

Loh, et al. [12] made a comparative study of heat pipes performances in different orientations. An experimental setup was prepared and three heat pipe wick structures and their performance with changes in the orientation were examined. The influence on the performance of the heat pipes with changes in the orientation angle from -90° to +90° was investigated. In the study, three different diameters (4 mm., 5 mm. and 6 mm.) were used and each of three heat pipe were tested with three different wick structures (sintered powder metal, groove and mesh). The study was concluded that heat source orientation and gravity had less effect on sintered powdered metal heat pipes due to the fact that the sintered powder metal wick had the strongest capillary action when compared to groove or mesh wick structures. It was mentioned that it was not desirable to use groove or mesh heat pipes when evaporator was on the top of the condenser (adverse tilt). Finally, for 6 mm. diameter heat pipe, the groove heat pipe had better thermal performance than mesh and sintered powder metal in +90° to 0° range. It was also shown in the study that by increasing tilt angle, from horizontal (0°) to evaporator at top condition (+90°), the thermal performance of all heat pipes used in the study decreased.

Finally, different working fluid performances were compared in the study of Bernardin [13]. He investigated the performance of methanol and water heat pipes for electronics cooling applications. Thermal analysis and performance testing of commercial copper heat pipes that utilize a sintered copper wick with methanol and water as working fluid were discussed in the study. The analytical thermal performance limitations of heat pipes were presented and empirical results were used to assess the thermal performance of heat pipes, determine the range of variations between heat pipes and assess the accuracy of analytical performance models. It was shown that the methanol heat pipes possessed a smaller maximum heat transport rate than the water heat pipes, in the range of 0°C to 30°C. Finally, of the five analytical performance limit models presented in the study, the capillary limit model appeared to predict the maximum heat transport rate for methanol heat pipes over the temperature range investigated. For water heat pipes, both the entrainment and capillary limit models appeared to satisfactorily predict the maximum heat transport rate, but it was not clear from data and modeling comparison which one was the governing limit.

1.3 Thesis Objective and Organization

The aim of this study is to analyze the performance characteristics of the heat pipes. The limits that govern the heat transfer capacity of the heat pipe are investigated. Parameters that affect those limits are examined. For this purpose, a computer code using MATLAB program is prepared. The code calculates the heat transfer capacity of a specified heat pipe for the capillary limit, the entrainment limit, the viscous limit, the sonic limit and the boiling limit. As a result, for specified conditions, it gives the governing limit and the corresponding heat transfer capacity of the heat pipe with specified properties.

The code can be used to perform the heat pipe analyses using water, ammonia and mercury as working fluids. The thermo physical properties of these fluids are entered into the database of the code.

CHAPTER 2

LIMITS OF HEAT PIPES, ANALYTICAL MODELS AND COMPUTER CODE TO CALCULATE LIMITS

Heat pipes are devices that can transfer large amount of heat with small temperature differences between evaporator and condenser parts. This property can be seen better when a heat pipe is compared with an aluminum or a copper rod. For example, 1.27 cm. long aluminum rod, copper rod and a heat pipe can be used to transfer 20 W of heat over over a 50 cm. distance [17]. When these three methods are compared, 460°C temperature difference should be satisfied for aluminum, 205°C temperature difference should be satisfied for copper, and only 6°C of temperature difference should be satisfied for heat pipe [17]. Nevertheless, heat pipes can not operate in any condition, without being affected by anything. There are some factors that the performance of the heat pipe depends on. The heat pipe has a limited heat transport capacity, and this capacity is determined by various factors like working fluid properties, wick structure properties, working conditions etc. There are some governing limits like capillary limit, viscous limit, entrainment limit, sonic limit and boiling limit for the maximum heat transport capacity of a heat pipe.

2.1 Heat Transfer Limits

The heat pipe's heat transfer limit is the maximum heat that can be transferred by heat pipe under nearly isothermal conditions. If the heat added to heat pipe exceeds the limit, than the vapor-liquid cycle inside heat pipe becomes disturbed and at the end dry-out can occur in evaporator section. With the dry out, the evaporator can no longer be supplied with liquid and the heat pipe cannot continue to transfer heat. The heat transfer limit depends on operating temperature, thermophyscal properties of working fluid and the wick structure. The heat transfer limits of heat pipe are;

- Capillary Limit
- Viscous Limit
- Entrainment Limit
- Sonic Limit
- Boiling Limit

For all operating temperatures, and environmental conditions, one of those limits is the determining factor for the performance of the heat pipe. For example, at lower operating temperatures, viscous limit becomes important but at normal operating temperatures, capillary limit is more important and at some higher temperatures, other limits become important for the performance of the heat pipe. The transition points between these limits depend on the type of working fluid used in the heat pipe.

2.1.1 Capillary Limit

The vapor and liquid flow in a heat pipe are provided by the pressure difference due to evaporation and by the capillary pumping pressure respectively. There are some pressure losses occurring throughout the vapor and liquid flows. For a heat pipe to function properly, the net capillary pumping pressure between evaporator and condenser regions must be greater than these pressure losses. Otherwise, insufficient liquid flow to evaporator region causes dry-out in the evaporator.
The losses in the liquid and vapor flows are [1];

- Inertial and viscous pressure drops occurring in vapor phase
- Inertial and viscous pressure drops occurring in liquid phase
- Pressure losses occurring in phase transition in evaporator
- Pressure losses occurring in phase transition in condenser

- Normal hydrostatic pressure losses
- Axial hydrostatic pressure losses

Using these losses, the capillary limit for a heat pipe is expressed in Equation 2.1 [1].

$$(\Delta P_c)_m \geq \int_{Leff} \frac{\delta P_v}{\delta x} dx + \int_{Leff} \frac{\delta P_l}{\delta x} dx + \Delta P_{ph,e} + \Delta P_{ph,c} + \Delta P_{+} + \Delta P_{ll} \qquad (2.1)\ [1]$$

Where ;

$(\Delta P_c)_m$ = maximum capillary pressure difference generated within capillary wicking structure between wet and dry points

$\frac{\delta P_v}{\delta x}$ = sum of inertial and viscous pressure drops occurring in vapor phase

$\frac{\delta P_l}{\delta x}$ = sum of inertial and viscous pressure drops occurring in liquid phase

$\Delta P_{ph,e}$ = pressure gradient across phase transition in evaporator

$\Delta P_{ph,c}$ = pressure gradient across phase transition in condenser

ΔP_{+} = normal hydrostatic pressure drop

ΔP_{ll} = axial hydrostatic pressure drop

The expression of the capillary limit can be formulated by some assumptions. These assumptions are;

a) For vapor phase, the viscous pressure losses are taken into account and the inertial effects are neglected.
b) The vapor phase transition losses are neglected.
c) The vapor flow is assumed as one dimensional in axial direction.

Using these assumptions, the expression for capillary limit can be obtained as shown in Equation 2.2 [1].

$$\frac{2\sigma}{r_{ce}} \geq \left(\frac{C.f_v.\mathrm{Re}_v.\mu_v}{2.r_{hv}^{2}.A_v.\rho_v.\lambda} L_{eff}.q\right) + \left(\frac{\mu_l}{K.A_w.\lambda.\rho_l} L_{eff}.q\right) + (\rho_l.g.d_v.\cos\psi) + (\rho_l.g.L_{eff}.\sin\psi) \quad (2.2)\ [1]$$

Where;

σ : Surface tension of the fluid

r_{ce} : Effective capillary radius of the wick

C : Correlation coefficient

f_v : Correlation coefficient for friction

Re_v : Reynolds number for vapor

μ_v : Viscosity of the vapor

r_{hv} : Hydraulic radius of the vapor space

A_v : Area of the vapor space

ρ_v : Density of the vapor

λ : Latent heat of vaporization

L_{eff} : Effective length of the pipe

μ_l : Viscosity of the liquid

K : Permeability of the wick

A_w : Area of the wick

ρ_l : Density of the liquid

g : Gravity

d_v : Diameter of the vapor space

ψ : Tilt angle

q: Heat transfer rate

This formula is derived using conservation of momentum equations. The control volume used in derivation and the steps of the derivation can be found in [1].

In above formula, capillary pumping pressure and other pressure losses are represented by;

Capillary pumping pressure = $\frac{2\sigma}{r_{ce}}$

Viscous pressure losses in vapor = $\frac{C.f_v.\mathrm{Re}_v.\mu_v}{2.r_{hv}^2.A_v.\rho_v.\lambda} L_{eff}.q$

Inertial and viscous pressure losses in liquid = $\frac{\mu_l}{K.A_w.\lambda.\rho_l} L_{eff}.q$

Normal hydrostatic pressure losses = $\rho_l.g.d_v.\cos\psi$

Axial hydrostatic pressure losses = $\rho_l.g.L_{eff}.\sin\psi$

The maximum heat transport capacity governed by the capillary limit can be found by rearranging the Equation 2.2;

$$q \le \frac{\frac{2\sigma}{r_{ce}} - \rho_l.g(d_v.\cos\psi + L_{eff}.\sin\psi)}{\left(\frac{C.f_v.\mathrm{Re}_v.\mu_v}{2.r_{hv}^{\ 2}.A_v.\rho_v.\lambda} + \frac{\mu_l}{K.A_w.\lambda.\rho_l}\right).L_{eff}} \tag{2.3}$$

Investigating Equation 2.3, it is obvious that the properties of working fluid and the wick structure are the main factors affecting the capillary limit of a heat pipe. Attention shall be paid in choosing working fluid and wick structure type in order to have desired heat transport capacity for capillary limit. The capillary limit is effective for the whole sections of a heat pipe.

In general, the thermophysical properties of working fluid, the properties of wick structure and the compatibility of fluid with pipe and wick structrures are important factors for heat pipe. However, for capillary limit, the important properties of the working fluid are latent heat of vaporization, surface tension and liquid viscosity. These numbers are collected by Chi (1976) [1] under one term which is referred as liquid transport capacity or liquid figure of merit, M. This is;

$$M = \frac{\rho_l \sigma \lambda}{\mu_l} \qquad (2.4)\ [1]$$

The Merit number can be used as a reference for preliminary selection of working fluid. As seen from Equation 2.4, as the thermophysical properties like surface tension, latent heat of vaporization and liquid density of working fluid are higher, the merit number becomes higher, meaning that the fluid is more appropriate for the selected condition. Therefore, for the working fluid selection, fluids with higher merit numbers in selected operating range should be chosen.

Another important parameter is wick structure for the operation of a heat pipe. The wick structure is the way in which liquid flows from condenser to evaporator. Low resistance is desirable for the fluid flow. Therefore, the permeability of the wicking structure must be high. However, the capillary pumping pressure is also provided by wick. And to have high capillary pumping pressure, wick structures with small pore sizes are desirable. The maximum capillary pumping pressure provided by the wick increases with decreasing pore size. On the other hand, wick permeability increases with increasing pore size. Therefore, this concept is a trade off concept. The optimum pore size and permeability must be selected for the wick structure. Other parameters for the wick are the compatibility with working fluid and the thermal conductivity.

2.1.2 Viscous Limit

Viscous limit depends on the viscous pressure losses in vapor phase and the vapor pressure of the working fluid. The vapor pressure is the force that drives the vapor from evaporator to condenser. When the viscous losses in vapor phase are larger than the vapor pressure, than the vapor can not be driven to condenser and the heat pipe can not operate. This limit becomes effective at lower temperatures where the vapor pressure of the working fluid is very low.

Mathematically, the viscous limit can be expressed as;

$$q_v = \frac{A_v.r_v^{\ 2}.\lambda.\rho_v.P_v}{16.\mu_v.L_{eff}} \qquad (2.5)\ [1]$$

Where;

P_v : Vapor pressure

Viscous limit is effective for the whole sections of the heat pipe, since it depends on the vapor pressure difference between evaporator and condenser sections.

2.1.3 Entrainment Limit

In a heat pipe, two phases of the working fluid – liquid and vapor - exists at the same time. In the adiabatic section of the heat pipe, vapor flow to condenser occurs in the hole of the pipe and at the same time, liquid flow to evaporator occurs in the wick structure. This two phase flow results in an interface between liquid and vapor flows.

Throughout the simultaneous two phases flow, due to counter flow of liquid and vapor, viscous shear forces occurs at the liquid-vapor interface. At the operation of heat pipe, if these viscous shear forces become larger than the liquid surface tension forces, the liquid particles will be forced to be added into vapor flow. This condition is called as the entrainment limit. This back flow of liquid to condenser causes liquid loss in evaporator and finally dry-out occurs at the evaporator. Dry-out means no liquid exists in evaporator and results in failure of heat pipe.

The mathematical expression presented by Cotter (1967) [1] for entrainment limit is;

$$q_e = A_v.\lambda.(\frac{\sigma.\rho_v}{2.r_{hw}})^{1/2} \qquad (2.6)\ [1]$$

Where;
r_{hw} : Hydraulic radius of the wick

When the equation for entrainment limit is investigated, it is obvious that as the surface tension of the fluid is increased, the limit increases, too. This is expected because in order to have liquid be added into vapor, viscous shear forces between liquid and vapor shall be higher than the surface tension forces of the liquid. Therefore, as the surface tension of the fluid increases, it becomes harder and harder to force the liquid be added into vapor flow. The entrainment limit is effective at the adiabatic section of a heat pipe, where the liquid and vapor flow occurs at the same time.

2.1.4 Sonic Limit

The sonic limit occurs when the vapor flow velocities approaches to sonic flow velocities at the exit of the evaporator. As the velocity of the vapor increases at evaporator exit, the flow reaches choked flow conditions and the mass flow of the vapor is limited with the critical mass flow at choked flow conditions. Therefore, the heat transfer capacity becomes limited and higher heat values can not be transferred. Unlike other limits, sonic limit does not cause a failure like dry-out in heat pipe. It only limits the capacity of the heat pipe. In heat pipes, as the operating temperature decreases, the density and the vapor pressure of the working fluid decreases. As a result, it is expected that the heat transport capacity of the heat pipe for sonic limit decreases, since the flow becomes closer to choked flow conditions. Therefore, sonic limit is effective at lower operating temperatures. Like capillary and viscous limits, sonic limit is also effective for the entire heat pipe.

The mathematical formula for the sonic limit is found by Levy (1968) [1] as follows;

$$q_s = A_v.\rho_v.\lambda.\left(\frac{\gamma_v.R_v.T_v}{2(\gamma_v + 1)}\right)^{1/2} \qquad (2.7)\ [1]$$

Where;

γ_v : Ratio of specific heat

R_v : Gas constant

T_v : Vapor temperature

For Equation 2.7 to be valid, there are some assumptions made. These are:

- One dimensional flow is assumed.

- Frictional effects are neglected.
- It is assumed that inertial effects dominate.
- Vapor flow is assumed as ideal gas flow.

Using above assumptions, the Equation 2.7 is derived beginning from conservation of energy and momentum equations. The details of derivation can be found in [1].

2.1.5 Boiling Limit

The final limitation for the operation of heat pipes is the boiling limit. This limit is caused by the bubble formation in the wick structure in evaporator section. In the evaporator section of the heat pipe, it is assumed that nucleate boiling occurs during the evaporation process. However, as the heat input increases, the boiling exceeds nucleate boiling bounds and film boiling occurs. This film boiling condition avoids liquid return to evaporator section and insufficient liquid flow into evaporator causes dry out of the evaporator section. For the determination of boiling limit, two separate approaches based on nucleate boiling theory are used in literature to determine the boiling limit; bubble formation and collapse of the bubbles.

The important parameters for bubble formation theory are the number and size of the nucleation sites and the temperature difference between fluid and pipe wall. The maximum heat flux can be expressed as;

$$q^{"} = (\frac{k_{eff}}{T_w})\Delta T_{cr} \qquad (2.9)\ [1]$$

Where;

k_{eff} : Effective thermal conductivity of the liquid-wick combination

ΔT_{cr} : Critical superheat

The critical superheat is defined by Marcus (1972) [1] as;

$$\Delta T_{cr} = (\frac{T_{sat}}{\lambda \rho_v})(\frac{2\sigma}{r_n} - \Delta P_{i,\mathrm{m}}) \qquad (2.10)\ [1]$$

Where;

r_n : Critical nucleation site radius

T_{sat} : Saturation temperature

$\Delta P_{i,\mathrm{m}}$: Maximum internal pressure difference

"The growth or collapse of a given bubble once established on a flat or planar surface is dependent upon the liquid temperature and corresponding pressure difference across the liquid vapor interface caused by the vapor pressure and surface tension of the liquid." [1]. Chi (1976) [1] expressed a formula for the heat flux depending on the bubble collapse. This formula is;

$$q_b = (\frac{2.\pi.L_e.k_{eff}.T_v}{\lambda.\rho_v.\ln(r_i / r_v)})(\frac{2\sigma}{r_n} - \Delta P_{c,m}) \qquad (2.11)\ [1]$$

Where;

L_e : Evaporator length

T_v: Vapor temperature

r_i: Inner radius of the pipe

r_v : Radius of the vapor space

r_n: Critical nucleation site radius

$\Delta P_{c,m}$: Maximum capillary pressure difference

When Equation 2.11 is investigated, it can be seen that it mostly depends on the fluid properties.

The boiling limit is effective for the evaporator section of the heat pipe where boiling process occurs. The equation expressed by Chi (1976) [1] is used in this study in order to evaluate the boiling limit.

Above, the five governing limits for the heat transport capacity of the heat pipe are described. All these limits are the governing limits at different conditions. Depending on the fluid properties, operating temperature and wick structure properties, one of these limits govern the heat transport capacity of the heat pipe. The transition points between limits depend on the fluid and wick structures.

2.2 The MATLAB Code

For a heat pipe to operate properly, all limits should be investigated and heat pipe design should be done considering the resulting heat transport capacities. If the design is done according to one limit, unexpected failure cases can occur and heat pipe may not perform properly at different conditions. Or this may cause the failure of heat source (i.e. computer CPU) since heat pipe may not be sufficient to cool the heat source.

For a heat pipe, at different operating temperatures, different limits become dominant. At low operating temperatures, viscous and sonic limits are dominant. As the temperature increases, entrainment limit takes the scene and as temperature continues to increase, capillary and boiling limits become important.

The transition points between limiting conditions depend on the type of fluid and wick structure used. Working fluid thermophysical properties are main factors for the performance of a heat pipe. Effective capillary radius, porosity and permeability properties are important for wick structure. Other effective factors are the physical properties of the pipe and the orientation. The outer diameter, wall thickness and the length of the pipe, evaporator and condenser section lengths have various effects on the performance of a heat pipe. The operating orientation is also important. The gravity can be used to increase the heat transport capacity or it may be the opposing force. If the pipe operates in a horizontal position, gravity has no effect on the performance. When it is decided to design a heat pipe, all these factors shall be considered.

A computer code using MATLAB program is written to investigate different heat transfer capacities of a heat pipe. The code simply consists of a main routine, called as “limits”, which includes sub-routines for all limiting conditions. When the main routine is called in the program, it starts to run the sub routines in order to calculate heat transport limits called as capillary, viscous, sonic, boiling and entrainment. The program code is presented in Appendix.

In order to perform all these calculations, the program needs some information as input. A graphical user interface (GUI) is composed of necessary parameters. The appearance of the GUI is shown in Figure 2.1.

Figure 2.1 – Appearance of the "Inputs" GUI

The important parameters for the calculation of the heat transport capacity depend on working fluid and wick structure. The GUI has input cells for these parameters. GUI has the following input cells;

- d; diameter of the heat pipe
- t; wall thickness of the pipe
- fi; tilt angle
- L; length of the adiabatic section of the heat pipe
- Levap; length of the evaporator of the heat pipe
- Lcon; Length of the condenser of the heat pipe
- rn; nucleation radius
- T; temperature
- Fluid type

- Wick type
- N; number of mesh screen (for mesh screen wick only)
- Lay; number of layers (for mesh screen wick only)
- dw; wire diameter (for mesh screen wick only)
- kw; thermal conductivity of wick
- w; groove width (for groove wick only)
- dep; groove depth (for groove wick only)
- Ng; number of grooves (for groove wick only)

The important assumption for the input information is:
- It is assumed that the lengths of evaporator and condenser sections entered in GUI provide sufficient surface areas at evaporator and condenser sections for the heat transfer.

As seen from the list, only one cell called as "type of fluid" exists for fluid type selection. The program runs a sub-routine that calls fluid thermo physical properties according to fluid type entered in input GUI. The fluid properties used in calculations are presented in Appendix part. Like fluid properties, wick structure properties are also calculated using the input properties by a sub-routine. Therefore, when necessary information is provided via input GUI, the wick properties are calculated inside the program and inputs are formed for the calculation of operating limits. Using the input, the wick porosity and the permeability of the wick are calculated using the formulations for open grooves and wire screen meshes. The properties of the wick structure that are calculated inside the program are;

Porosity;

a) For wire screen meshes;

$$\varepsilon = 1 - \frac{\pi . S . N . d_w}{4} \qquad (2.12)\ [1]$$

Where;
ε : Porosity of the wick

S : Crimping factor of wick structure

N : Mesh number of the wick

d_w : Wire diameter of the wick

b) For open grooves;

$$\varepsilon = w/(w + w_f) \tag{2.13) [1]}$$

Where;

w: groove width

w_f: distance between grooves

<u>Permeability;</u>

a) For wire screen meshes;

$$K = \frac{d_w{}^2 . \varepsilon^3}{122(1-\varepsilon)^2} \tag{2.14) [1]}$$

Where;

ε : Porosity of the wick

d_w : Wire diameter of the wick

K : Permeability of the wick

b) For open grooves;

$$K = \frac{2.\varepsilon . rh^2}{f \, \mathrm{Re}_{l,h}} \tag{2.15) [1]}$$

Where;

ε : Porosity of the wick

rh : Hydraulic radius of groove

f : Non-dimensional friction coefficient

$\mathrm{Re}_{l,h}$: Axial Reynolds number for hydraulic radius

Thermal Conductivity;

a) For wire screen meshes

$$k_{eff} = \mathrm{k_l}.\frac{[(\mathrm{k_l}+\mathrm{k_w})-((1-\varepsilon).(\mathrm{k_l}-\mathrm{k_w}))]}{[(\mathrm{k_l}+\mathrm{k_w})+((1-\varepsilon).(\mathrm{k_l}-\mathrm{k_w}))]} \qquad (2.16)\ [1]$$

Where;

k_{eff} : Effective thermal conductivity

$\mathrm{k_l}$: Liquid thermal conductivity

$\mathrm{k_w}$: Wick thermal conductivity

ε : Porosity of the wick

b) For open grooves;

$$k_{eff} = \frac{[(\mathrm{w_f}.\mathrm{k_l}.\mathrm{k_w}.\mathrm{dep})+\mathrm{w}.\mathrm{k_l}.(0.185.\mathrm{w_f}.\mathrm{k_w}+\mathrm{dep}.\mathrm{k_l})]}{[(\mathrm{w}+\mathrm{w_f}).(0.185.\mathrm{w_f}.\mathrm{k_w}+\mathrm{dep}.\mathrm{k_l})]} \qquad (2.17)\ [1]$$

Where;

k_{eff} : Effective thermal conductivity

$\mathrm{k_l}$: Liquid thermal conductivity

$\mathrm{k_w}$: Wick thermal conductivity

ε : Porosity of the wick

w_f: distance between grooves

w: width of groove

dep: depth of groove

Effective Capillary Radius;

a) For wire screen meshes

$$r_{ce} = \frac{1}{2N} \qquad (2.18)\ [1]$$

Where;

r_{ce} : Effective capillary radius of the wick

N : Mesh number of the wick

The wire diameter (d_w) depends on the mesh number of the wire screen. The mesh numbers and corresponding wire diameters used in calculation are given in Table 2.1 [1].

b) For open grooves;

$$r_{ce} = w \qquad (2.19)\ [1]$$

Where;

r_{ce} : Effective capillary radius of the wick

w : Width of groove

Table 2.1 – Some Mesh Numbers and corresponding Wire Diameters for Wire Screen Mesh Structures

Mesh / in.	Mesh / m	dw (in.)	dw (m)	P_c (Pa)	ϵ	K
8.0000	314.96	0.0280	0.00071	89.32	0.8153	6.58×10^{-8}
12.0000	472.44	0.0230	0.00058	133.98	0.7724	2.49×10^{-8}
16.0000	629.92	0.0180	0.00046	178.65	0.7625	1.35×10^{-8}
18.0000	708.66	0.0170	0.00043	200.98	0.7477	1.00×10^{-8}
24.0000	944.88	0.0140	0.00036	267.97	0.7229	5.10×10^{-9}
30.0000	1181.10	0.0135	0.00034	334.96	0.6660	2.55×10^{-9}
40.0000	1574.80	0.0100	0.00025	446.61	0.6701	1.46×10^{-9}
50.0000	1968.50	0.0090	0.00023	558.27	0.6289	7.74×10^{-10}
60.0000	2362.20	0.0075	0.00019	669.92	0.6289	5.37×10^{-10}
80.0000	3149.60	0.0055	0.00014	893.23	0.6371	3.14×10^{-10}
100.0000	3937.00	0.0045	0.00011	1116.53	0.6289	1.93×10^{-10}

After necessary values are entered into the program using input GUI, the main routine called "limits" is ready to run. It takes the input values entered, starts other sub-routines for the calculation of operating limits. The sub-routines use these input values to calculate the limits according to formulations written inside them.

Various combinations of the parameters can affect the heat pipe performance. But in the purpose of seeing individual effects of these factors, others than the selected parameter is kept constant. For example, if the effect of pipe length on the capacity is investigated, other parameters like pipe diameter, working fluid and wick structure, orientation etc. are kept constant. By this way, the effect of the pipe length can be seen.

All these calculations are done using the formulas that are mentioned in heat pipe limits chapter. These formulas are entered into sub routines of the program. Necessary correlations also take place inside sub routines.

The flow chart of the code is shown in Figure – 2.3.

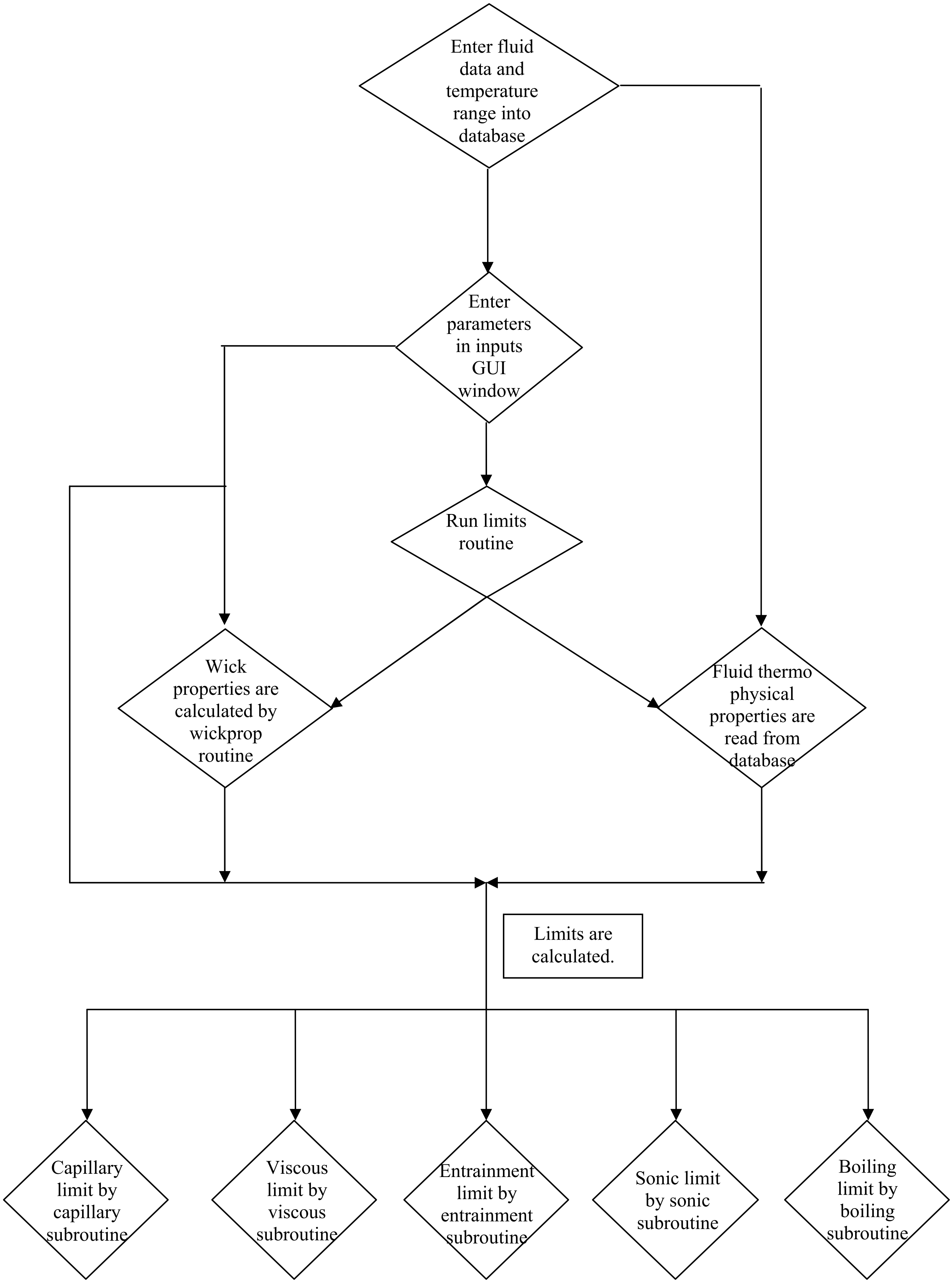


Figure 2.2 – Flow chart of the Computer Code

The inputs of the program are;

- Thermo physical properties of working fluid
- Geometrical properties of the pipe (length, diameter)
- Wick structure properties
- Temperature
- Orientation of the pipe (tilt angle)

The outputs of the program are;

- Maximum heat transfer value of the specified heat pipe according to the governing limit

Moreover, the results of all limits are kept at database in capillary, boiling, viscous, sonic and entrainment data.

CHAPTER 3

RESULTS AND DISCUSSION

The heat transport capacity of a heat pipe depends on various parameters. These parameters used in calculations and their ranges are shown in Table 3.1.

Table 3.1 – Parameters Affecting The Heat Pipe Performance

Wick Types	**Wire Screen Mesh**	**Open Grooves**	
Mesh Number	8 - 100 mesh/inch		
Wire Diameter	1.143×10^{-4} - 7.1×10^{-4}		
Groove Number		5 - 30	
Groove Width		0.00025 - 0.001	
Groove Depth		0.00025 - 0.001	
Fluid Type	**Water**	**Ammonia**	**Mercury**
Temperature Range (K)	273 - 373	196 - 390	473 - 1023

The analysis can be done for all parameters presented in Table 3.1. To see the direct effect of a parameter, the others shall be kept constant. Specific values for the parameters can be entered into input GUI and the result of the calculation according to given specific parameters can be obtained. Therefore, the program can be used for two purposes;

a) General investigation to see the effects of various parameters on heat transfer capacity of a heat pipe
b) To obtain the heat transfer capacity of a heat pipe for specific conditions

As seen in the Table 3.1, two common types of mesh structures are used. Water, ammonia and mercury are used as working fluids.

3.1 Analysis for Wire Screen Meshes

First analyses are performed for wire screen meshes, selected as wick structure. Three types of working fluids are used; Water, Ammonia and Mercury. Therefore, three different types of heat pipe materials are used in the analyses. These are Copper, Aluminum and Steel. The same material, for example Copper, can not be used with Ammonia or Mercury, since it is not compatible with those fluids and usage will create corrosion and chemical interactions between working fluid and the pipe material. Therefore Water-Copper, Ammonia-Aluminum and Mercury-Steel pipes are used in the analyses. In this section, the results of the analyses for wire screen meshes are presented.

The heat pipes used in the analyses have same properties except working fluid and the pipe and wick material. The properties of the heat pipe configuration that is used for first analysis are shown in Table 3.2

Table 3.2- Heat Pipe Configuration as the Baseline

Diameter	0.020 m.
Length	0.75 m.
Evaporator length	0.25 m.
Condenser length	0.25 m.
Adiabatic section length	0.25 m.
Wall thickness	0.0025 m.
Pipe and wick material	Copper for Water Aluminum for Ammonia Steel for Mercury
Orientation	Horizontal
Wick Structure;	Wire screen
Number of meshes	100
Wire diameter	1.143×10^{-4} m.
Number of layers	3

The first configuration is formed using these properties. The temperature scale is different for working fluids, since the operating ranges of the working fluids are different. The operating range of the working fluid is between its melting temperature and the critical temperature. A fluid can be used between these temperatures. However this does not mean that the selected working fluid shows good performance in all of that range. Therefore, the useful operating range is generally smaller than the whole temperature range between melting point and the critical point.

In the analyses, the temperature range for Water is selected from 273 K (0 degrees Celsius, Psat = 611 Pa) to 373 K (100 degrees Celsius Psat = 101.4 kPa). This is the temperature range with the melting and boiling points of water. It is obvious that the melting point is the lower temperature, and boiling point is selected as maximum since the saturation pressure is equal to atmospheric pressure. At higher temperatures, sealing problems may occur since the internal pressure of the pipe will be much higher than atmospheric pressure. Moreover, this range is the operating temperature range for most of the electronic equipments.

The temperature range of the Ammonia is selected as 196 K (Psat= 6.3 kPa) - 395 K (Psat= 9.4 MPa). It is selected to see the performance of the heat pipe for lower temperatures where water can not be used, and in order to make a comparison of Ammonia and Water in the temperature range of Water. Finally, Mercury range is selected as 473 K (Psat=1000 Pa) – 1023 K (Psat=6.3 MPa). This is selected to see the performance of high temperature heat pipes, where Water and Ammonia can not be used. Using these temperature ranges and the specified heat pipe properties, analyses are performed for three types of working fluids.

The results of the first analysis for Water can be seen in the Figure 3.1.

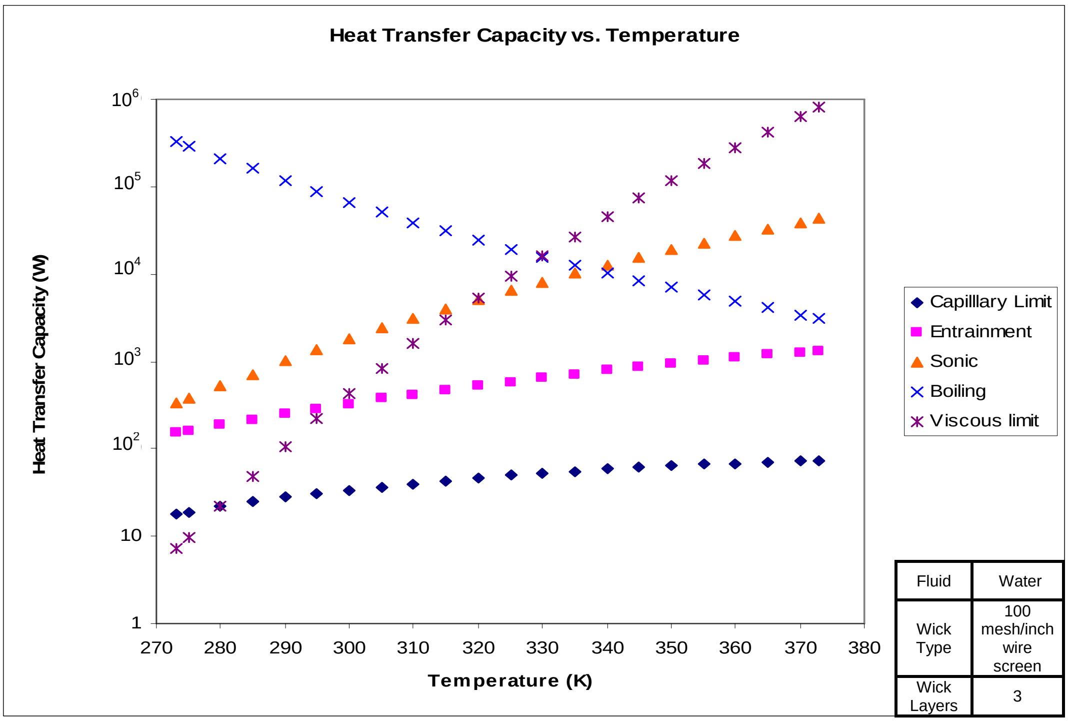


Figure 3.1– Results of Calculation Done by Using 100 Mesh/inch Wire Screen Wick Structure with 3 Layers, Water as working fluid

The heat pipe used is a Water-Copper heat pipe with Copper wire screen meshes as wick structure. The heat pipe has three equal length parts as evaporator, adiabatic and condenser. Each part has a length of 0.25 m. The pipe has a diameter of 0.020 m. with wall thickness 0.0025 m. The wick structure consists of three layers of 100 mesh/inch screen meshes. This provides wire diameter of 1.143×10^{-4} m. The orientation of the pipe is horizontal.

With this configuration, it is seen that the dominant limit that determines the heat transport capacity of the heat pipe is capillary limit. The heat transfer value for the capillary limit is nearly always lower than the other limits' heat transfer values. The exception is the viscous limit. For lower temperatures, especially temperatures up to 280 K (Psat =1 kPa), heat transfer values of the viscous limit are lower than the capillary limit. But when the temperature passes 280 K, viscous limit heat transfer values increase rapidly and capillary limit again becomes the determiner for the heat transfer capacity of the heat pipe.

The results show the general behavior of the heat pipe limits. Viscous limit is important for lower temperatures. At lower temperatures, the vapor pressure becomes lower and lower. This causes the vapor pressure difference between the evaporator and condenser regions of the heat pipe be smaller. As a result, viscous forces in the vapor region become larger than the vapor pressure difference and pressure difference can not be sufficient to generate vapor flow through heat pipe. Therefore, it is expected for viscous limit to be effective at lower temperatures. For entrainment limit, as the temperature increases, the liquid viscosity decreases. As a result, viscous forces between liquid and vapor at liquid-vapor interface decrease, causing an increase in the entrainment limit of the heat pipe. If the formula of entrainment limit is investigated, although the latent heat of vaporization and the surface tension of the Water decrease by increasing temperature, the increase of vapor density results in increase of heat transport capacity for entrainment limitation. The sonic limit depends on the temperature, the density of vapor and the vapor pressure. Therefore, as the temperature increases, the density of vapor and the vapor pressure increases. As a result, the flow goes further away from chocked flow conditions. For the boiling limit concept, it is seen from the graph that as the temperature increases, the heat transport capacity of the boiling limit decreases. This is expected because according to the collapse of a bubble theory, the vapor pressure and surface tension of the working fluid are the factors that form the pressure difference at liquid-vapor interface, and this pressure difference and the temperature are main factors for bubble formation. As temperature increases, the vapor pressure increases and the surface tension decreases, resulting in a decrease in pressure difference. Therefore, decrease in the boiling limit occurs as the temperature increases.

The results of analyses for ammonia are shown in Figure 3.2.

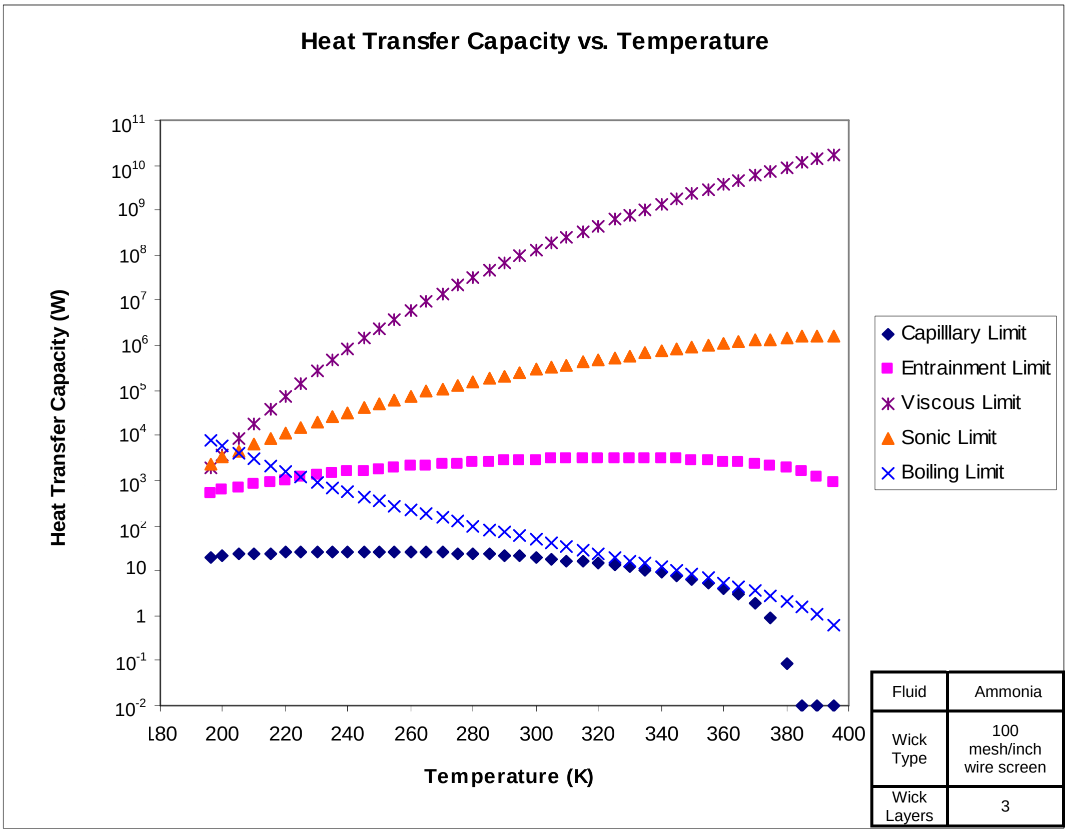


Figure 3.2– Results of Calculation Done by Using 100 Mesh/inch Wire Screen Wick Structure with 3 Layers, Ammonia as working fluid

The behaviors of the limits for Ammonia are similar to the Water heat pipe. By increasing the temperature, all limits except boiling limit shows an increasing trend. The capillary limit continues its increase up to 260 K, and it shows a decreasing trend after that temperature. Similarly, the entrainment limit increase up to 330 K and it starts to decrease for higher temperatures. Different than Water, the governing limit is the capillary limit for Ammonia, even if for lower temperatures. This is due to the high vapor pressures of Ammonia. Although the values of boiling limit come very close to capillary limit values at higher temperatures, it is the capillary limit governing the capacity of the pipe.

The final analyses are performed for Mercury. There is an important difference for Mercury. The Mercury heat pipe can not operate at a horizontal position, because the capillary pumping pressure can not provide liquid flow since the liquid density of Mercury is very high. Therefore, the Mercury heat pipe is analyzed for two degrees of favorable tilt, by keeping other properties of the specified heat pipe constant. The results of Mercury analyses are shown in Figure 3.3.

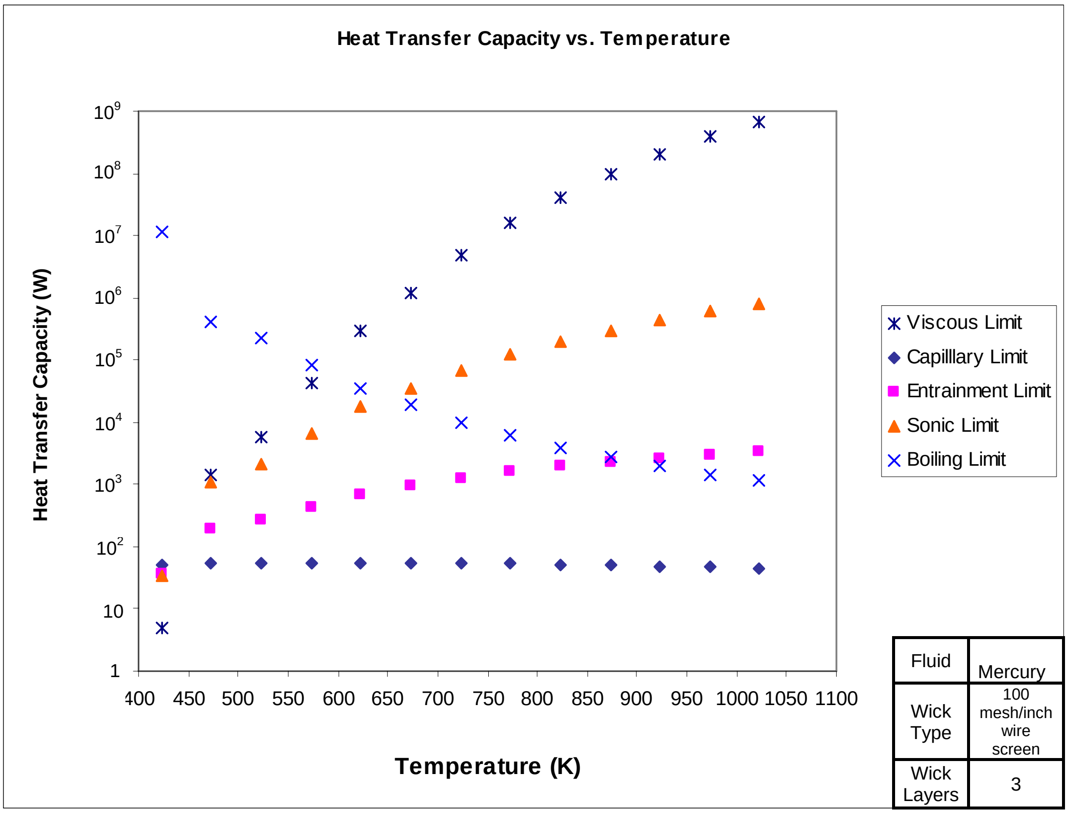


Figure 3.3– Results of Calculation Done by Using 100 Mesh/inch Wire Screen Wick Structure with 3 Layers, Mercury as working fluid

As seen from Figure 3.3, the behavior of the Mercury is similar to other fluids. Viscous, sonic and entrainment limits increase as temperature increases. The temperature increase causes a decrease in boiling limit. However, like in Ammonia, the capillary limit increases for a while, as the temperature increases, up to 673 K. After that temperature, the capillary limit starts to decrease. The governing limit is the viscous limit for lower temperatures, and the capillary limit becomes dominant for average operating temperatures.

The performances of the three heat pipes with different working fluids are compared and tabulated in Table 3.3. If the performances of the three different working fluids are investigated, it is seen that Water shows the best performance for the temperature range of 280 K-373 K. For lower temperatures, Ammonia shows better performance. The Mercury heat pipe can not operate under horizontal orientation. But in favorable tilt conditions of 2 degrees, it operates and becomes effective for high temperatures. For the operating temperature of 335 K, the Water heat pipe can transmit 55 W of heat, whereas Ammonia heat pipe can only provide 10.4 W of heat transfer. The Mercury heat pipe can not be used for 335 K since the temperature is lower than the melting temperature of Mercury.

Table 3.3 – Comparison of the Performances of Three Working Fluids

Wick Properties:	3 layers of 100 mesh/inch wire screen mesh with wire diameter of 1.143×10^{-4} m.			
Working Fluid	Water	Ammonia	Mercury	Governing Limit
Maximum Capacity (W)	74	26.8	55.4	Capillary Limit
Temperature & Psat	373 K & 101.3 kPa	245 K & 130.6 kPa	673 K & 24200 Pa	
Minimum Capacity (W)	7	20	4.8	Viscous Limit, Capillary Limit for Ammonia
Temperature & Psat	273 K & & 611 Pa	196 K & 6.3 kPa	423 K & 1000 Pa	
Capacity for Temperature 335 K (W)	55 W & Psat=51 kPa	10.4 & Psat= 27 Mpa	N/A	N/A

3.1.1.1 Analysis of Mesh Number Effect for Wire Screen Meshes

One of the important parameters that have a dominant effect on heat pipe performance is the number of meshes used in wick structure. The wick structure provides capillary pressure difference which is the driving force for the liquid-vapor flow between evaporator and condenser sections. Therefore, the effect of the mesh number on the capacity of the heat pipe is investigated first.

The structure of a wire screen mesh is shown in Figure 3.4 [21].

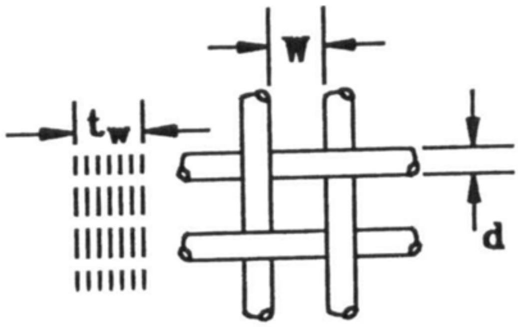


Figure 3.4- Structure of a Wire Screen Mesh (Adapted from [21])

Where;

-w: Distance between two wires (m)

-d: Wire diameter (m)

-$t_{w:}$ Wick thickness (m)

As the number of meshes decreases, the distance between wires (w) and wire diameter (d) increase. As a result of increase in distance, the frictional losses in liquid phase and the capillary pumping pressure decreases.

A 100 meshes/inch screen consisting of three layers is used in the first analysis. To see the effect of mesh number, keeping all other parameters constant, mesh numbers of 80 meshes/inch, 60 meshes/inch, 50 meshes/inch, 40 meshes/inch, 30 meshes/inch, 24 meshes/inch, 18 meshes/inch, 16 meshes/inch, 12 meshes/inch and 8 meshes/inch are used in calculations.

The results of the analysis performed with 80 meshes/inch wire screen mesh can be seen in the Figures 3.5 – 3.7. The analyses are performed for all three working fluids, keeping the specified heat pipe properties constant, except the mesh number. But a distinction is performed for Mercury. The orientation of the Mercury heat pipe is set as 2 degrees (in favorable tilt position), since it can not operate under horizontal pipe conditions.

First results are shown for water in Figure 3.5.

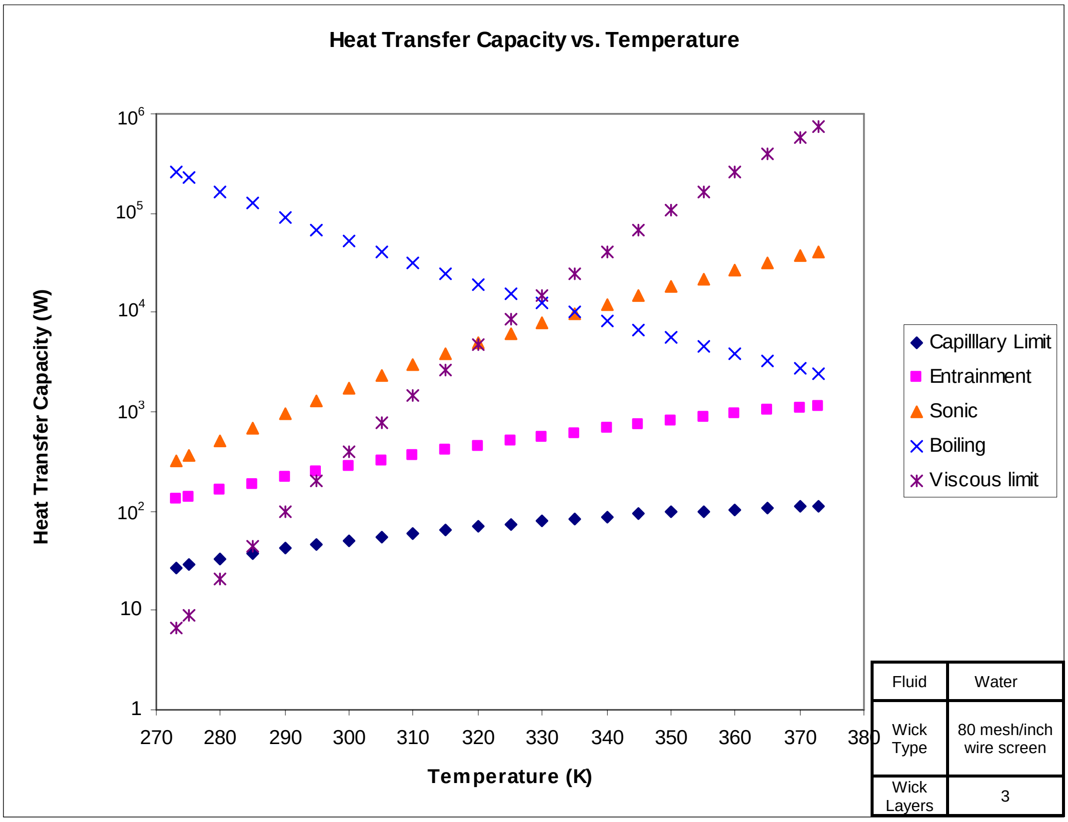


Figure 3.5 – Results of Calculation Done by Using 80 Meshes/inch Wire Screen Wick Structure with 3 Layers, Water as Working Fluid

The heat pipe configuration used is the same with previous analyses. It is a 0.75 m length copper pipe with a diameter of 0.020 m. and wall thickness of 0.0025 m. The parts have equal lengths, and the orientation is horizontal (except for Mercury). Only the mesh number and the wire diameter of the wick structure are changed. Instead of 100 meshes/inch, 80 meshes/inch is used and the wire diameter becomes 1.4×10^{-4} m.

If the Figure 3.5 is investigated, it is seen that heat transfer values for all limits except capillary limit are decreased for Water, when compared with the Water heat pipe with 100 meshes/inch Copper screen structure. Only capillary limit value is increased. But the governing limit is same. At lower temperatures viscous limit is the dominant limit. After 285 K (P_{sat} =1.4 kPa), capillary limit becomes determiner for the heat transfer capacity of the heat pipe.

The analyses performed for ammonia are shown in Figure 3.6.

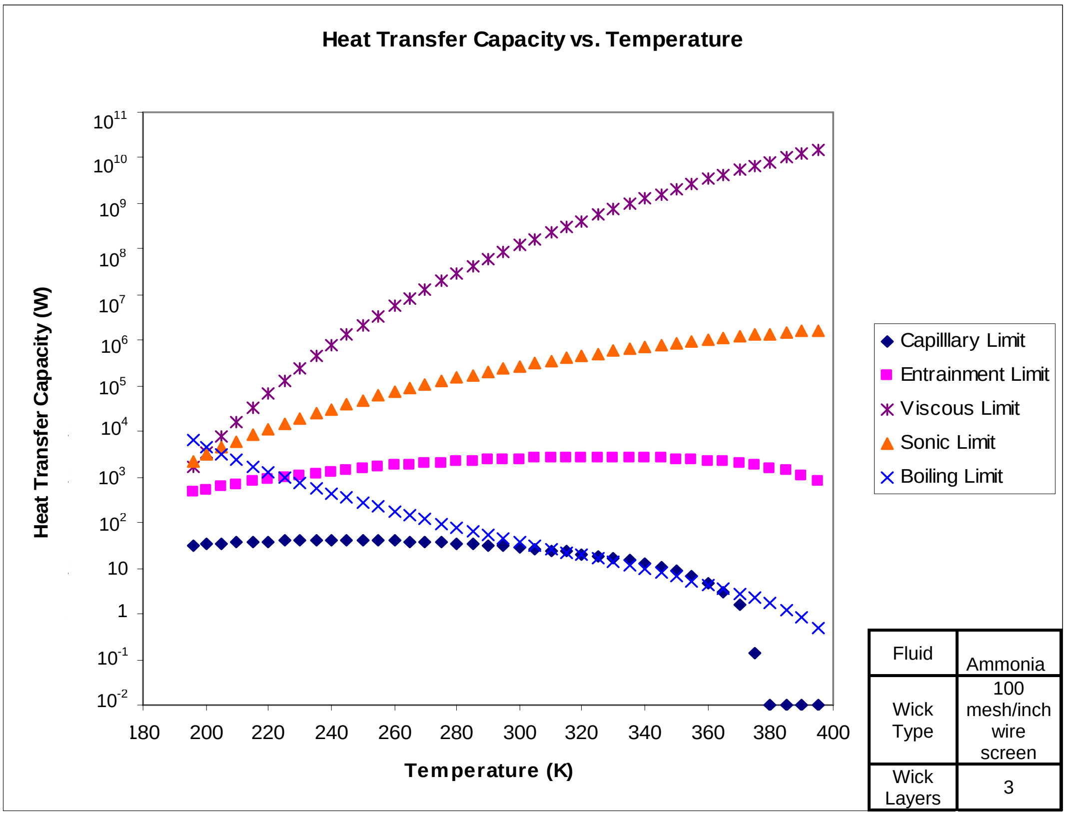


Figure 3.6 – Results of Calculation Done by Using 80 Mesh/inch Wire Screen Wick Structure with 3 Layers, Ammonia as working fluid

If the Figure 3.6 is investigated, it is seen that heat transfer values for all limits except capillary limit are decreased for Ammonia, when compared with the Ammonia heat pipe with 100 mesh copper screen structure. However, capillary limit shows an increasing trend. The governing limit differs for Ammonia. It is the capillary limit for the whole temperature scale except 315 K -360 K interval. Between these temperatures, it is the boiling limit that limits the heat pipe capacity.

Finally, the analyses are performed for Mercury, with a tilt angle of 2 degrees. The results are shown in Figure 3.7.

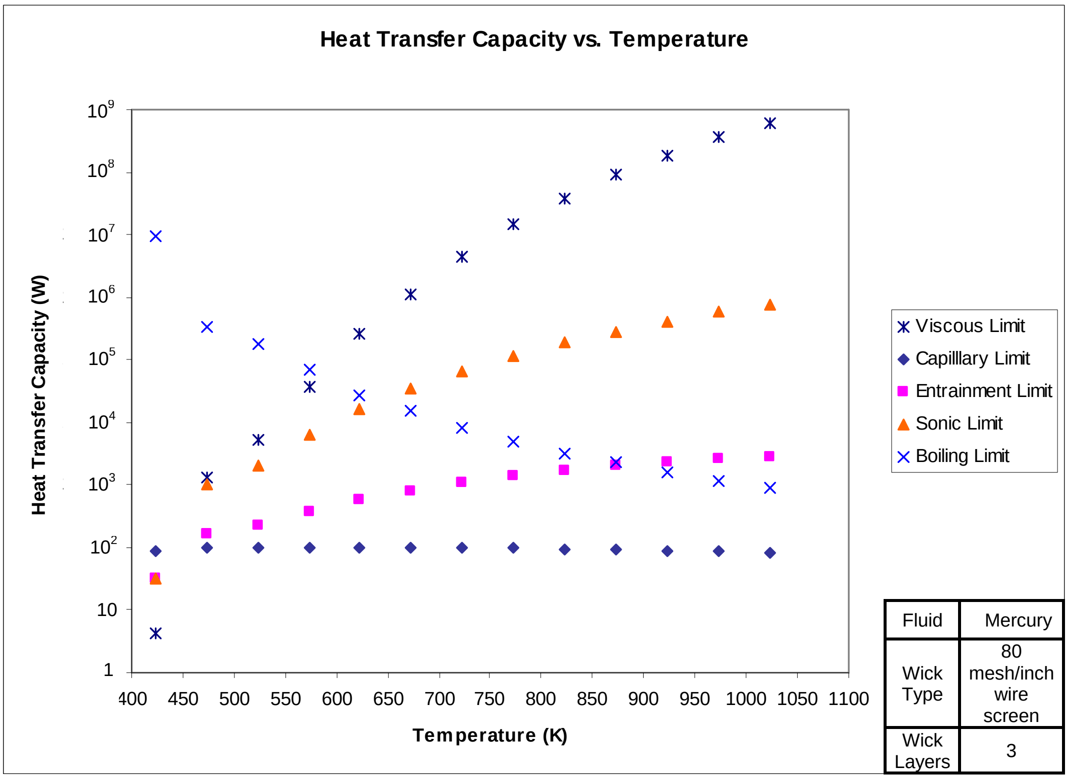


Figure 3.7 – Results of Calculation Done by Using 80 Mesh/inch Wire Screen Wick Structure with 3 Layers, Mercury as working fluid

If the Figure 3.7 is investigated, it is seen that the trend of the limits are similar to the ones in Ammonia and Water. The heat transfer values for all limits except capillary limit are decreased, whereas the values of capillary limit increases. It is the viscous limit which limits the capacity of the pipe for lower temperatures. But for average operating temperatures, the determiner is the capillary limit again.

All these analyses are performed for wick structures having three layers of 60, 50, 40, 30, 24, 18, 12 and 8 meshes/inch wire screens. It is seen that the decreasing trend for viscous, entrainment, sonic and boiling limits continue as the mesh number decreases. However, the capillary limit shows an increasing trend up to a specific mesh number, and then it starts to decrease. These are all valid for three of the working fluids. But the mesh number where capillary limit changes its trend differs from fluid to fluid. Moreover, the governing limit changes for fluids. All of these analyses are presented in Tables 3.4 – 3.6. The maximum and minimum heat transfer capacities, corresponding temperatures, pressures, mesh numbers and the governing limits are shown in the table.

Table 3.4 – Effect of the Mesh number on the Performance of Water Heat Pipe

Mesh Number (meshes/inch)	Water						
	Governing Limit		Maximum Capacity		Minimum Capacity		Heat Transfer for T=335 K
			T (K) & Psat	Capacity (W)	T (K) & Psat	Capacity (W)	
100	Viscous up to 280 K	Capillary	373 & 101.3 kPa	74	273 & 611 Pa	7	55 W & Psat=21.6 kPa
80	Viscous up to 280 K	Capillary	373 & 101.3 kPa	112	273 & 611 Pa	6.5	84 W & Psat=21.6 kPa
60	Viscous up to 285 K	Capillary	373 & 101.3 kPa	181	273 & 611 Pa	5.4	136 W & Psat=21.6 kPa
50	Viscous up to 290 K	Capillary	373 & 101.3 kPa	244	273 & 611 Pa	4.6	184 W & Psat=21.6 kPa
40	Viscous up to 295 K	Capillary	373 & 101.3 kPa	380	273 & 611 Pa	4.3	288 W & Psat=21.6 kPa
30	Viscous up to 300 K	Entrainment	373 & 101.3 kPa	573	273 & 611 Pa	3	433 W & Psat=21.6 kPa
24	Viscous up to 305 K	Entrainment	373 & 101.3 kPa	784	273 & 611 Pa	2.7	581 W & Psat=21.6 kPa
18	Viscous up to 305 K	Entrainment	373 & 101.3 kPa	931	273 & 611 Pa	2	618 W & Psat=21.6 kPa
16	Viscous up to 305 K	Entrainment	373 & 101.3 kPa	894	273 & 611 Pa	1.6	552 W & Psat=21.6 kPa
12	Viscous up to 305 K	Entrainment	373 & 101.3 kPa	499	273 & 611 Pa	0.8	261 W & Psat=21.6 kPa
8	Viscous up to 305 K	Capillary	373 & 101.3 kPa	138	273 & 611 Pa	0.3	88.5 W & Psat=21.6 kPa

Table 3.5 – Effect of the Mesh number on the Performance of Ammonia Heat Pipe

Mesh Number (meshes/inch)	Ammonia						
	Governing Limit		Maximum Capacity		Minimum Capacity		Heat Transfer for T=335 K
			T (K) & Psat	Capacity (W)	T (K) & Psat	Capacity (W)	
100	Capillary		245 K & 130 kPa	26.8	381 & 7.2 Mpa	0	10.4W & Psat=2.7 Mpa
80	Capillary	Boiling For 315 K-360 K	245 K & 130 kPa	40.6	376 & 6.6 MPa	0	11.2 & Psat=2.7 Mpa
60	Capillary	Boiling For 282 K-362 K	245 K & 130 kPa	66	369 & 5.7 kPa	0	8.25 & Psat=2.7 Mpa
50	Capillary	Boiling For 265 K-360 K	245 K & 130 kPa	89	363 & 5.1 MPa	0	6.7 & Psat=2.7 Mpa
40	Capillary	Boiling For 250 K-350 K	245K & 130 kPa	140	354 & 4.2 MPa	0	5.5 & Psat=2.7 Mpa
30	Capillary	Boiling For 234K-342 K	230K & 60.4 kPa	215	344& 3.3 MPa	0	3.9 & Psat=2.7 Mpa
24	Capillary	Boiling For 225 K-330 K	238K & 92.4 kPa	304	332& 2.5 MPa	0	0 & Psat=2.7 Mpa
18	Capillary	Boiling For 215K-315 K	240K & 102.2 kPa	395	316& 1.6 MPa	0	0 & Psat=2.7 Mpa
16	Capillary	Boiling For 235K-343 K	240K & 102.2 kPa	408	309& 1.4 MPa	0	0 & Psat=2.7 Mpa
12	Capillary	Boiling For 215K-296K	250K & 165 kPa	337	297& 968 kPa	0	0 & Psat=2.7 Mpa
8	Capillary	Boiling For 225K-275K	245K & 130 kPa	164.5	276& 477 kPa	0	0 & Psat=2.7 Mpa

Table 3.6 – Effect of the Mesh number on the Performance of Mercury Heat Pipe

Mesh Number (meshes/inch)	Mercury						
	Governing Limit		Maximum Capacity		Minimum Capacity		Heat Transfer for T=335 K
			T (K) & Psat	Capacity (W)	T (K) & Psat	Capacity (W)	
100	Capillary		673 K & 242 kPa	55.4	423 K & 1 kPa	4.8	N/A
80	Capillary		673 K & 242 kPa	100.5	423 K & 1 kPa	4.3	N/A
60	Entrainment up to 573 K	Capillary	673 K & 242 kPa	220	423 K & 1 kPa	3.6	N/A
50	Entrainment up to 573 K	Capillary	673 K & 242 kPa	370	423 K & 1 kPa	3.1	N/A
40	Entrainment up to 723 K	Capillary	673 K & 242 kPa	748	423 K & 1 kPa	2.8	N/A
30	Entrainment up to 873 K	Boiling above 873 K	773 K & 886 kPa	1773	423 K & 1 kPa	1.9	N/A
24	Entrainment up to 873 K	Boiling above 873 K	873 K & 2.3 MPa	3382	423 K & 1 kPa	1.8	N/A
18	Entrainment up to 873 K	Boiling above 873 K	1023 K & 6.3 MPa	7193	423 K & 1 kPa	1.8	N/A
16	Entrainment up to 873 K	Boiling above 873 K	1023 K & 6.3 MPa	9013	423 K & 1 kPa	1.8	N/A
12	Entrainment up to 873 K	Boiling above 873 K	1023 K & 6.3 MPa	7326	423 K & 1 kPa	0.5	N/A
8	Entrainment up to 923 K	Boiling above 923 K	1023 K & 6.3 MPa	1745	423 K & 1 kPa	0.25	N/A

In the Table 3.5, minimum capcity of Ammonia is shown as 0 W. It is because of the reason that after a specific temperature, the capacity of Ammonia heat pipe in terms of capillary limit falls to 0. This specific temperature depens on the mesh number. The heat pipe can not transfer any amount of heat. Therefore, the minimum capacity is shown as 0 W in the table.

To sum up, the effect of mesh number is examined for three different types of heat pipe; Water – Copper, Ammonia – Aluminum and Mercury - Steel heat pipes having 0.75 m total length (evaporator, condenser and adiabatic sections have the equal lengths of 0.25 m.), with the pipe diameter of 0.020 m, having wall thickness of 0.0025 m., located in horizontal position (exception; 2 degrees of favorable tilt is used for Mercury). The wick structure is composed of 3 layers of wire screen meshes. As the mesh number decreases, heat transfer capacity of all limits shows similar behaviors for different working fluids.

As the mesh number decreases, heat transfer capacity of all limits except capillary limit decreases as well. When the mesh number of the wire screen is decreased, the wire diameter of wick structure increases. This wire diameter increase causes decrease in vapor space area. Therefore, as the vapor space area decreases, the capacities of all four limits – viscous, entrainment, sonic and boiling - decrease. But capillary limit has different behavior than other limits. Capillary limit values increase and reaches their maximum at a specific mesh number. If mesh number decreases more, capillary limit values begin to decrease. As mentioned before, capillary pumping pressure and frictional losses are two important factors for capillary limit. By decreasing mesh number, capillary pumping pressure decreases because of increasing effective pore radius. However, frictional losses decrease, too. As mesh number decreases, the spacing between the wires increases, and this causes decrease in frictional losses. Up to that specific mesh number, decrease in frictional losses is much more than decrease in pumping pressure, causing an increase in capillary limit. For lower mesh numbers, decrease in pumping pressure become higher than decrease in frictional losses, and as a result capillary limit values start to decrease.

This critical point is 18 mesh/inch for Water, 16 meshes/inch for Ammonia and 16 meshes/inch for Mercury, when the specified heat pipe configuration is used. This may be change if the configuration is changed.

The trends of the limits of Water – Copper heat pipe for a selected temperature of 335 K (Psat =21.8 kPa) are given in the Figure 3.8. The for this graph. The behaviors of the limits are in accordance with mentioned behaviors.

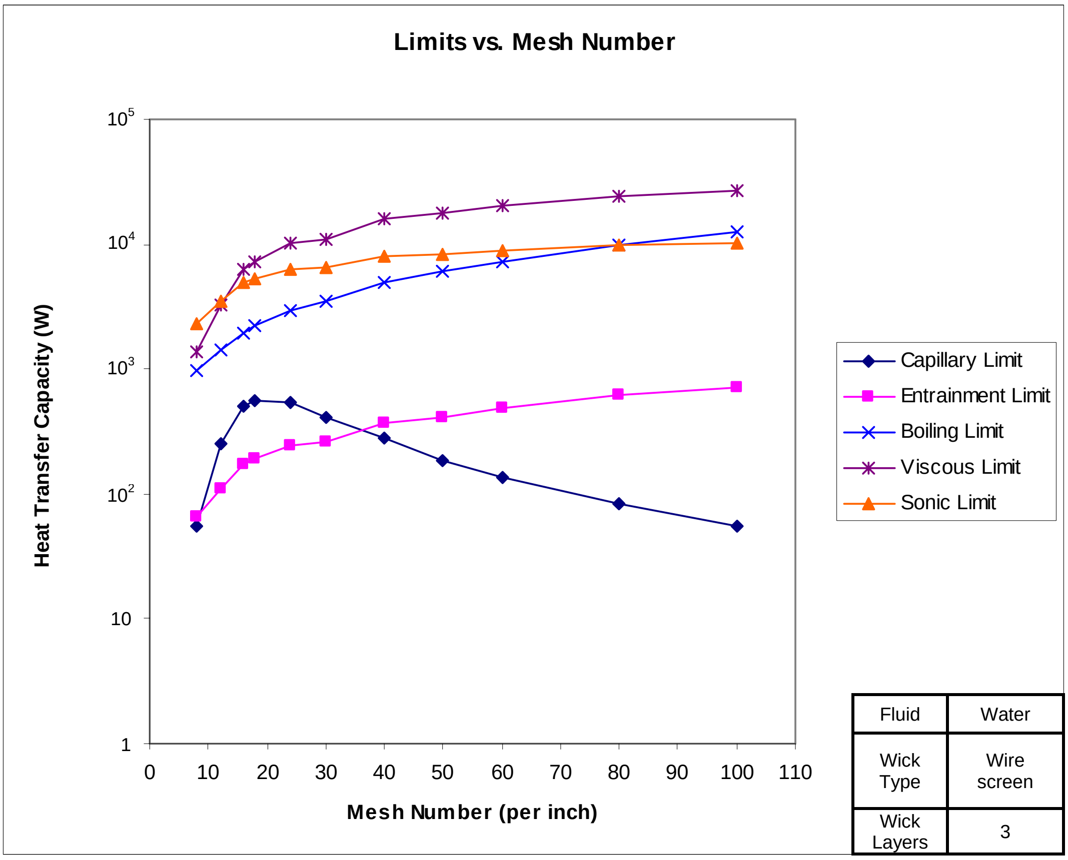


Figure 3.8 – Changes in Limits with respect to Mesh Numbers at 335 K (Psat =21.8 kPa), Water as Working Fluid

If entrainment and capillary limits are examined for the temperature 335 K., the graph in Figure 3.9 can be obtained, showing the transition between entrainment and capillary limits according to mesh number.

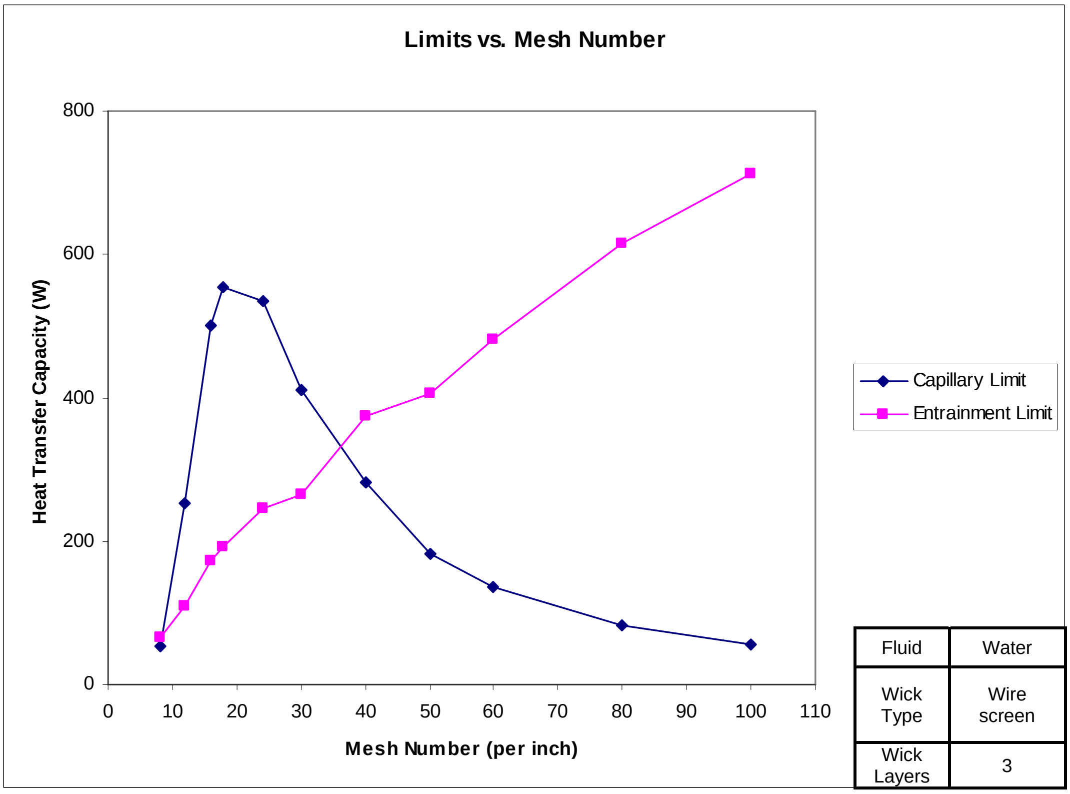


Figure 3.9– Transition between Capillary and Entrainment Limits with respect to Mesh Numbers at 335 K (Psat =21.8 kPa), Water as Working Fluid

If Figure 3.9 is investigated, it can be seen that entrainment limit becomes dominant after 40 meshes/inch. In order to get maximum heat capacity, 40 meshes/inch wick structure shall be used for Water, giving 288 W capacity at 335 K operating temperature.

For Ammonia, the situation is different than Water- Copper heat pipe. It is the boiling and capillary limits that govern the performance of the Ammonia heat pipe. The behavior of the limits with respect to mesh number can be seen in Figure 3.10 for an operating temperature of 245 K (Psat=130 kPa).

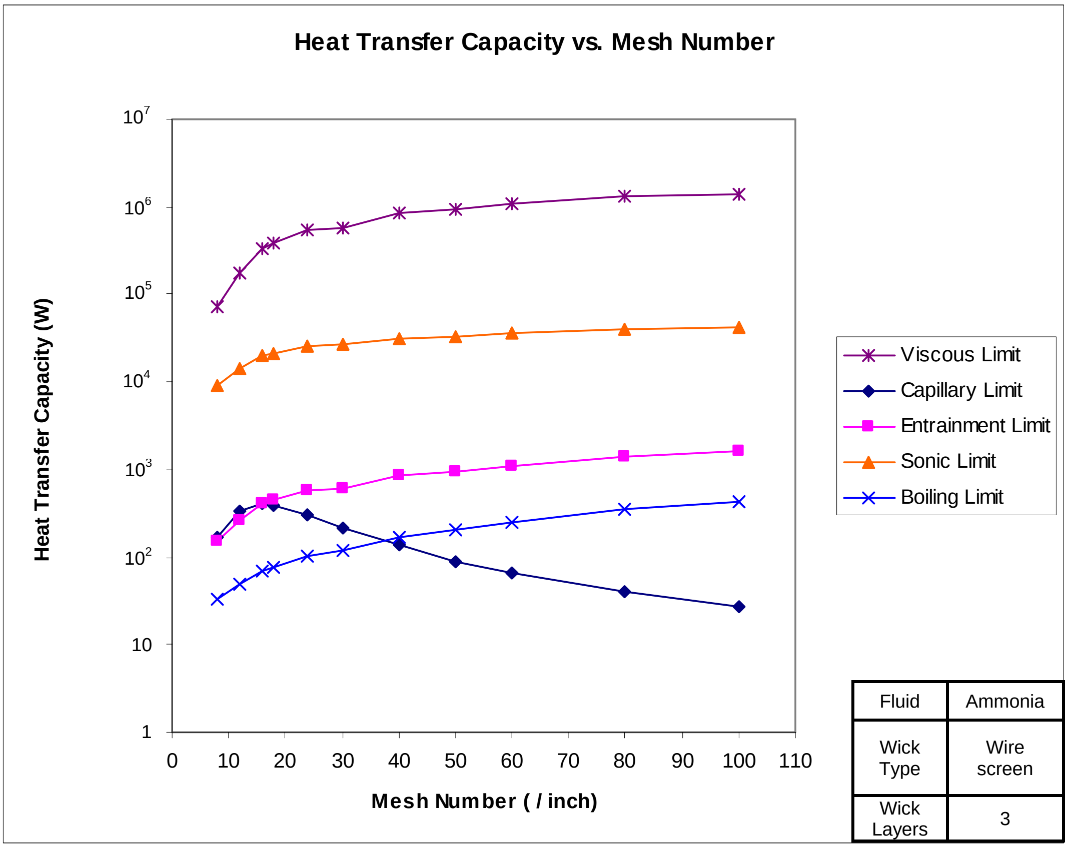


Figure 3.10 – Changes in Limits with respect to Mesh Numbers at 245 K (Psat =130 kPa), Ammonia as Working Fluid

The transition points between two limits can be seen in Figure 3.11 for 245 K operating temperature.

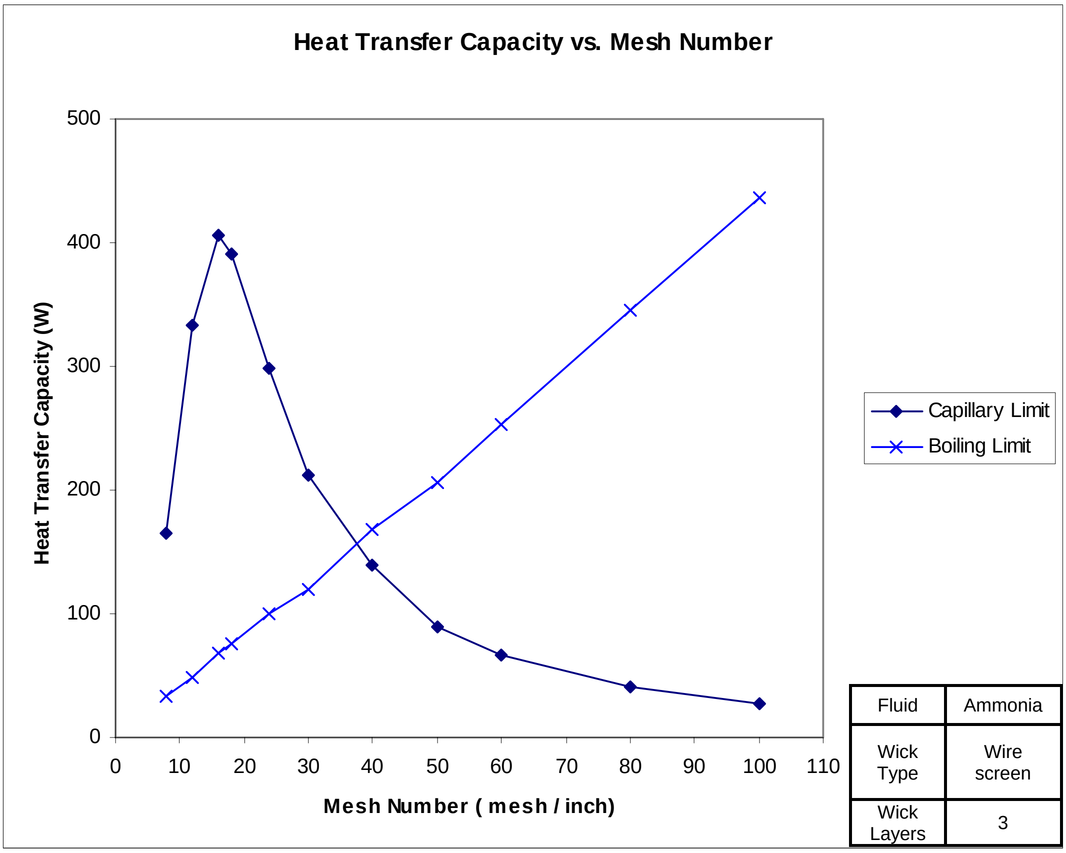


Figure 3.11– Transition between Capillary and Boiling Limits with respect to Mesh Numbers at 245 K (Psat =130 kPa), Ammonia as Working Fluid

The capillary limit increases up to 16 meshes/inch. The boiling limit becomes important after 40 meshes/inch. Therefore, to get maximum heat transport capacity from Ammonia –Copper heat pipe at 245 K operating temperature, a 40 meshes/inch wick structure shall be used, giving 138 W capacity.

Finally, the results of mercury analyses are shown in Figure 3.12 for an operating temperature of 673 K. The behavior of the limits are similar to the ones in Ammonia and Water analyses.

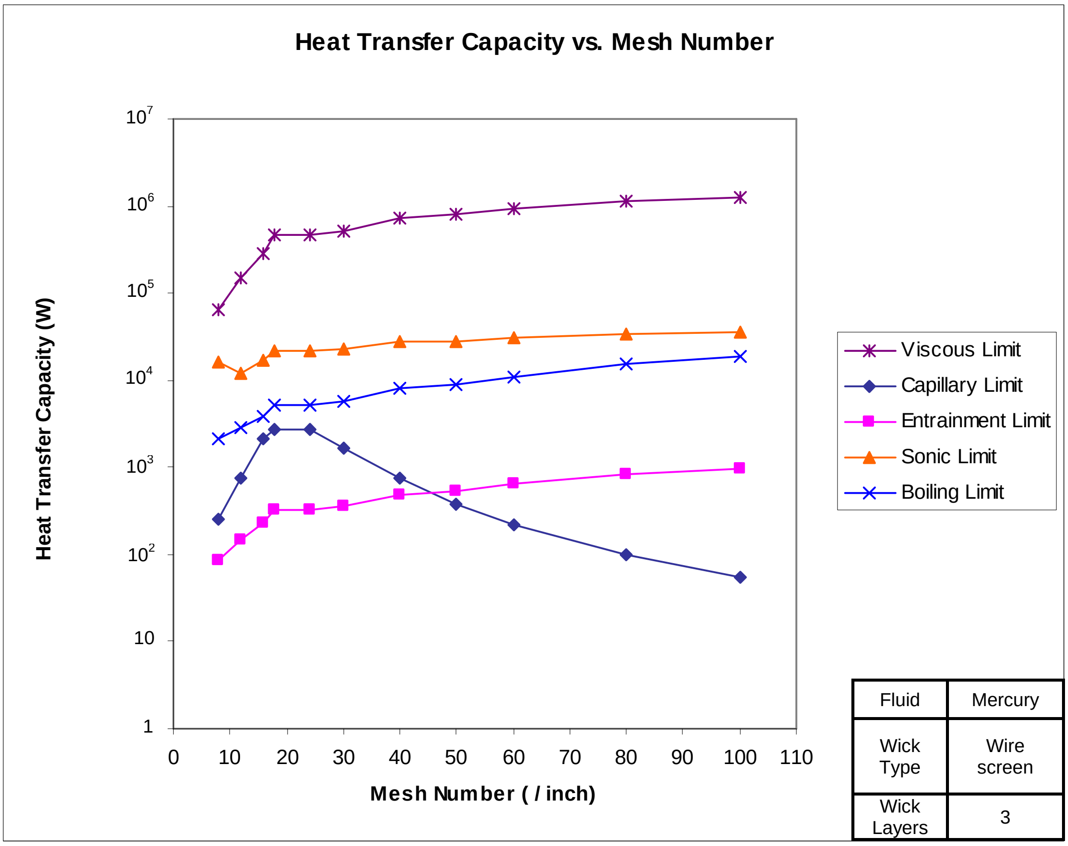


Figure 3.12 – Changes in Limits with respect to Mesh Numbers at 673 K (Psat =242 kPa), Mercury as Working Fluid

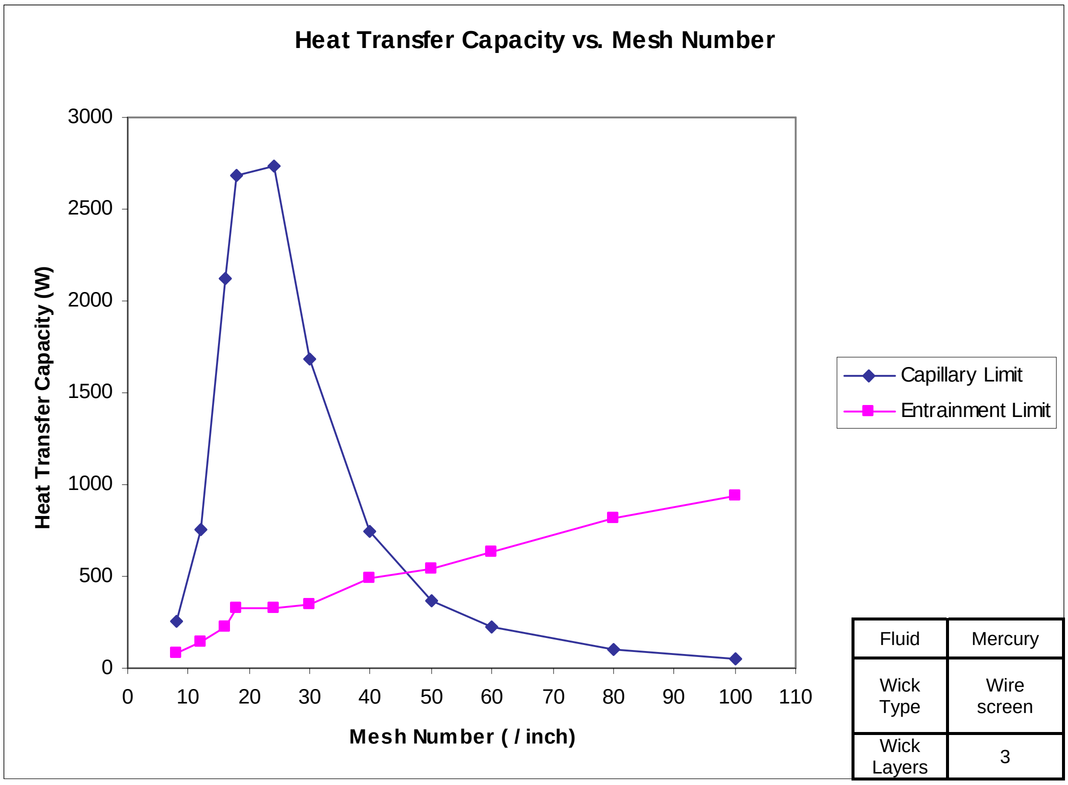


Figure 3.13 – Transition between Capillary and Entrainment Limits with respect to Mesh Numbers at 673 K (Psat =242 kPa), for Mercury

For mercury, entrainment and capillary limits are the limits that set the capacity of the heat pipe. In Figure 3.13, the transition between these two limits are shown for 673 K operating temperature. The capillary limit increases up 24 meshes/inch. Up to 50 meshes, the capillary limit is important. But if the mesh number is lower, it is the entrainment limit that determines the capacity of the heat pipe. In order to get maximum heat, 40 meshes/inch wick structure shall be used for Mercury. The entrainment limit gives 494 W capacity for 40 meshes/inch as maximum.

Finally, in the analyses it is seen that Water shows better performance from Ammonia for temperatures higher than 280 K. Therefore, it is more suitable than Ammonia for the temperature range of 280 K – 373 K. The Mercury heat pipe shows better performances for higher temperatures.

3.1.1.2 Analysis of Layer Number Effect

Layer number is another parameter that affects the performance of a heat pipe. In first configuration, a heat pipe that has wick structure consisting of three layers is used. But what will be the effect of changing layer numbers, i.e. from three to two, on performance of heat pipe? In order to see the layer number effect, following analyses are done and results are presented.

Increasing or decreasing the layer number may have different effects on heat transfer capacity of a heat pipe. Analyses are done to see what happens if the layer number of meshes of wick structure is decreased or increased. For this purpose, firstly the layer number of the main configuration is decreased from three to two, keeping other parameters constant. Secondly, to see the effect of layer number increase on the heat transfer limits, the layer number of the main configuration is increased from three to five by keeping all other parameters constant. The heat pipe used in the analyses has 0.75 m. total length with evaporator, condenser and adiabatic sections each having 0.25 m. length. The pipe diameter is 0.020 m. and it is oriented horizontally. The wick structure inside the pipe is a copper wire screen mesh with 100 mesh/inch with layer numbers two or five. The configuration used is:

- The baseline configuration with number of layers two and five.

Using this configuration, analyses are performed using the computer code. The comparison of three layer configuration results with two and five layer ones are given in b graphs in logarithmic scales, for all limits.

a) Capillary Limit:

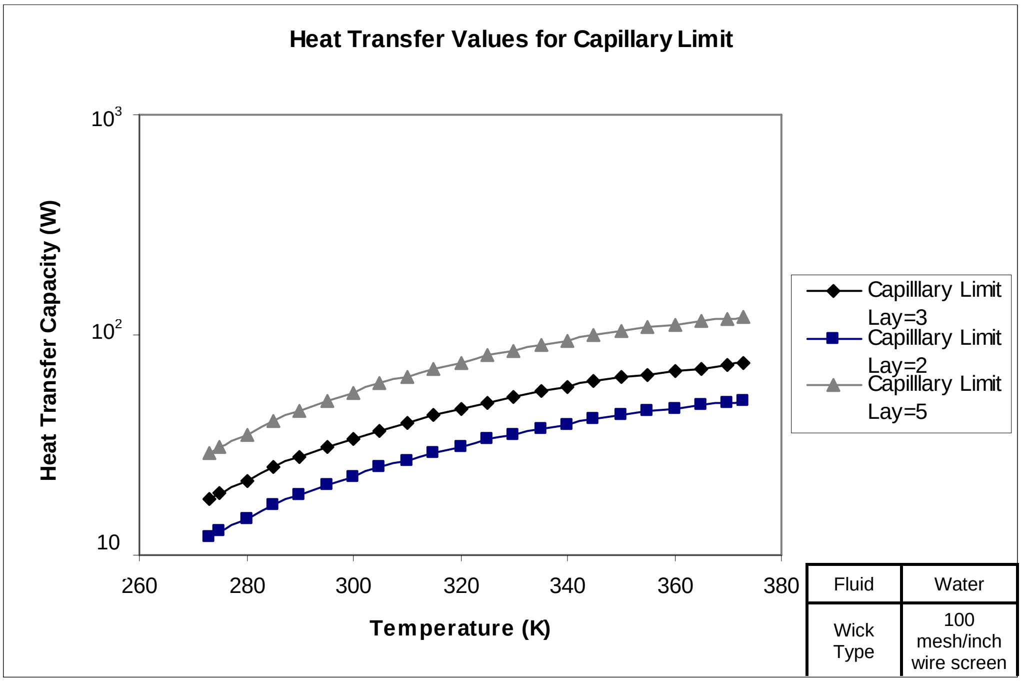


Figure 3.14 – Effect of Layer Number on Capillary Limit for Water

It is obvious from the Figure 3.14 that there is a decrease in heat transfer values of capillary limit, when the layer of wick structure is decreased from three layers to two layers. Decrease in the number of layers means decrease in wick area and increase in vapor space. If the vapor space increases, more vapor can flow from evaporator to condenser. However, this increase in flow increases the liquid flow losses inside the wick structure, causing a decrease in the total heat transfer capacity of the heat pipe, in terms of capillary limit. This causes decrease of capillary limit values. On the other hand, the increase in layer number causes increase in capillary limit, as expected.

The situation is same for Ammonia and Mercury. The results of analyses for Ammonia and Mercury are shown in Figure 3.17 and Figure 3.18 for capillary limit.

b) Entrainment limit:

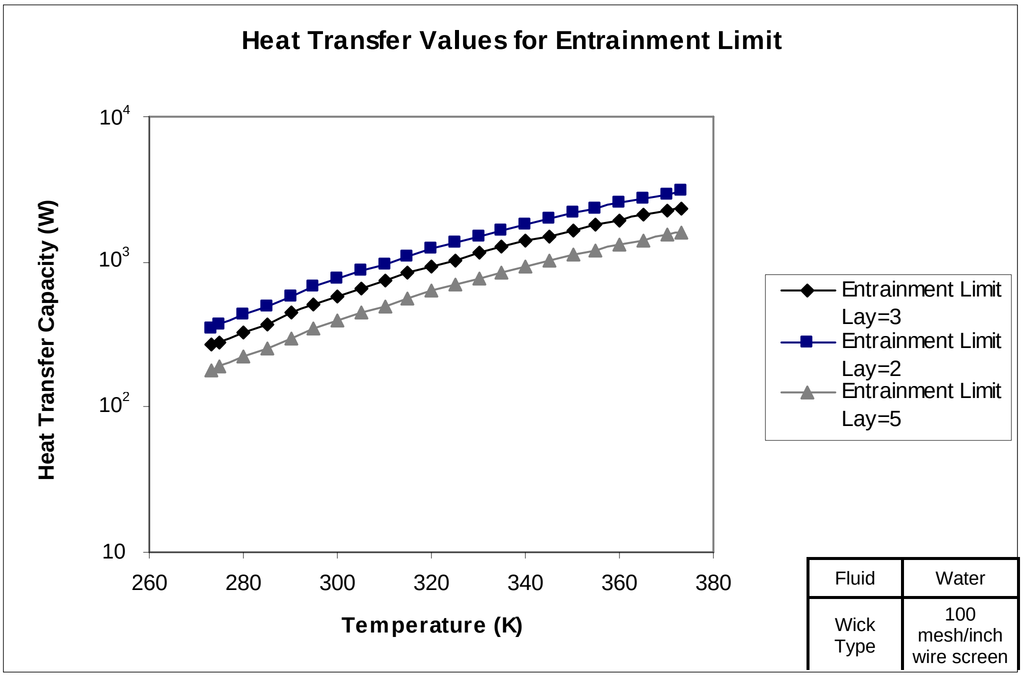


Figure 3.15 – Effect of Layer Number on Entrainment Limit for Water

As can be seen from the Figure 3.15, decreasing the layer number of wick structure causes increase in the heat transfer capacity for entrainment limit. Like in capillary limit, decreasing the layer results in increase of vapor space area. As vapor space area increases, the capacity of entrainment limit increases, too. Effect of shear forces between liquid and vapor becomes less and as a result, more vapor can be transferred from evaporator to liquid when compared with three layer case, for a certain temperature. This can be seen when the formula of the entrainment limit is investigated.

Layer number increase causes an opposing effect when it is compared with the effect of layer number decrease on entrainment limit. Heat transfer capacity of the entrainment limit decreased with increase in layer number. The reason is if the layer number increases, vapor flow area decreases causing an increase in shear forces losses. This decreases the capacity of the heat pipe in terms of entrainment limit.

The results of analyses for Ammonia and Mercury are shown in Figure 3.17 and Figure 3.18 for entrainment limit. As seen from these figures, the entrainment limit of Ammonia and Mercury show similar behavior to the one in Water.

c) Viscous, Sonic and Boiling Limits:

The Figure 3.16 shows the effect of decrease in layer number to heat transfer capacity of heat pipe regarding viscous, sonic and boiling limits.

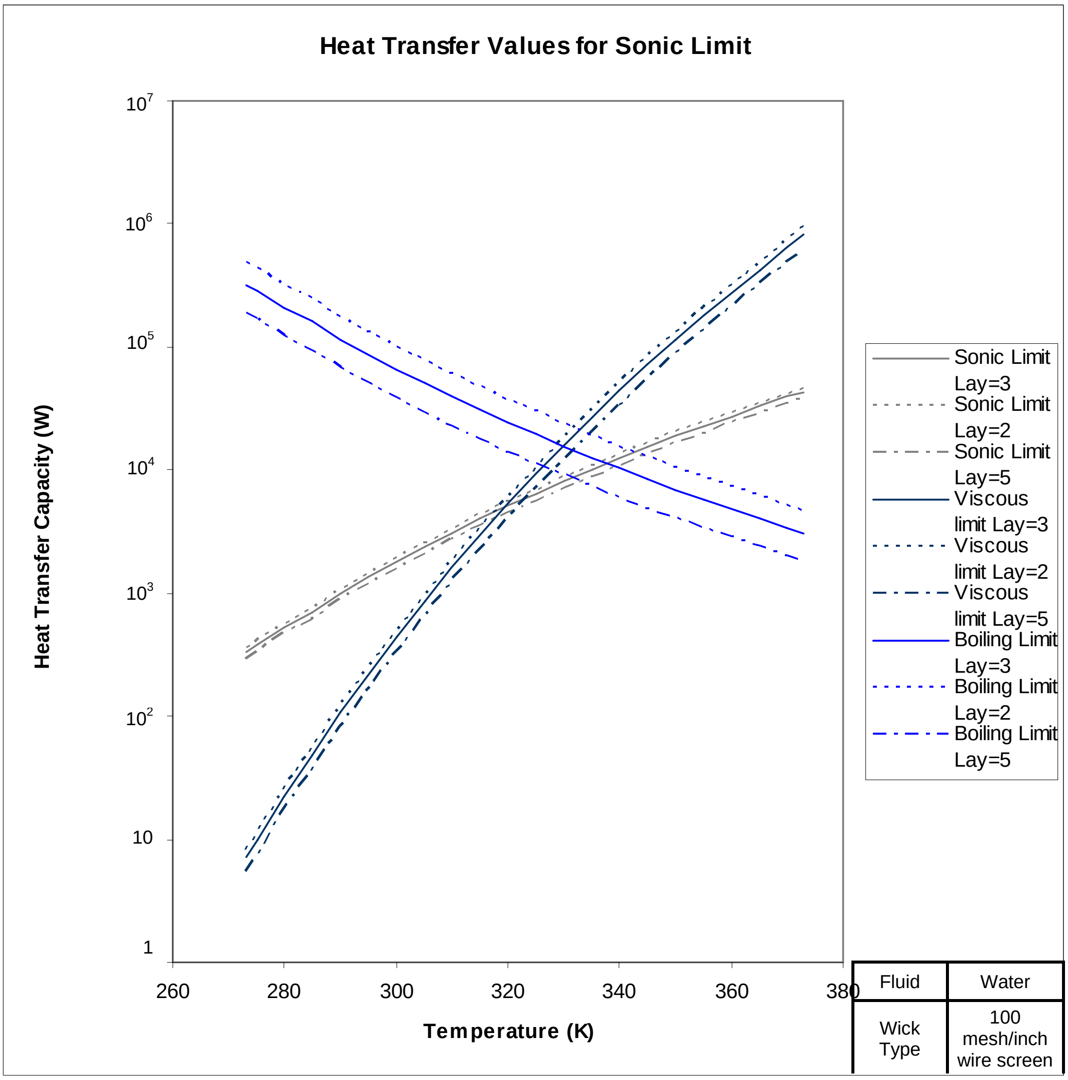


Figure 3.16– Effect of Layer Number on Viscous, Sonic and Boiling Limit for Water

Examining the Figure 3.16, it is seen that the situation is different than capillary limit. Decrease in layer number causes a decrease in capillary limit. However, for viscous, sonic and boiling limits, decrease in layer number provides increase in heat transfer capacity.

This seems logical since decrease in layer number causes increase in vapor space area. Vapor can find a larger area to flow. This causes decrease in viscous losses, as a result more vapor can be transported from evaporator to condenser. Opposing to effect of layer number decrease – increase in layer number results decrease in viscous limit.

For sonic limit, results of two layer and three layer structures seem closer. By looking at the results, it can be said that decreasing the layer number from three to two does not have a certain effect on sonic limit. This can thought as logical since the sonic limit mainly depends on the density of vapor and pressure. Density of vapor and pressure values depends on temperature. Therefore, decreasing the layer number without changing temperature does not affect the heat transfer capacity of sonic limit so much. This can be seen in the formula for sonic limit, too. Similarly, increasing the layer number does not have certain effect on sonic limit.

The layer number change affects the boiling limit since it changes the vapor area and as a result the vapor diameter. If vapor diameter changes, the boiling limit changes. For boiling limit, it is obvious from the graph that layer number decrease affects the heat transfer capacity of boiling limit at lower temperatures. The decrease in layer number causes increase in boiling limit. At higher temperatures, the results of both cases are closer. Similarly, the layer number increase affects the boiling limit at lower temperatures. The heat transfer capacity of boiling limit decreases with increase in layer number. However, at higher temperatures, it seems that layer number increase does not affect the boiling limit so much.

The effects of layer number on the viscous, sonic and boiling capacities of Ammonia and Mercury are shown in Figures 3.17 - 3.20. The decrease or increase in the layer number affects the limits of Ammonia and Mercury in the same way with Water.

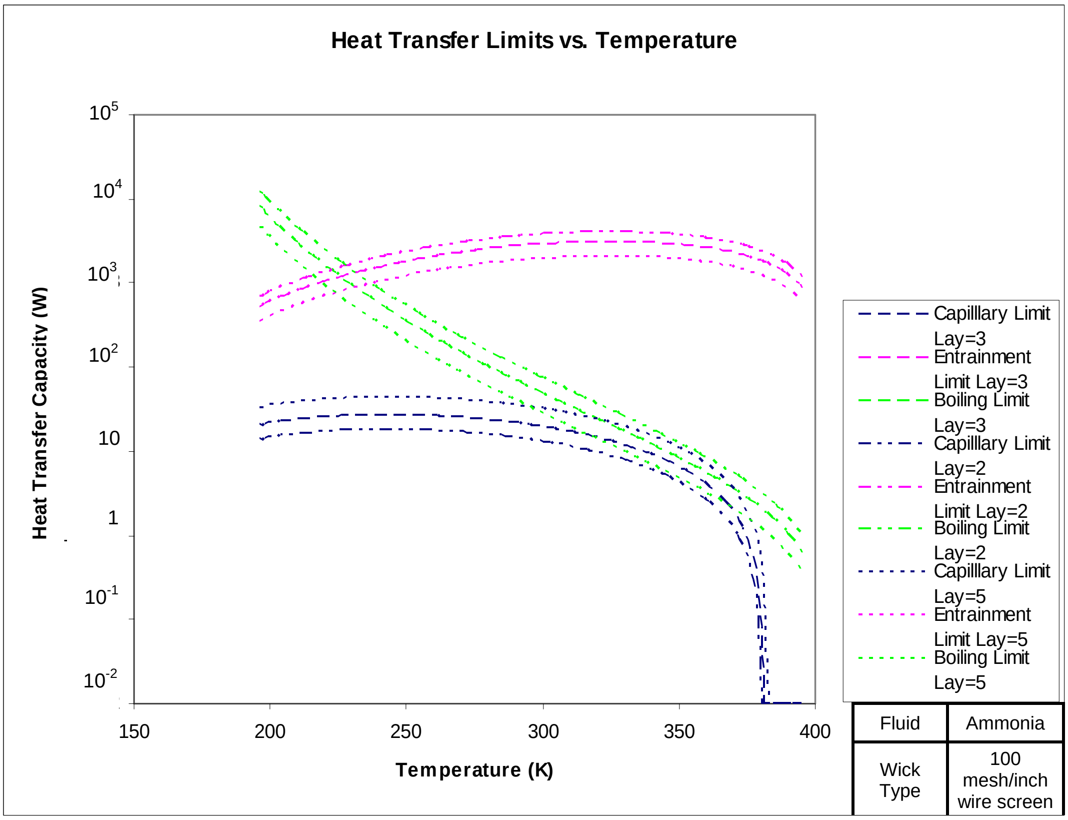


Figure 3.17– Effect of Layer Number on Capillary, Entrainment and Boiling Limits of Ammonia

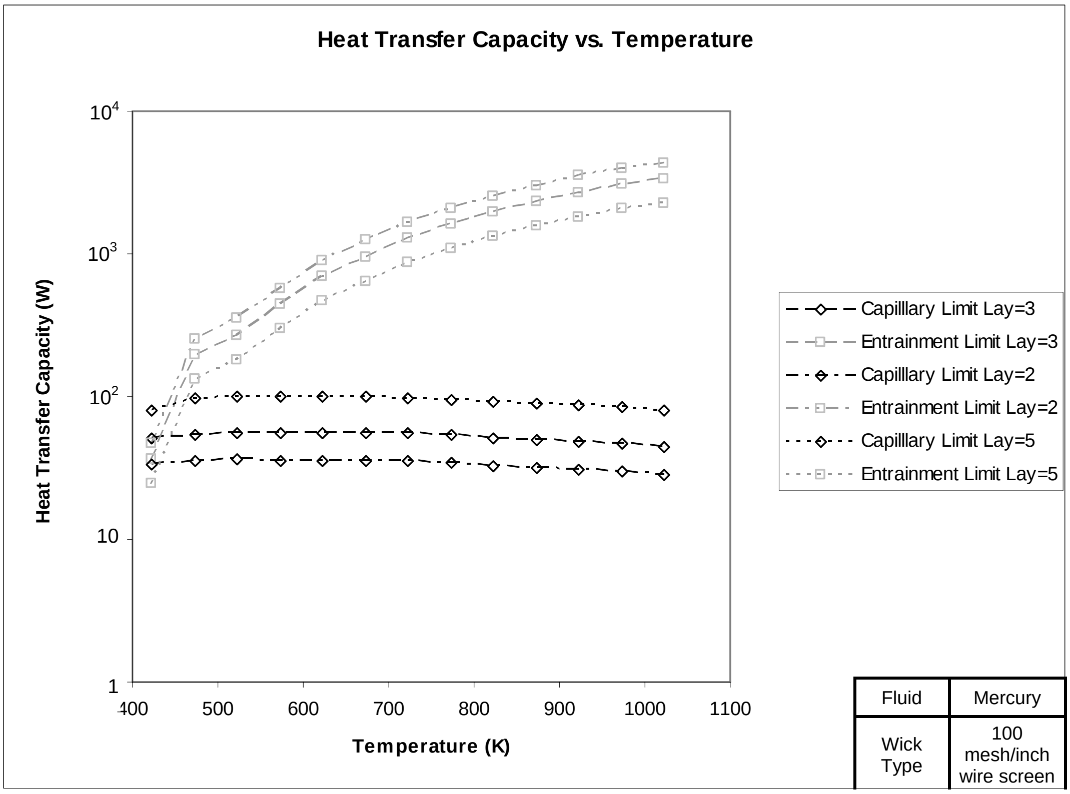


Figure 3.18– Effect of Layer Number on Capillary and Entrainment Limits of Mercury

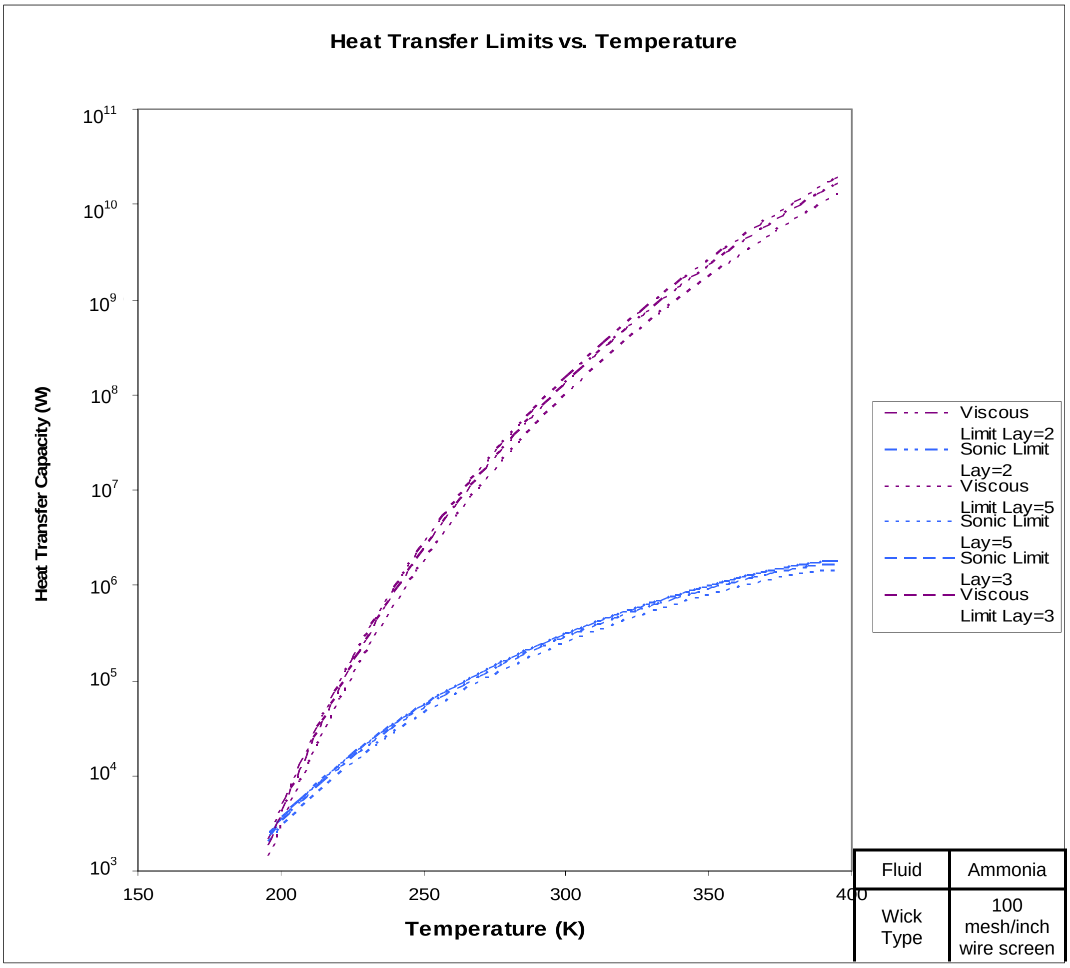


Figure 3.19– Effect of Layer Number on Viscous and Sonic Limits of Ammonia

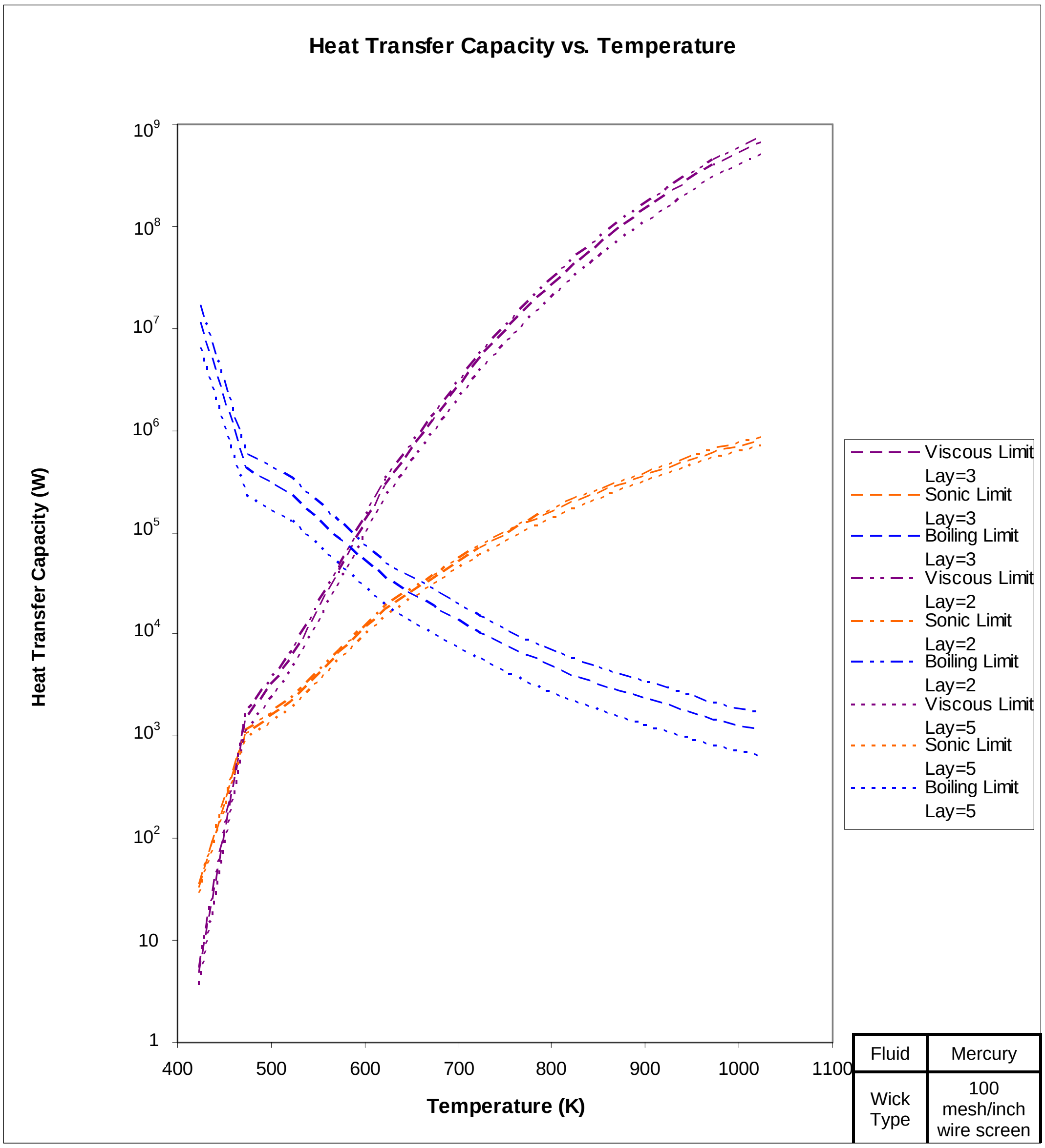


Figure 3.20– Effect of Layer Number on Viscous, Sonic and Boiling Limits of Mercury

In the analysis explained above, the effect of decreasing/increasing the layer number of 100 mesh copper screen wick structure from three to two/five is examined. The results show that by decreasing/increasing the layer number, heat transfer capacities of all limits except capillary limit increase/decrease respectively. The capillary limit decreases/increases respectively as well.

Two parameters are investigated in the analyses; mesh number and layer number. Increasing or decreasing these parameters has different effects on the heat transport capacity of heat pipe, in terms of limits. In the analysis, it is seen that entrainment limit and capillary limit are two dominant limits that determines the heat transfer capacity of a heat pipe for Water and Mercury, except at lower temperatures. It is the boiling and capillary limits for Ammonia.

The Figure 3.21 shows effects of mesh number and layer number on entrainment and capillary limits for Water. The selected temperature for the figure is 335 K (Psat =21.8 kPa).

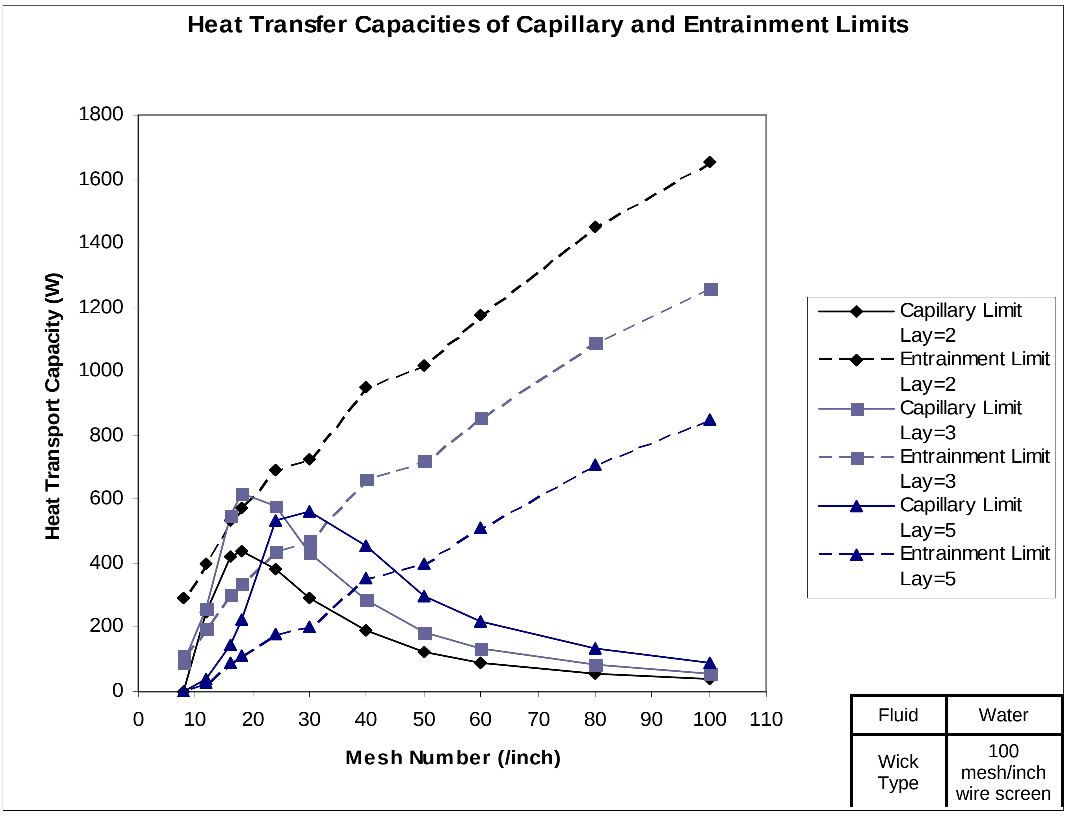


Figure 3.21 – Changes in Capillary and Entrainment Limits for Heat Pipes Having 2, 3 and 5 Layers Wick Structures, and Various Mesh Numbers, for Water

If the Figure 3.21 is investigated, for operating temperature of 335 K, the Water – copper heat pipe can provide maximum heat transfer at 24 meshes/inch copper wire screen with 2 layers. It is 380 W.

The layer number and mesh number effects on capillary and entrainment limits of Mercury are shown in Figure 3.22.

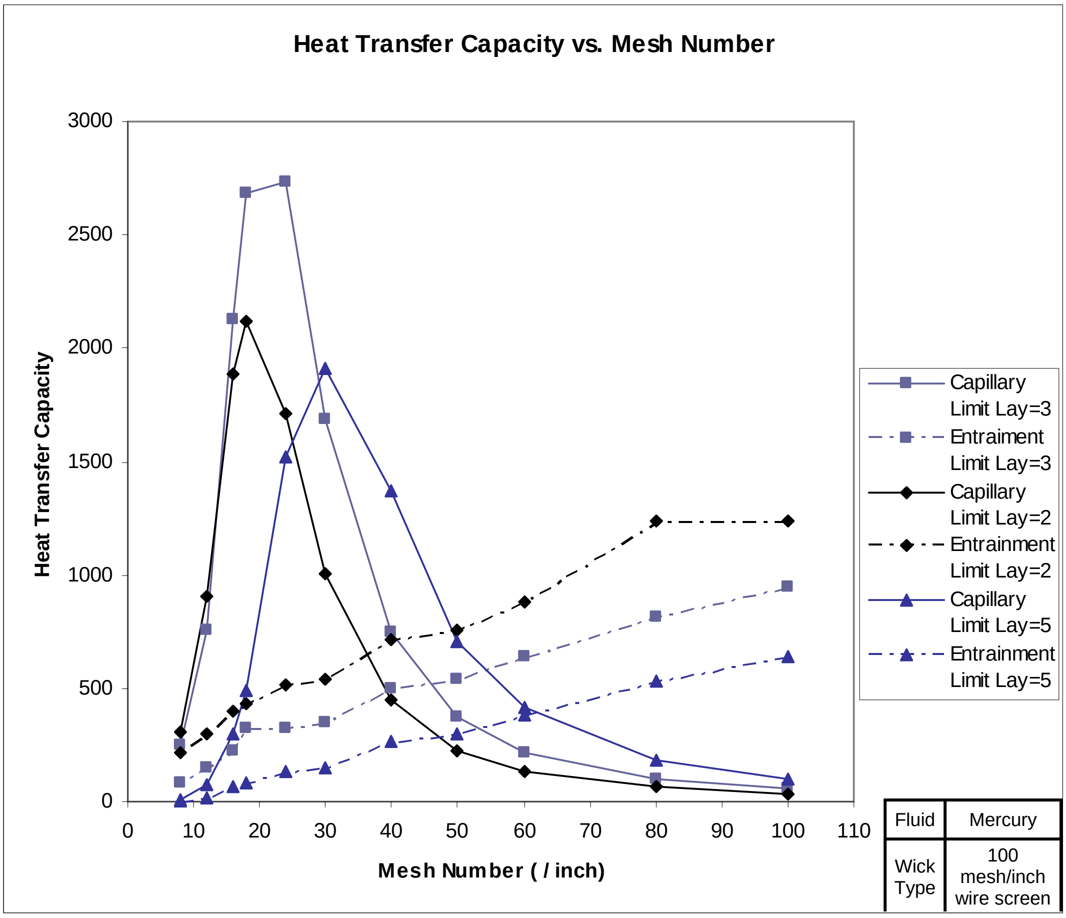
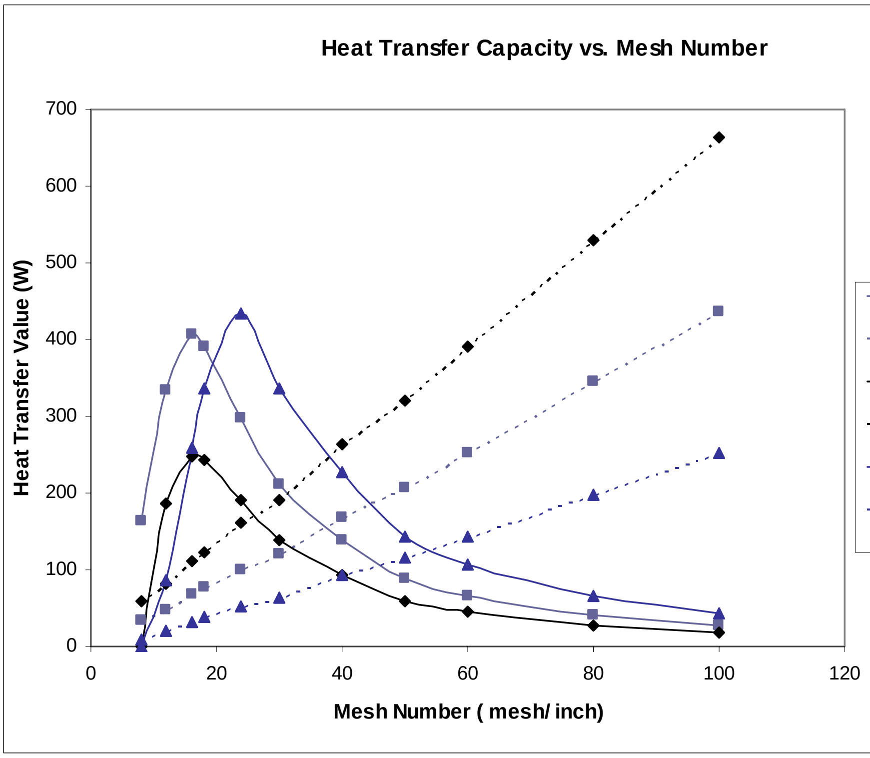


Figure 3.22 – Changes in Capillary and Entrainment Limits for Heat Pipes Having 2, 3 and 5 Layers Wick Structures, and Various Mesh Numbers, for Mercury

For Mercury, using 30 meshes/inch wick structure with two layers gives the maximum heat transfer capacity. It is the entrainment limit that governs the capacity and it gives 544 W heat capacity at 673 K.

The situation is a bit different for Ammonia. It is the boiling and capillary limits that set the heat transfer values for Ammonia. The values of these limits with respect to layers and mesh numbers are given in Figure 3.23.



Fluid	Ammonia
Wick Type	100 mesh/inch wire screen

Figure 3.23 – Changes in Capillary and Boiling Limits for Heat Pipes Having 2, 3 and 5 Layers Wick Structures, and Various Mesh Numbers, for Ammonia

The maximum capacity of the Ammonia heat pipe can be gained by using two layers of 24 meshes/inch wick structure at 245 K operating temperature. In that case, the boiling limit allows 190 W of capacity.

The maximum heat transfer values with corresponding layer and mesh numbers are given in Table 3.7 for three fluids.

Table 3.7 – Maximum Performances of Working Fluids

Working fluid	**Selected Operating Temperature (K)**	**Saturation Pressure**	**Layer Number**	**Mesh Number (meshes /inch)**	**MaximumHeat Transfer Value (W)**	**Orientation**
Water	335	21.8 kPa	2	24	380	Horizontal
Ammonia	245	130 kPa	2	24	190	Horizontal
Mercury	673	242 kPa	2	30	544	2 degrees favorable tilt

3.1.1.3 Analysis for Tilt Angle

The orientation is important for the operation of a heat pipe. Depending on conditions, a heat pipe can operate in horizontal position or in vertical position. For the horizontal position of a heat pipe, gravity has no effect. But in vertical position gravity can assist or oppose to the operation of the heat pipe.

The tilt of a heat pipe is classified into two types; favorable tilt and adverse tilt. Favorable tilt is the tilt position where gravity assists heat pipe operation. In favorable tilt, condenser is positioned above evaporator. By this way, liquid return from condenser to evaporator is assisted by gravity. Therefore, capillary pumping pressure can overcome more pressure losses and this increases the heat transfer capacity of the heat pipe, in terms of capillary limit. Other type is adverse tilt. In this tilt condition, evaporator is positioned above condenser. Therefore, the liquid in the condenser shall overcome gravity force to return to evaporator. This creates extra drag for capillary pumping pressure to overcome. As a result, heat transfer capacity of the heat pipe decreases. Therefore, it is preferable for a heat pipe to operate in favorable tilt position, if possible. The positions for adverse and favorable tilt are shown in Figure 3.24.

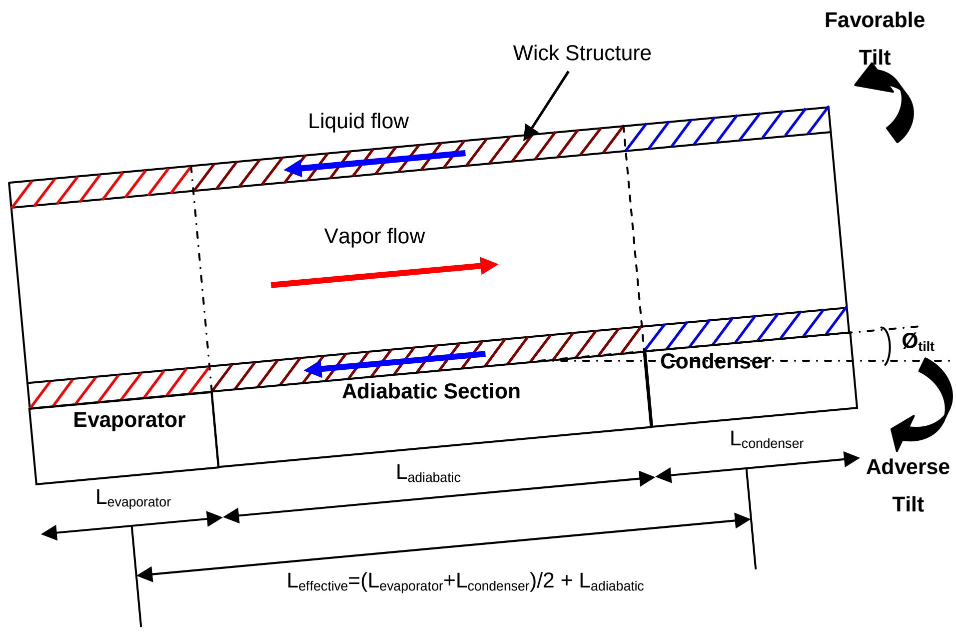


Figure 3.24 – Tilt Angle Positions for a Heat Pipe

The analyses are done to see the effect of tilt angle on heat pipe. The baseline configuration with two layers is used in the analyses. The heat pipe used for tilt angle analyses have a total length of 0.75 m. with sections having equal lengths. The diameter of the pipe is set as 0.020 m. with a wall thickness of 0.0025 m., and two layers of wire screen meshes with 100 mesh/inch are used as wick structure. Analyses are performed for three different working fluids; water, ammonia and mercury. Initially, the heat pipe is at horizontal position. Then, the tilt angle is increased. The angles up to -180 degrees represent adverse tilt. The angles up to +180 degrees represent favorable tilt.

The results presented in Figure 3.25 show tilt angle effect for the heat pipe having Water as working fluid. It shows that the favorable tilt increases the heat transfer capacity of capillary limit. At maximum favorable tilt, +90 degrees, when the heat pipe is in vertical position and condenser is positioned above evaporator, the maximum heat transfer capacity of heat pipe is about 240 W at 335 K (Psat =21 kPa). When the heat pipe horizontal position is considered, the maximum capacity of the pipe is about 37 W at 335 K. From these values, it can be concluded that tilt angle has an important effect on heat transfer capacity of capillary limit for a heat pipe. While the favorable tilt has increasing effect on capacity of heat pipe, adverse tilt has the opposite effect, as expected. It decreases the capacity of the heat pipe. Moreover, the Water heat pipe can operate up to -10 degrees at adverse tilt position. This means that the heat pipe can still transmit 2,7 W of heat for adverse tilt position of 10 degrees.

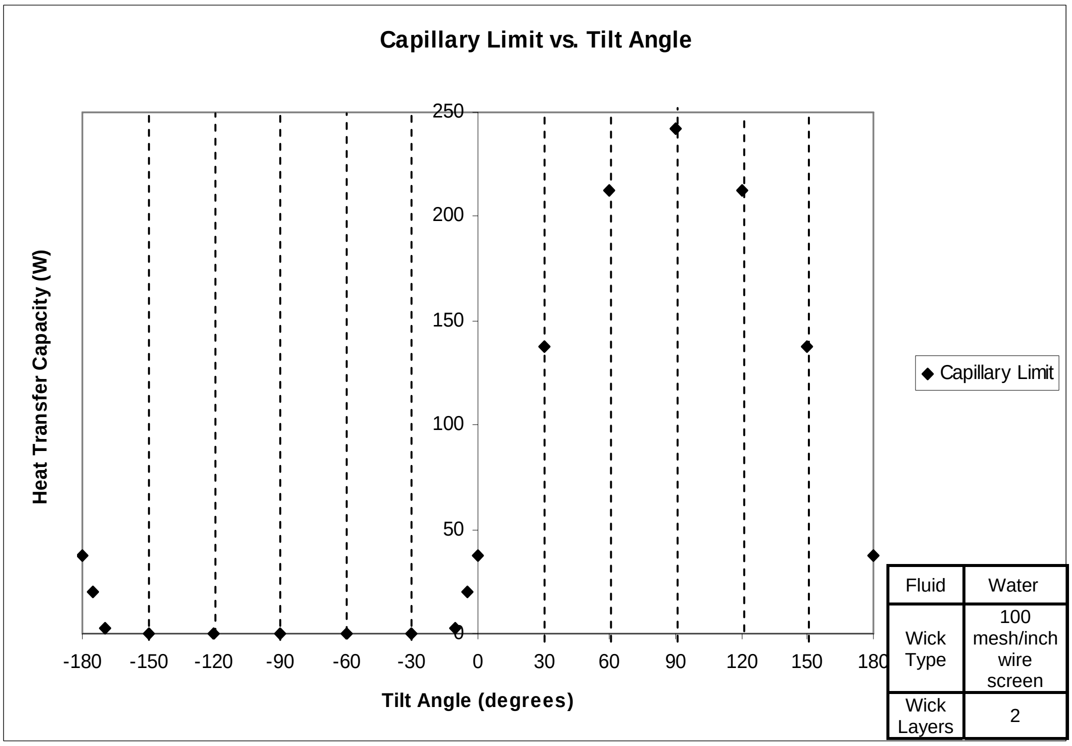


Figure 3.25 – Tilt Angle Effect on Capillary Limit at 335 K, for Water

The tilt angle has the similar effect for the same heat pipe configuration, using Ammonia as working fluid. The results are shown for 245 K temperature in Figure 3.26. The favorable tilt angles increase the heat transfer capacity; whereas the adverse tilt angles decrease the capacity. The Ammonia heat pipe can operate up to 10 degrees adverse tilt, with specified configuration. The maximum capacity reaches to 124 W, and it can transmit 0.2 W of heat at 10 degrees adverse tilt angle.

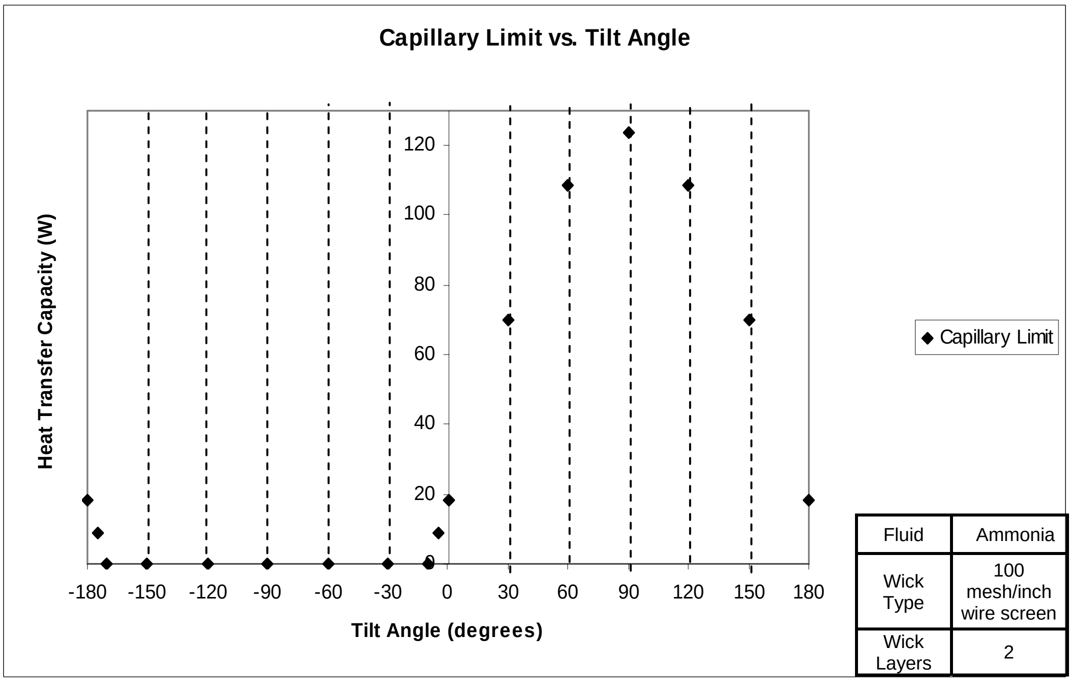


Figure 3.26 – Tilt Angle Effect on Capillary Limit at 245 K, for Ammonia

The tilt angle has the same effects on Mercury. The important thing for Mercury is that it can not operate at horizontal position or for adverse tilt conditions. It needs at least two degrees of favorable tilt to operate. The results are shown in Figure 3.27. for the temperature of 673 K.

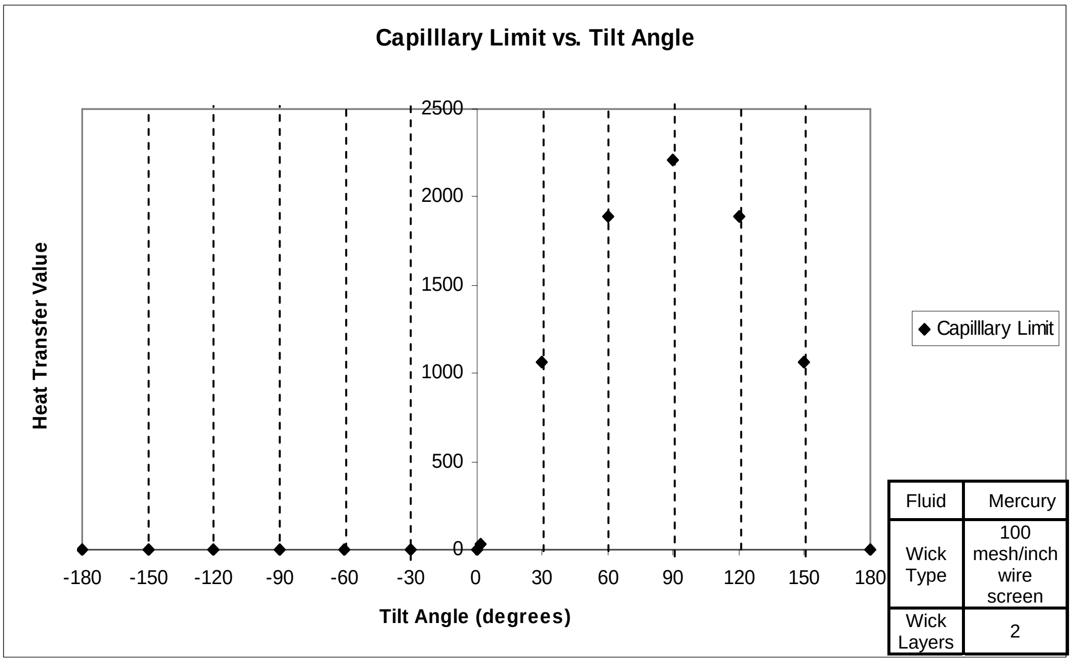


Figure 3.27 – Tilt Angle Effect on Capillary Limit at 673 K, for Mercury

To sum up, the tilt angle has an important effect on the performance of a heat pipe. The analyses are performed for three different working fluids and the same effects are seen for all fluids. The tilt angle affects the capillary limit of the heat pipe. For adverse tilt positions, capillary limit decreases and for favorable tilt positions, it increases. The maximum heat capacity can be gained for +90 degrees favorable tilt position. The performances of three heat pipes are presented in the Table 3.7 for various tilt angles.

The Mercury heat pipe can operate at least 2 degrees favorable tilt conditions. Therefore, the heat transfer capacity at horizontal orientation is 0 W. Similarly, it can nt operate at adverse tilt conditions and maximum adverse tilt angle is 0°.

Table 3.8 – Performances of Working Fluids for Tilt Angles

Workfing fluid	Selected Operating Temperature (K)	Heat Transfer at Maximum Tilt Angle (W)	Maximum Adverse Tilt Angle (Degrees)	Heat Transfer at Maximum Adverse Tilt (W)	Heat Transfer at Horizontal Position (W)
Water	335	240	10	2.7	37
Ammonia	245	124	10	0.2	18
Mercury	673	2200	0	0	0

3.1.1.4 Analysis for The Effect of the Pipe Length and Diameter

Regardless of the wick structure composition and the orientation, the physical properties of the pipe have significant effect on the performance of a heat pipe. These physical properties are the diameter and the length of the pipe. In the literature, it is mentioned in several studies that it is better for a heat pipe to have less length and higher diameter in terms of performance. This means that the heat pipe performance increases as the length of the pipe is decreased and the diameter of the pipe is increased.

In order to see the effect of physical pipe properties, analyses are performed. In the analyses, the effects of diameter and pipe length are investigated individually.

3.1.1.4.1 Analysis for the Effect of the Pipe Length

The first analyses are performed to see the effect of the pipe length on the capacity of the heat pipe. For this purpose, the same configuration of the heat pipe used in mesh number analyses is used. The heat pipe has a pipe diameter of 0.020 m. with 0.0025 m. wall thickness, and 0.75 m. total length initially. As wick structure, 2 layers of wire screens with 100 mesh/inch is used. The orientation of the pipe is horizontal. The length of the pipe is changed by changing the evaporator, condenser and adiabatic section lengths.

The results of the analyses are shown in Figure 3.28, for changing the each sections length equally. The capillary limit values are presented since it is the governing limit for all three fluids.

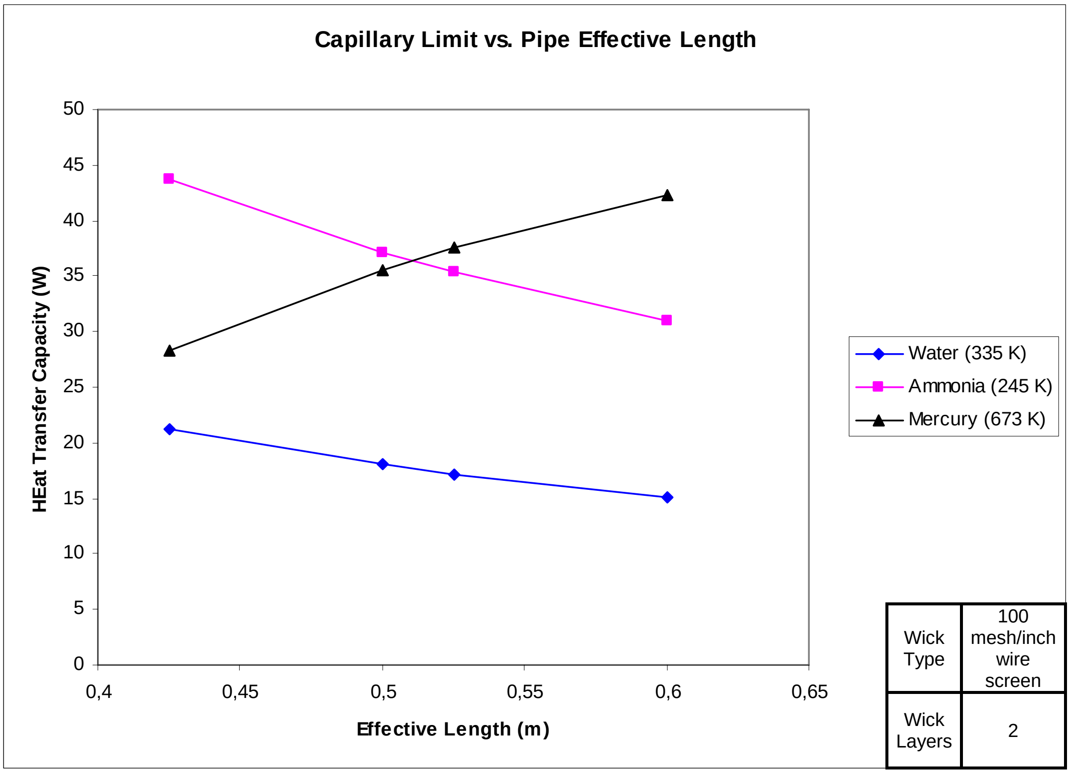


Figure - 3.28 Effect of Pipe Effective Length on Capillary Limit

The effective length of the pipe affects the capillary limit. If the effective length does not change, the capacity of the capillary does not change. As seen from Figure 3.31, Ammonia and Water lines have negative slope, whereas the Mercury line has positive slope with respect to pipe length. The slope of Water line is -40, the slope of Ammonia line is -80 and the slope of Mercury line is 80. This means that as the effective length of the pipe increases, the capacity of Water and Ammonia heat pipes decrease. Oppositely, if the effective length decreases, the capacities of the heat pipes using these fluids increase as well. This can be seen well if Equation 2.3 is investigated. As the length of the pipe increases, the losses also increase and this causes decrease in capacity. The situation is a bit different for Mercury. The results are shown for two degrees of favorable tilt for Mercury heat pipe, since it can not operate at horizontal conditions. Therefore, as the effective length increases, the pressure gain due to gravity increases. However, the losses also increase. But, since the density of the Mercury is very high, the pressure gain due to gravity becomes larger than the increase in losses, resulting in an increase in the capacity of capillary limit.

The entrainment limit and sonic limit do not be affected from the pipe length. The viscous limit decreases as effective length increases and increases as the length decreases. The boiling limit depends on the evaporator section length. If the length of the evaporator section decreases, values of the boiling limit decrease, too. Consequently, if the evaporator length is increased, the values are also increase for boiling limit.

3.1.1.4.2 Analysis for the Effect of the Pipe Diameter

The next physical property of the pipe is pipe diameter. The heat pipe configuration used as baseline is the same; a heat pipe having total length of 0.75 m. with diameter of 0.020 m. with 0.0025 m. wall thickness. The orientation is horizontal and two layers of wire screen with 100 mesh/inch is used as wick structure. Water, ammonia and mercury are used as working fluids. To see the effect of pipe diameter, the diameter is increased and decreased by keeping other parameters constant.

Firstly, the effect of the pipe diameter on water-copper heat pipe is investigated. The results are shown in Figure 3.29 for an operating temperature of 335 K (Psat=21.6 kPa).

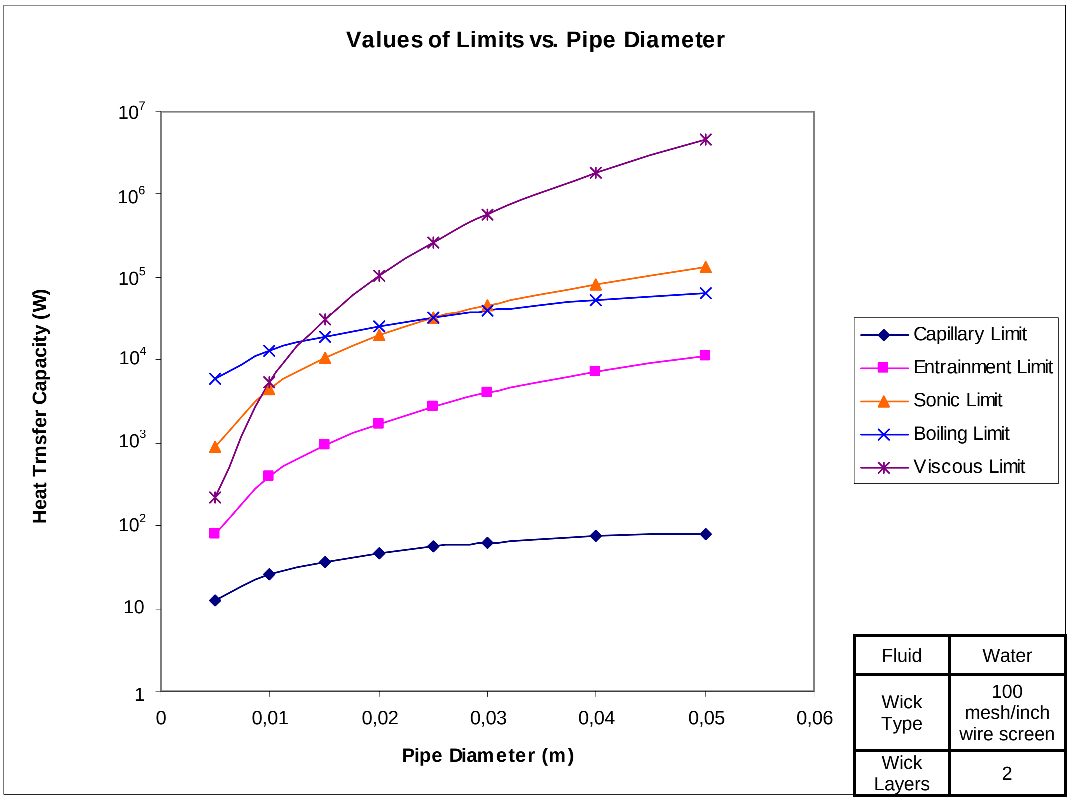


Figure 3.29 – The Effect of the Pipe Diameter on the Water Heat Pipe at 335 K

As seen from the Figure 3.29., the decrease in the diameter of the pipe causes decrease in all limits, whereas increase in diameter results in an increase in capacity. When the pipe diameter is decreased, the vapor flow area is decreased too. Therefore, the losses in the vapor flow increases and this causes a decrease in all limits. This is the mentioned situation in the literature. As the pipe diameter decreases, the capacity of pipe decreases, too. On the other hand, increasing the diameter means increasing the vapor flow area. This allows more vapor to flow from evaporator to condenser, resulting an increase in the mass flux. As the mass flux increases, the heat that is transferred also increases.

The results of the analyses performed for Ammonia and Mercury are shown in Figures 3.30 and Figure 3.31 respectively. As seen from the figures, the effect of the pipe diameter change is similar to Water heat pipe for Ammonia. The increasing diameter increases the capacity of the heat pipe, in terms of all limits. However, the decreasing diameter decreases the capacity.

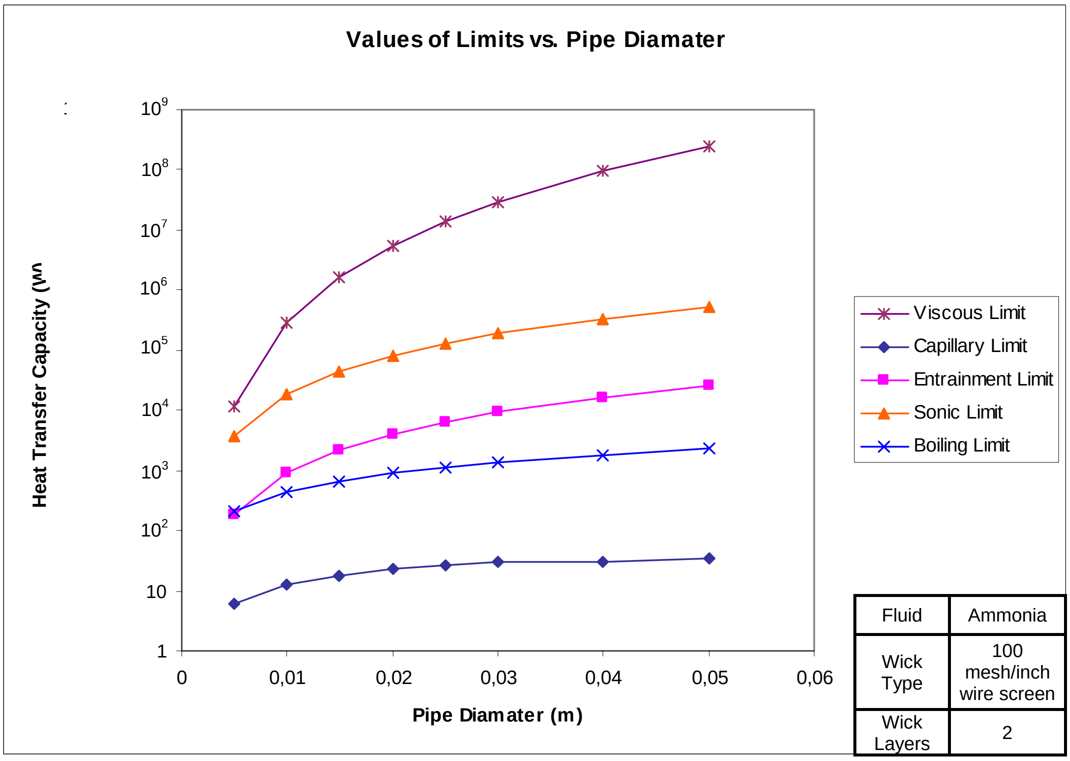


Figure 3.30 – The Effect of the Pipe Diameter on the Ammonia Heat Pipe at 245 K

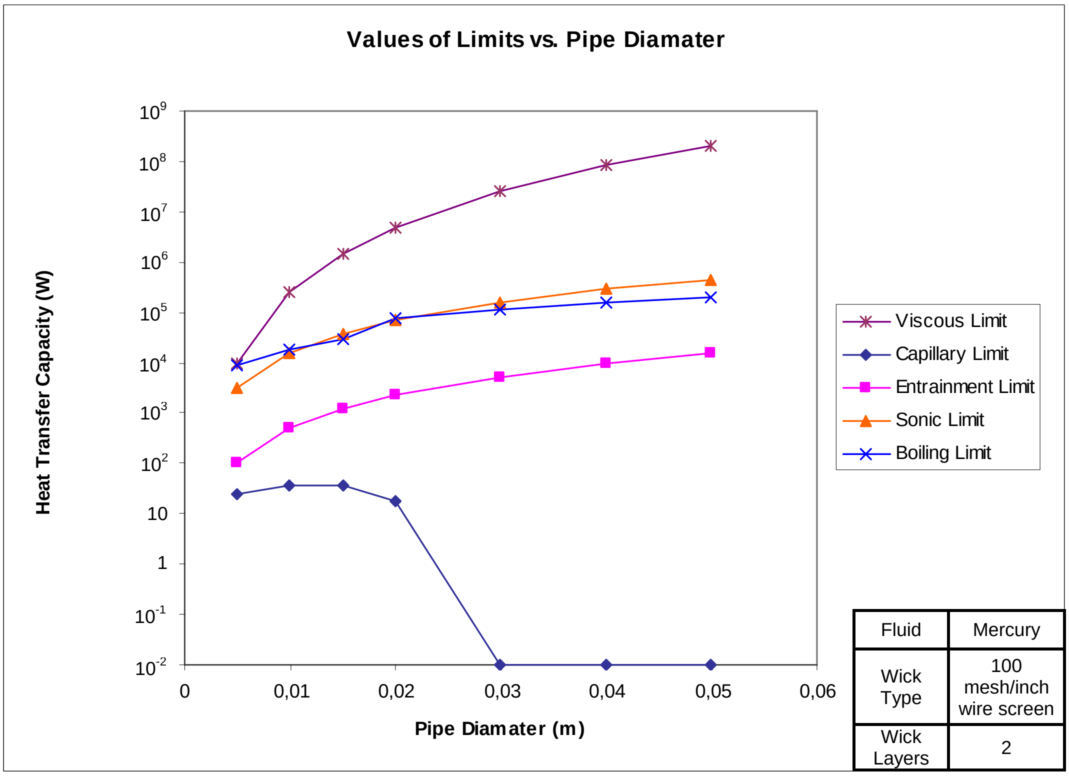


Figure 3.31 - Effect of Pipe Diameter for Mercury Heat Pipe at 673 K

Figure 3.31 shows that Mercury heat pipe behaves different than Water and Ammonia in terms of capillary limit. All other limits increase/decrease as diameter increase/decrease respectively. However, the capillary limit shows an increasing – decreasing trend as the diameter increases from 0.005 m. (minimum diameter) to 0.05 m.(maximum diameter). It increases up to 0.01 m. diameter and shows a decreasing trend after that diameter. It falls to zero as diameter continues to increase. This situation can be explained when Equation 2.3 is investigated.

$$q \leq \frac{\frac{2\sigma}{r_{ce}} - \rho_l . g(d_v . \cos\psi + L_{eff} . \sin\psi)}{\left(\frac{C.f_v.\mathrm{Re}_v.\mu_v}{2.r_{hv}^{\ 2}.A_v.\rho_v.\lambda} + \frac{\mu_l}{K.A_w.\lambda.\rho_l}\right).L_{eff}} \tag{2.3}$$

When the diameter of the pipe is increased, the diameter of the vapor space and the vapor flow area inside the pipe increases since the wick area is kept constant. The increase in vapor flow area increases the capacity. However, increase in vapor space diameter causes decrease in capacity. Due to the high liquid density of mercury, the decrease caused by vapor space diameter becomes larger than the increase caused by vapor flow area. Therefore, decrease in capillary limit occurs. This situation exists when the diameter of the pipe exceeds 0.01 m.

In order to see the effect of diameter change on the capacity of heat pipe, a heat pipe composed of 0.75 m. length, with an initial diameter of 0.015 m. is investigated. Two layers of 100 mesh/inch copper wire screen are used as wick structure in the heat pipe. The diameter is decreased and increased incrementally. As a result, it is seen that the capacity of the heat pipe increases with an increase in diameter, and it decreases as the diameter of the pipe is decreased. These are convenient with the results mentioned in the literature.

3.2 Open Groove Wick Analyses

Open groove wick structure is analyzed as second type of wicks. In this section, the analyses performed using open grooves are presented and the results are discussed.

The open grooves wick is composed of grooves that are drilled into the wall of the pipe or another material that is stick to pipe wall. The grooves provide the capillary pumping pressure and the liquid flow occurs inside these grooves. The wick structure is shown in Figure 3.32 [18].

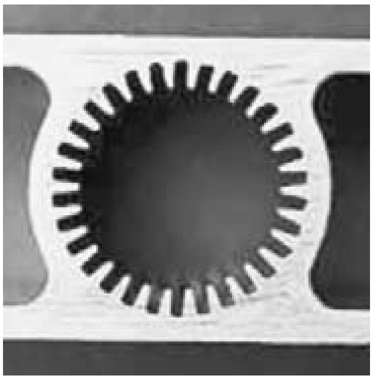
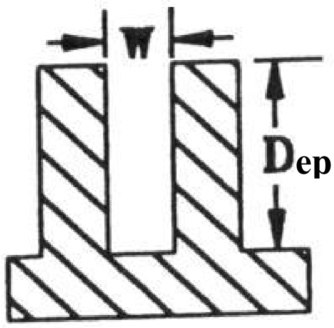


Figure 3.32 – Open Groove Wick Structure (Adapted from [18])

The important factors of the open grooves are the depth, the width and the number of the grooves. These form the groove area (wick are). In the analyses, individual effects of groove width, groove depth and groove number are investigated. The analyses are performed for square grooves and the rectangular grooves. It is the width and depth of the groove that determines whether it is square or rectangular. The groove structure is shown in Figure 3.33 [21].



W: groove width

Dep: groove depth

Figure 3.33 – Groove Structure (Adapted from [21])

Finally, the performance of open grooves wick is compared with the performance of wire screen meshes.

In the Figure 3.34, the performance of the Water heat pipe having initial configuration is shown in logarithmic scale.

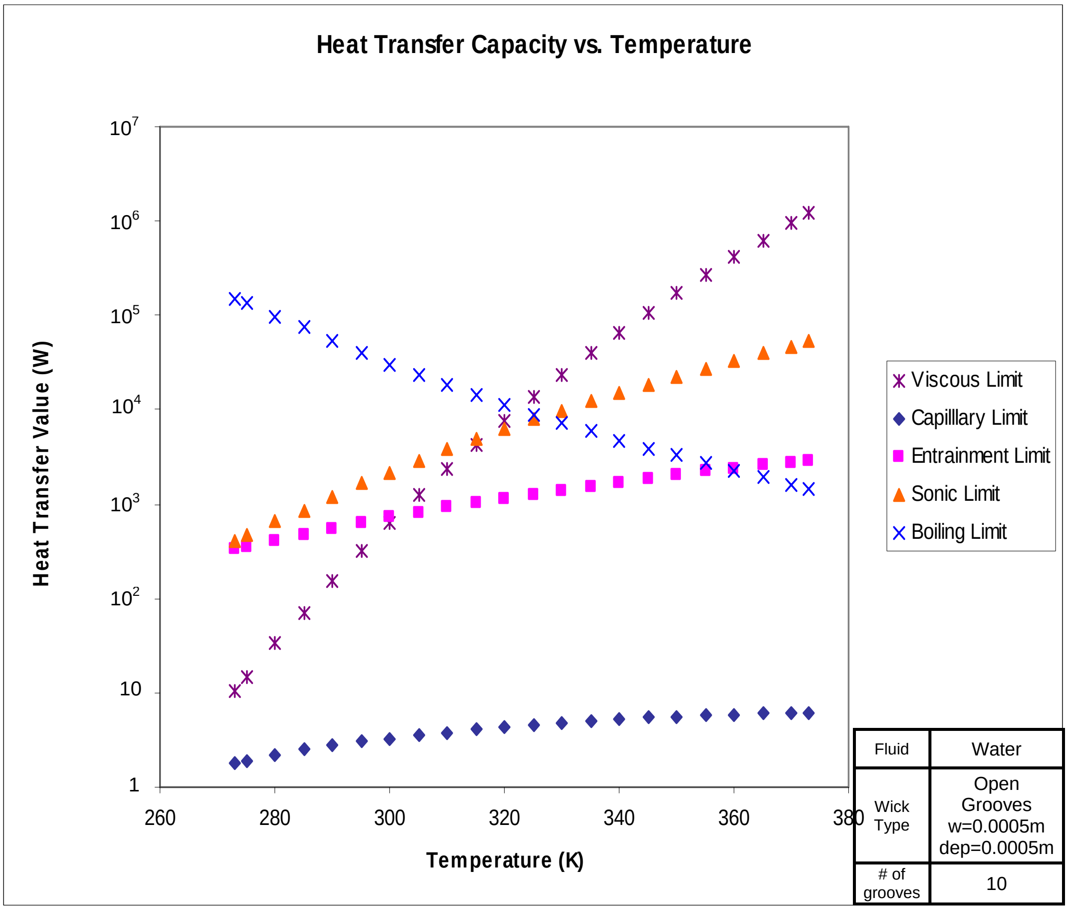
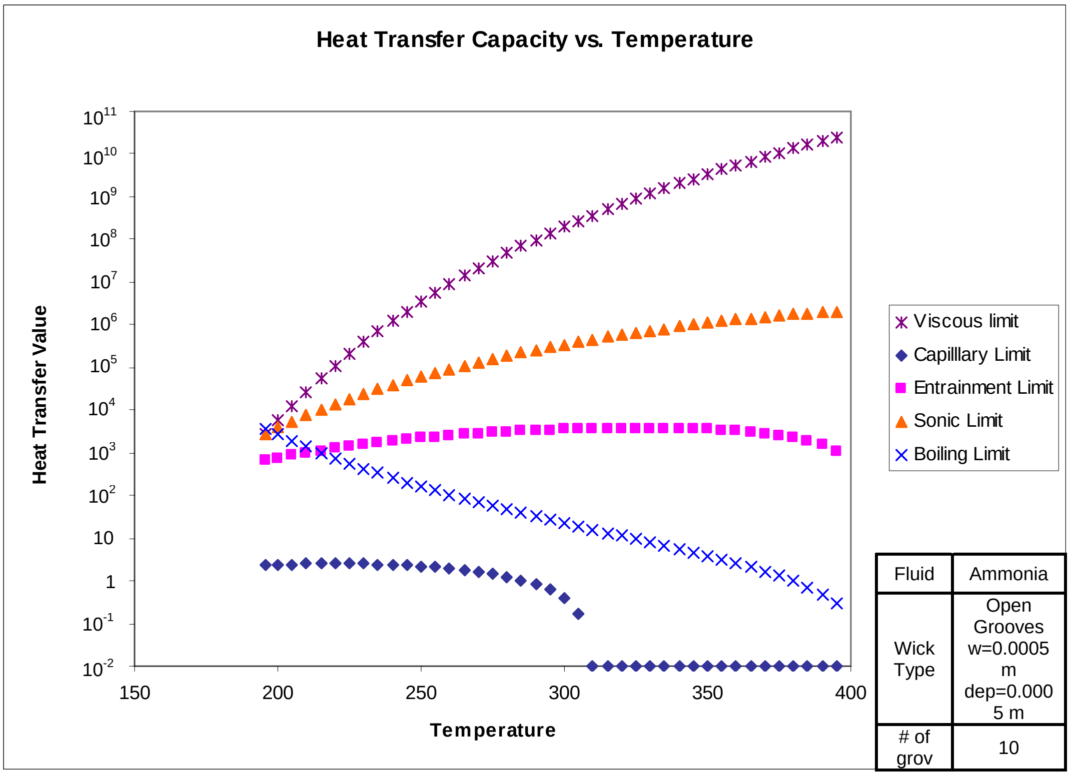


Figure 3.34 – Performance of the Heat Pipe for Open Grooves for Water

If Figure 3.34 is investigated, it can be seen that the behavior of the heat pipe is similar to the configuration with wire screen meshes. But in this case the capillary limit is the dominant limit for the whole temperature range and it determines the heat transfer capacity of the pipe.

For ammonia, the results of initial configuration can be seen in Figure 3.35. The results show that the capillary limit is the determiner for the capacity of the ammonia heat pipe, as in the wire screen mesh wick structure. Moreover, the values of all limits except boiling limit increases as temperature increases.



Fluid	Ammonia
Wick Type	Open Grooves w=0.0005 m dep=0.0005 m
# of grov	10

Figure 3.35 – Performance of the Heat Pipe for Open Grooves for Ammonia

The performance of the Mercury heat pipe with open grooves wick structure is shown in Figure 3.36. The viscous limit is important for lower temperatures. As temperature increase, the capillary limit shows its effect.

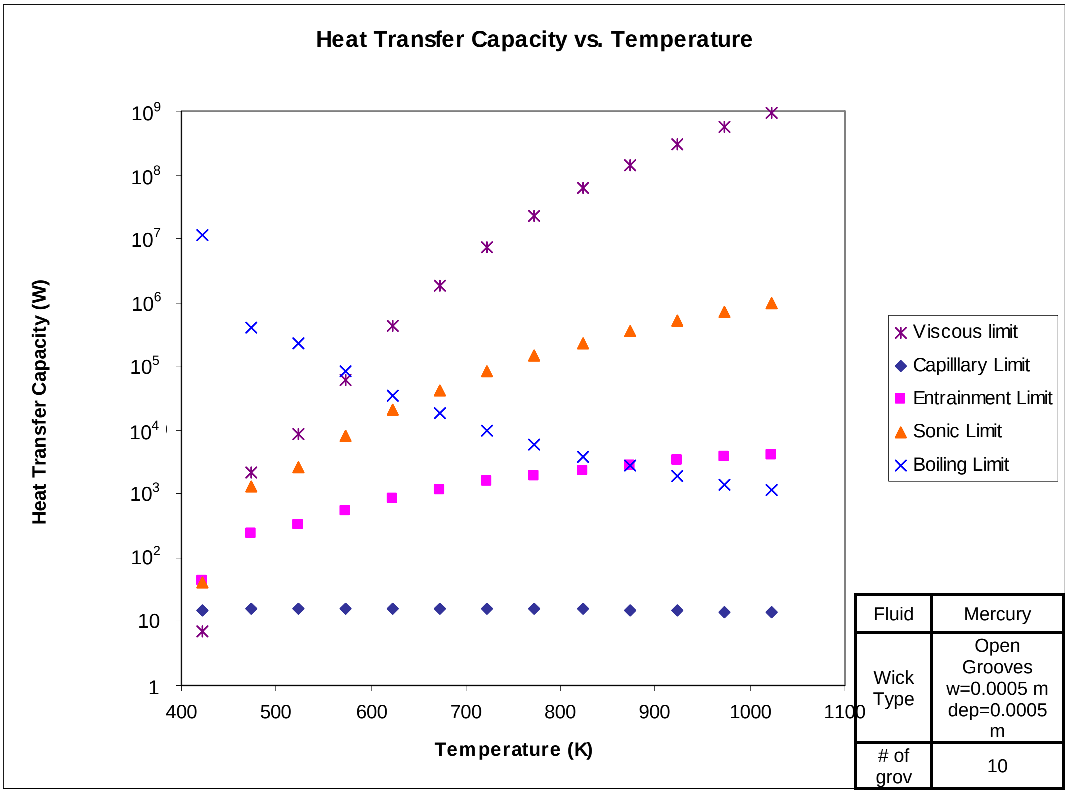


Figure 3.36 – Performance of the Heat Pipe for Open Grooves for Mercury

3.2.1 The Effect of Groove Width

One of the important parameters for open grooves is the width of the grooves. In the analyses, the heat pipe configuration of 0.75 m. pipe length, 0.020 pipe diameter with the wall thickness of 0.0025 m. is used. Analyses are performed for all three working fluids; Water, Ammonia and Mercury. The pipe orientation is set as horizontal (except for Mercury). Two degrees of favorable tilt is provided for mercury heat pipe since it can not operate under horizontal orientation. The depth and width of the grooves are set as 0.0005 m. initially. The open grooves wick is composed of 10 grooves.

To see the groove width effect, the width of the grooves are increased to 0.001 m. and decreased to 0.00025 m. respectively. The results of the analyses showing the groove width effect are shown in Figure 3.37. In the figure, only the capillary limit values are shown since it is the governing limit for the average operating temperatures of all three fluids.

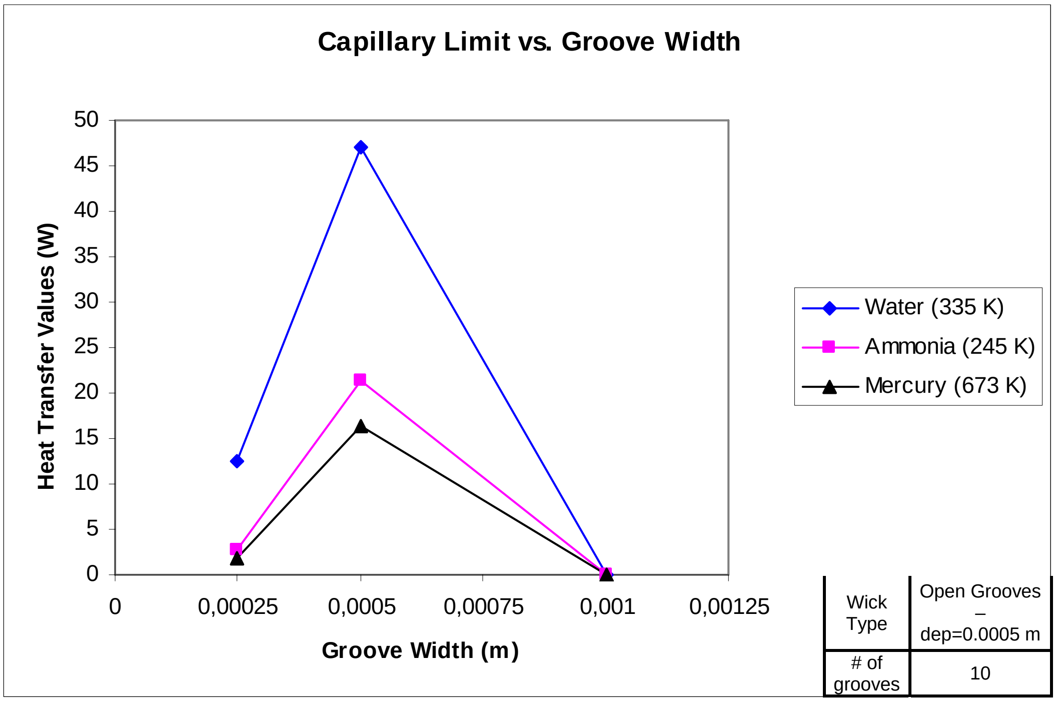


Figure 3.37 – Groove Width Effect on Capillary Limit

The change in groove width affects the groove area and therefore total wick area. As a result, the capillary pumping pressure and the frictional losses decrease or increase according to changes in groove width. Therefore, the capacity of the capillary limit is affected by groove width.

As seen from the Figure 3.37, increasing the groove width from 0.0005 m. to 0.001 m. causes significant decrease on the performance of the heat pipes of all three fluids. On the other hand, decreasing the groove width from 0.0005 m. to 0.00025 m. causes decrease in the capillary limit of all three working fluids. The operating temperature of the fluids are different and mentioned in the parenthesis.

Decreasing or increasing the groove width has a decreasing or increasing effect respectively, depending on the fluid type used. The viscous and sonic limits are not affected from groove width change for all three working fluids; entrainment and boiling limits show an increasing or a decreasing behavior.

3.2.2 Groove Depth Effect

Another parameter affecting the heat pipe performance is the groove depth. Like groove depth, it affects the groove and wick area. Similar to the analyses made for groove width, the groove depth is decreased and increased respectively, to see the individual effects. Figure 3.38 shows the effect of groove depth on the performance of the heat pipe. Again the results are shown for capillary limit.

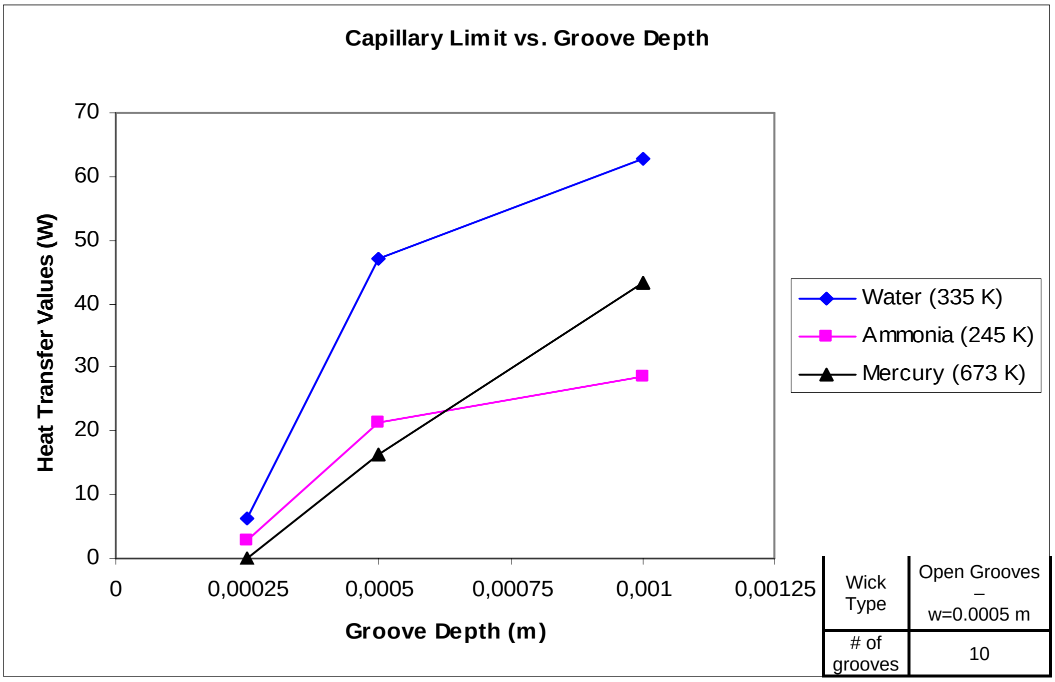


Figure 3.38 – The Effect of Groove Depth on the Performance of Heat Pipe

If the Figure 3.38 is investigated, it can be concluded that groove depth is more effective on capillary limit than the groove width. The effects are also different. As the groove depth increased, the capillary limit for all three liquids increases, of course with different amounts. On the contrary, if the groove depth is decreased, the capillary limit also decreases.

The changes in the groove depth have different effects on other limits. The viscous and sonic limits do not been affected by the groove width change, either it is increased or decreased. Opposite to the capillary limit, the decrease in groove width results in an increase in boiling and entrainment limits. Similarly, as the groove depth increased, a decrease occurs in boiling and entrainment limits. These are observed for all three fluids.

3.2.3 Combined Effect of Groove Width and Groove Depth

Up to now, the individual effects of decreasing or increasing groove width and depth are investigated. What can happen if the square groove formation is preserved by decreasing or increasing the width and the depth of the groves at the same time, with the same amount? In Figure 3.39 the combined effect of groove properties are shown, in terms of capillary limit. Only the capillary limit is shown since it is the governing limit for open grooves case of all three working fluids.

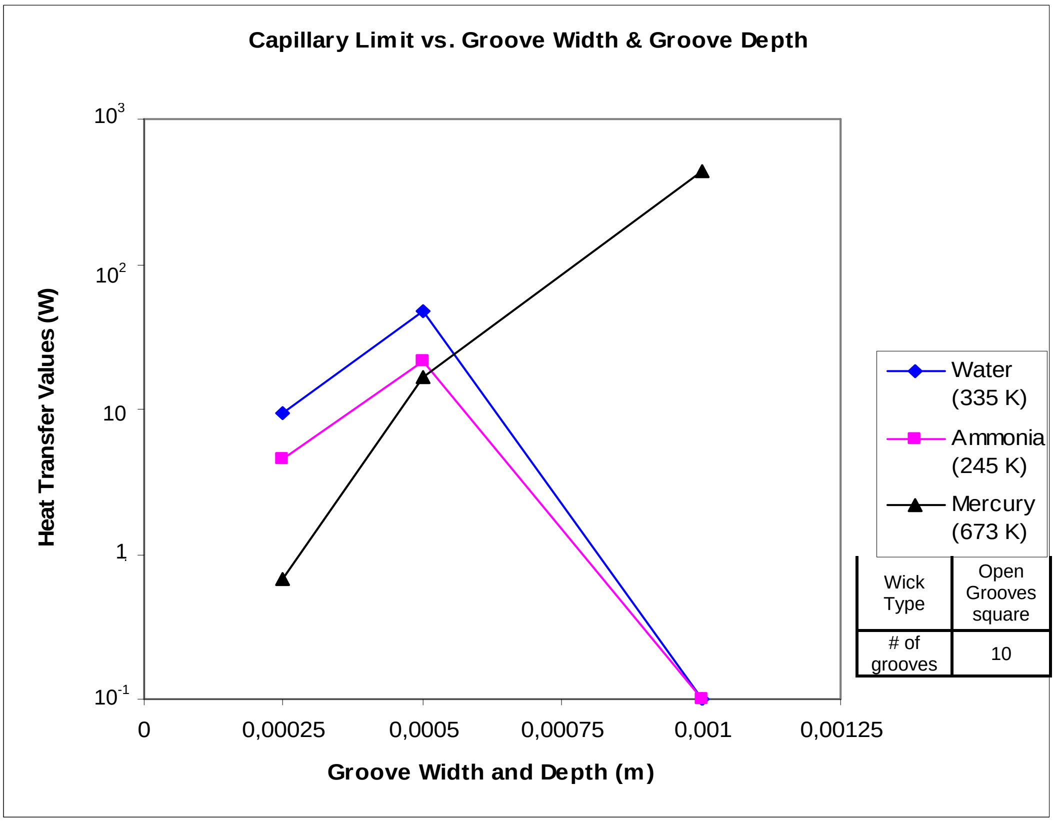


Figure 3.39 – Combined Effect of Groove Width and Groove Depth

As seen from Figure 3.39, as the square formation of the groove is kept, the effect of decreasing or increasing the depth and width of the grooves together has different effects. Increasing the depth and width causes decrease in capillary limits of Water and Ammonia. However, it increases the capillary limit value of Mercury. Decrease in the depth and width of the grooves causes decrease in capillary limits of all three working fluids.

3.2.4 Effect of Groove Number

The final property of the open grooves that affect the heat pipe performance is the number of grooves. The analyses are performed on a 0.75 m. long heat pipe, with pipe diameter of 0.020 m, having wall thickness of 0.0025 m. Open grooves are used as wick structure and groove width and groove depth are set as 0.0005 m. Analyses are performed by increasing and decreasing the groove number, using water, ammonia and mercury as working fluids.

The results are shown for the capillary limit only, as it is the governing limit that governs the heat pipes of water – copper, ammonia – aluminum and mercury – steel for open grooves wick structure. Figure 3.40 shows the results for three working fluids. The operating temperatures selected as 335 K (P_{sat} =21.6 kPa) for water, 245 K (P_{sat} =130 kPa) for ammonia and 673 K (P_{sat}=242 kPa) for mercury.

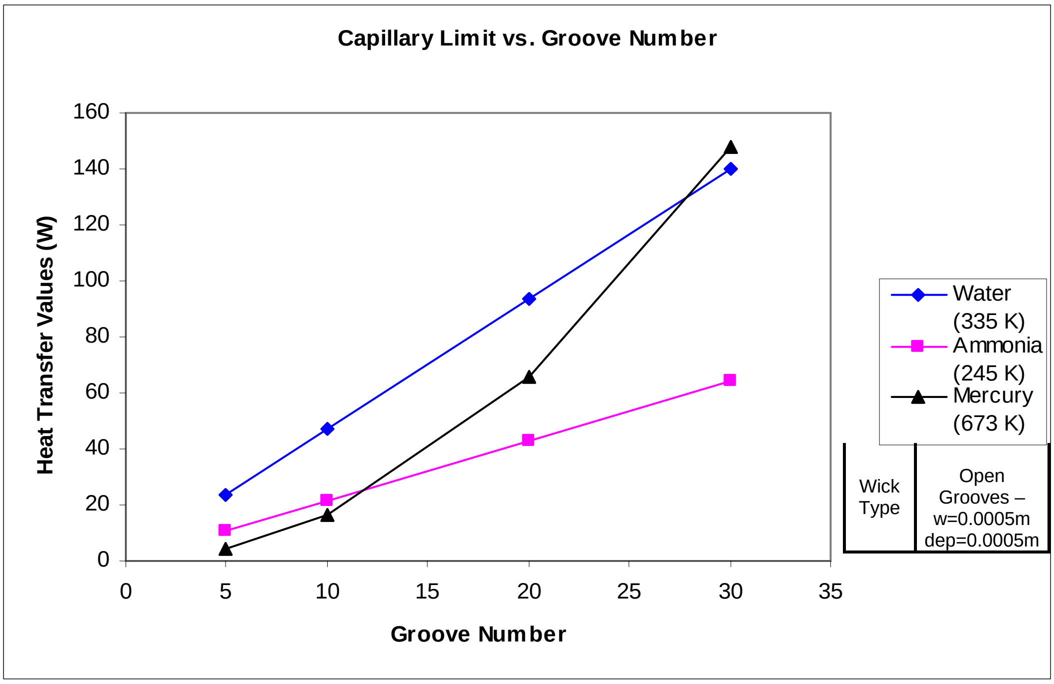


Figure 3.40 – The Effect of Groove Number on the Capillary Limit

Analyses are done to see the effect of increasing groove number from 10 to 20 and 30. To see the groove number decrease effect, the groove number decreased from 10 to 5. If the results of the figure are compared with the results of the case where 10 grooves are used, it can be seen that the capillary limit is affected from groove number change. Increasing the groove number from 10 to 20 and 30 caused an increase in the capillary limit. However, as the groove number is decreased, the capillary limit decreases too. These results are valid for all three fluids.

Other four limits of all three heat pipes – viscous, sonic, entrainment and boiling – are not affected from the groove number changes.

The performances of the fluids for open groove wick structures are tabulated in Table 3.9. The maximum heat transfer values and corresponding open groove wick composition are summarized for the pipe configuration of 0.75 m. long with 0.020 m. diameter. The wall thickness is 0.0025 m.

Table 3.9 – The Performance of Three Heat Pipes for Open Groove Wick

Working Fluid	Selected Op. Temp. (K)	Initial Config.			Maximum Capacity for Initial Config. (W)	Max. Capacity Config.			Maximum Capacity (W)
		# of grooves	w (m)	dep (m)		# of grooves	w (m)	dep (m)	
Water	335	10	0.0005	0.0005	5.2	30	0.0005	0.0005	45
Ammonia	245	10	0.0005	0.0005	2.3	30	0.0005	0.0005	20.6
Mercury	673	10	0.0005	0.0005	16.5	30	0.001	0.001	437

3.2.5 Merit Number Comparison

The working fluid thermo physical properties are one of the main factors that affect the capacity of a heat pipe. A dimensionless number is presented by Chi including the surface tension, liquid viscosity, liquid density and the latent heat of vaporization of the working fluid. This number is called as Merit number. The equation for Merit number is given in Equation 2.4. For a heat pipe, it is better to choose working fluids with higher merit numbers since working fluids with high surface tension, high latent heat of vaporization, high liquid density and low liquid viscosity are desirable for the operation of the heat pipe.

The merit numbers for all three working fluids are presented in Tables 3.10 – 3.12.

Table 3.10 - Merit Numbers of Water

Temperature (K)	Water				
	Liquid Density (kg/m3)	Liquid Viscosity (N.m/s	Surface Tension (N/m)	Hfg (kJ/kgK)	Merit number
273	1000	0,00175	0.0755	2502	1.08×10^{11}
323	987.95	0.0005476	0.06782	2382.3	2.91×10^{11}
373	957.854	0.000279	0.0589	2257	4.56×10^{11}
335	982.318	0.000453	0.0658	2354	3.36×10^{11}

Table 3.11 - Merit Numbers of Ammonia

Temperature (K)	Ammonia				
	Liquid Density (kg/m3)	Liquid Viscosity (N.m/s	Surface Tension (N/m)	Hfg (kJ/kgK)	Merit number
196	732.37	0.00055	0.062	1464	1.21×10^{11}
245	675.51	0.000238	0.0428	1354.17	1.65×10^{11}
295	607.47	0.00013582	0.0258	1178.85	1.36×10^{11}
395	376.2	0.000047	0.0012	449.6	4.32×10^{9}
335	541.85	0.000093	0.0142	986.9	8.17×10^{10}

Table 3.11 - Merit Numbers of Mercury

Temperature (K)	Mercury				
	Liquid Density (kg/m3)	Liquid Viscosity (N.m/s	Surface Tension (N/m)	Hfg (kJ/kgK)	Merit number
423	13230	0.00109	0.0445	308.8	1.67×10^{11}
773	12308	0.0008	0.0341	291.3	1.53×10^{11}
1023	11800	0.00077	0.0275	277	1.17×10^{11}
673	12656	0.00086	0.0374	296.3	1.63×10^{11}

It is seen from the tables that the merit number of Water increases as temperature increases. This is consistent with the fact that the water heat pipe reaches its maximum capacity at 373 K. For Ammonia, the highest Merit number occurs at 245 K, at which the highest capacity of Ammonia heat pipe is reached. The situation is different for Mercury. The highest merit number occurs at the lowest temperature, 423 K, but the highest capacity occurs at 673 K.

If the tables are investigated, it is seen that for the temperature of 335 K, Water has higher merit number than Ammonia, validating that a water heat pipe has higher capacity than Ammonia heat pipe at that temperature.

Summary

The two common types of wick structures- wire screen meshes and open grooves- are analyzed for three different working fluids; water, ammonia and mercury. The performance of three different heat pipe configurations; Water – Copper, Ammonia – Aluminum and Mercury – Steel, are analyzed.

The different wick structures have different effective parameters. Numbers of meshes, wire diameter, number of layers are investigated for wire screen meshes. The groove depth, the groove width and the number of grooves are examined for open grooves.

Moreover, the effects of the tilt angle, the diameter and the lengths of the pipe are also investigated.

The maximum performances of working fluids for two types of wick structures are tabulated in Table 3.13.

Table 3.13 – Performances of Different Working Fluids with Different Wick Types

Wick Type	Water		Ammonia		Mercury	
	Wire Screen	Open Groove	Wire Screen	Open Groove	Wire Screen	Open Groove
Mesh # (meshes/inch)	24	-	24	-	30	-
Layer #	2	-	2	-	2	-
Groove #	-	30	-	30	-	30
Groove Width (m.)	-	0.0005	-	0.0005	-	0.001
Groove Depth (m).	-	0.0005	-	0.0005	-	0.001
Operating Temp. (K) & Psat	335 K 21.8 kPa		245 K 130 kPa		673 K 242 kPa	
Maximum Capacity (W)	380	45	190	20.6	544	437

If the table is investigated, it can be seen that using wire screen meshes gives better performance for all of three working fluids. Water gives the best performance for temperature range of 280 K – 373 K. Ammonia is better for lower temperatures, such as cryogenic applications and Mercury can be used for higher temperature applications.

CHAPTER 4

HEAT PIPE APPLICATIONS

Heat pipes can be used for various applications, with a wide range of operating temperature. According to the operating temperature of the heat pipe, appropriate working fluid and wick type selections shall be done. The optimum performance can be provided with the selection of the working fluid and the wick structure. The thermophysical properties of the working fluid are important factors that affect the performance of the heat pipe. Likewise, the physical properties of the wick structure affect the heat transfer capacity of the pipe.

Three different applications of heat pipes are investigated. First application is an electronic cooling application. Water – copper heat pipe is used to cool the CPU of the notebook computer. Secondly, a cryogenic application like satellites is examined. Appropriateness of the ammonia – aluminum heat pipe is checked. Finally, for higher temperatures, mercury – steel heat pipe is analyzed.

4.1 Electronics Cooling Application

The thermal management of the equipments is an important issue for electronics area. The thermal management of microprocessors, cards etc. become important and important as the performance of the equipments increases. May be the best simulation of the thermal management problem can be seen in computers. As the performance of the CPUs of computers become higher, the heat that is dissipated from the processor increases. The matter is taking the heat out of computer casing. General approaches like using copper heat sinks need larger and larger sinks, and this creates installation problems, especially for notebook computers. Therefore, heat pipes are being widely used for the thermal management issue. The capability of dissipating satisfying heat amounts with smaller sizes takes attention for heat pipes.

Today, an Intel® Core™ Duo T2300 1.66 GHz processor dissipates 31 W of energy [20]. The temperature of the processor shall be kept under 125°C as maximum. The normal operating temperature is 85°C. In order to transfer heat that is dissipated from CPU, water – copper heat pipe having 20 cm. length and 1 cm. pipe diameter is used. Wire screen meshes are used as wick structures. The material of the wick is copper. With the usage of two layers of 24 meshes/inch copper wire screen mesh, heat load of 469 W can be transmitted. The values are for horizontal position, where the favorable tilt conditions provide more heat transfer capacity. However, for an adverse tilt position of five degrees, the capacity decreases to 23 W. This capacity is not enough in order to transfer the heat dissipated from CPU. Therefore, it is not recommended to use the notebook computers in adverse tilt positions.

4.2 Cryogenic Application

The thermal management in satellites is one of the main areas where cryogenic heat pipes can be used. They are used for cooling heat dissipating equipments. The heat is transferred from equipments to radiation panels by heat pipes. One of the satellite applications is presented in [19]. Two heat pipes each having total length of 951 mm. is used. The lengths of evaporator, condenser and adiabatic section are 325 mm, 285 mm. and 341 mm. respectively. Open groove wick structure with 0.5 mm. width and 1 mm. depth is used. The wick consists of 30 grooves. Ammonia is used as working fluid. The expected heat transfer capacity of the heat pipes are 90 W for the temperature range of -50°C (P_{sat}=40.5 kPa) to 60°C(P_{sat}=2.6MPa). The given specifications are applied to computer program. As a result, it is seen that the governing limit is the capillary limit for the specified case. For the temperatures up to 0°C(P_{sat}=427 kPa) heat pipes can provide more than 90 W of heat transfer.

However, as the temperature increases, the capacity decreases and at the temperature of 60°C, heat pipes can provide 15 W of heat capacity. The variation of heat transfer capacities with respect to temperature are shown in Figure 3.41.

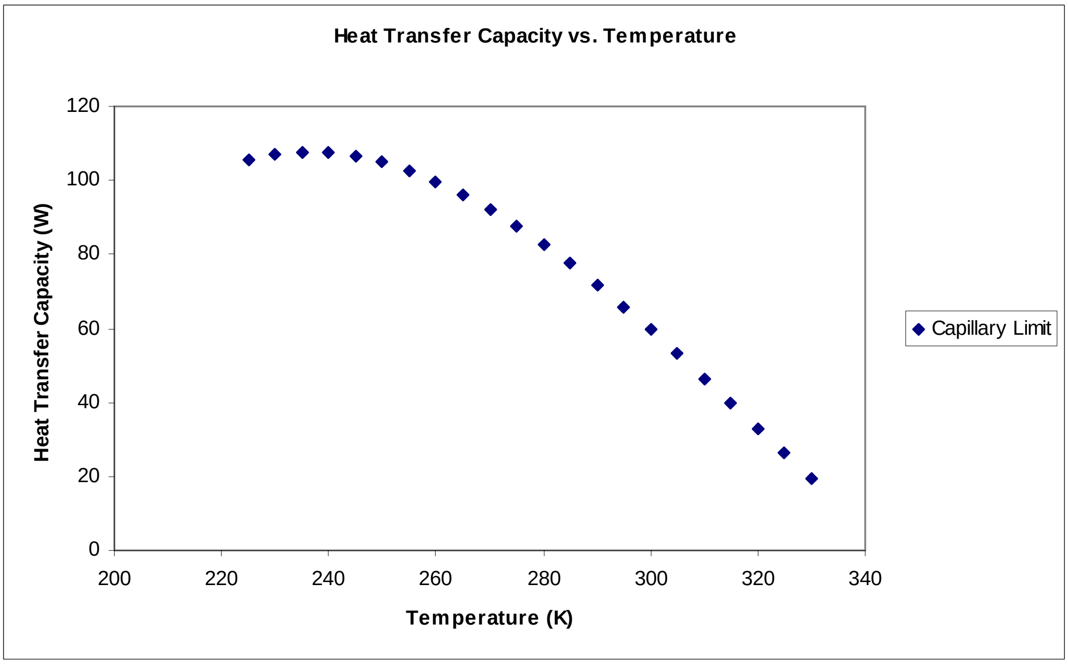


Figure 3.41 - Results of the cryogenic application of Ammonia heat pipe

Therefore, although the expected capacity was mentioned as 90 W in [20], this can not be provided for the whole temperature range. The maximum heat transfer capacity can be achieved as 107.5 W at the temperature of -33°C (Psat=102 kPa).

4.3 High Temperature Application

The heat pipes can be used for high temperature applications like cooling of molds, tools of machining equipments like drills or ovens etc. Liquid metals like Mercury, Sodium, Lithium etc. can be used as working fluids for high temperature heat pipe applications.

CHAPTER 5

CONCLUSION

In this study, the governing limits for the heat transfer capacity of the heat pipe are investigated. Today, heat pipes are widely used in electronic cooling applications. Despite the fact that they have the capability of transferring large rates of heat with small temperature differences, their operations are limited by some factors. There are five limits that govern the capacity of a heat pipe; capillary limit, viscous limit, entrainment limit, sonic limit and boiling limit. There are various parameters that affect the heat transfer capacity of the heat pipe governed by these limits. These parameters are working fluid properties, wick structure properties, geometry of the heat pipe etc. Throughout the study, the effects of those parameters on the performance of the heat pipe and on the limits are investigated. By keeping all the parameters constant, except the investigated one, individual effects are examined.

In order to see the working fluid effects, three different working fluids are used in the analysis: Water, Ammonia and Mercury. Two commonly used wick structures are selected to demonstrate the wick structure effect on the heat transfer capacity of the heat pipe. These are wire screen mesh wick and groove wick. Moreover, effects of the length and the diameter of the pipe are also investigated in the study. The analyses are performed at different tilt angles, in order to see the effect of orientation on the heat pipe performance.

The heat pipe used in the analyses has 0.75 m. total length, 0.020 m. pipe diameter with 0.0025 m. wall thickness. The orientation is set as horizontal initially.

The first analyses are done using Water as working fluid. In these analyses, the effects of different properties of wire screen mesh wick structure like mesh number, layer number are investigated. For open grooves, the effects of groove width, groove depth and groove number are examined. As the baseline for calculations, a 100 mesh wire screen wick is used. For Water heat pipe, it is seen that as the temperature increases, the capacities for all limits increases except boiling limit. The capacity of the boiling limit decreases as temperature increases. These are valid for both of the wick structures.

The wick structure with two layers of 24 meshes/inch wick provides 380 W of heat transfer capacity. The open grooves with 30 grooves provide 45 W of heat with the same diameter and length of the pipe. Therefore, it is preferable for water to use wire screen meshes with two layers of 24 meshes/inch.

The analyses are also performed for Ammonia. As the temperature increases, all the limits except boiling limit for ammonia heat pipe increases. However, the values of boiling limit decreases as the temperature becomes higher and higher. It is the capillary and boiling limits that govern the ammonia heat pipe for the temperature range of 196 K – 395 K, for both of the wick structures.

The ammonia heat pipe gives its maximum capacity at 245 K when two layers of 24 meshes/inch wire screen is used as wick structure. The maximum heat transfer rate is 190 W. For open grooves of 30 grooves number, the ammonia can provide only 20.6 W heat transfer. According to these results, it is better to use wire screen meshes for Ammonia.

The analyses of Mercury show that the temperature increase causes decrease in boiling limit and increase in all remaining limits. The capillary and entrainment limits are the governing limits for mercury, whether the wire screen or open grooves are used as wick structures, in the temperature range of 423 K -1023 K.

The mercury heat pipe gives its maximum heat transfer capacity when 30 meshes/inch of two layers wire screen mesh is used for the operating temperature of 673 K. It provides 544 W of heat transfer. The open grooves with 30 groove number provide 437 W heat transfer. These results show that in order to get maximum capacity from Mercury heat pipe, wire screen meshes shall be preferred.

As a conclusion, it is seen that using wire screen meshes as wick structure provides maximum heat transfer capacities for all three heat pipe working fluids. Water provides the maximum heat transfer capacity for the temperature range of 280 K – 373 K temperature range. Therefore, water is better than Ammonia and Mercury for room temperature applications. Ammonia shows better performance for cryogenic heat pipe applications, and Mercury can be used for high temperature heat pipe applications.

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APPENDIX

1. The Fluid Thermophysical Properties

a) Water (Incropera, F.P, DeWitt, D.P., 2002, *Fundementals of Heat and Mass Transfer,* John Wiley & Sons.):

Temp. (K)	Latent Heat of Vap. (kj/kg)	Liq. Density (kg/m^3)	Vapor Density (kg/m^3)	Liq. Visc. x 10^3 (Ns/m^2)	Vap. Visc. x 10^{-6} (Ns/m^2)	Vapor Pressure (Pa)	Surface Tension (N/m)	Liquid Conduct. (W/mK)
273	2502	1000.0	0.00485	1.750	8.02	611	0.0755	0.569
275	2497	1000.0	0.00550	1.652	8.09	697	0.0753	0.574
280	2485	1000.0	0.00767	1.422	8.29	990	0.0748	0.582
285	2473	1000.0	0.01006	1.225	8.49	1387	0.0743	0.590
290	2461	999.0	0.01435	1.080	8.69	1917	0.0737	0.598
295	2449	998.0	0.01925	0.959	8.89	2617	0.0727	0.606
300	2438	997.0	0.02556	0.855	9.09	3531	0.0717	0.613
305	2426	995.0	0.03362	0.769	9.29	4712	0.0709	0.620
310	2414	993.0	0.04361	0.695	9.49	6221	0.0700	0.628
315	2402	991.1	0.05612	0.631	9.69	8132	0.0692	0.634
320	2390	989.1	0.07153	0.577	9.89	10530	0.0683	0.640
325	2378	987.2	0.09042	0.528	10.09	13510	0.0675	0.645
330	2366	984.3	0.11338	0.489	10.29	17190	0.0666	0.650
335	2354	982.3	0.14104	0.453	10.49	21670	0.0658	0.656
340	2342	979.4	0.17422	0.420	10.69	27130	0.0649	0.660
345	2329	976.6	0.21354	0.389	10.89	33720	0.0641	0.668
350	2317	973.7	0.26001	0.365	11.09	41630	0.0632	0.668
355	2304	970.9	0.31447	0.343	11.29	51000	0.0623	0.671
360	2291	967.1	0.37807	0.324	11.49	62090	0.0614	0.674
365	2278	963.4	0.45208	0.306	11.69	75140	0.0605	0.677
370	2265	960.6	0.53735	0.289	11.89	90400	0.0595	0.679
373	2257	957.9	0.59559	0.279	12.02	101330	0.0589	0.680

b) Ammonia (http://webbook.nist.gov/chemistry/fluid/)

Temp. (K)	Latent Heat of Vap. (kj/kg)	Liq. Density (kg/m^3)	Vapor Density (kg/m^3)	Liq. Viscosity x 10^3 (Ns/m^2)	Vapor Viscosity x 10^{-4} ($Ns/m^{2)}$)	Vapor Pressure (kPa)	Surface Tension (N/m)	Liquid Conductivity (W/mK)
196	1463.99	732.37	0.07	0.5533	0.0685	6.3	0.0620	0.817
200	1473.90	728.12	0.09	0.5073	0.0695	8.7	0.0604	0.803
205	1461.89	722.7	0.13	0.4575	0.0708	12.5	0.0583	0.786
210	1449.64	717.16	0.17	0.4150	0.0721	17.7	0.0563	0.768
215	1437.13	711.52	0.24	0.3784	0.0735	24.7	0.0543	0.751
220	1424.19	705.76	0.32	0.3467	0.0748	33.8	0.0523	0.733
225	1410.99	699.91	0.42	0.3191	0.0763	45.5	0.0504	0.716
230	1397.36	693.95	0.55	0.2949	0.0777	60.4	0.0485	0.699
235	1383.39	687.9	0.71	0.2737	0.0791	79.1	0.0466	0.682
240	1369.00	681.75	0.90	0.2549	0.0806	102.2	0.0447	0.665
245	1354.17	675.51	1.13	0.2381	0.0821	130.6	0.0428	0.649
250	1338.91	669.18	1.40	0.2231	0.0836	164.9	0.0410	0.632
255	1323.22	662.75	1.73	0.2096	0.0851	206.2	0.0392	0.616
260	1307.11	656.22	2.12	0.1973	0.0866	255.3	0.0374	0.600
265	1290.47	649.59	2.57	0.1862	0.0881	313.3	0.0357	0.584
270	1273.29	642.87	3.09	0.1761	0.0896	381.1	0.0340	0.569
275	1255.58	636.03	3.69	0.1667	0.0911	459.9	0.0323	0.554
280	1237.31	629.08	4.38	0.1581	0.0927	550.9	0.0306	0.539
285	1218.50	622.01	5.17	0.1502	0.0942	655.3	0.0290	0.524
290	1199.01	614.81	6.07	0.1427	0.0958	774.4	0.0274	0.509
295	1178.85	607.47	7.10	0.1358	0.0973	909.4	0.0258	0.495
300	1157.99	599.97	8.25	0.1293	0.0989	1061.7	0.0242	0.480
305	1136.32	592.32	9.55	0.1232	0.1006	1232.8	0.0227	0.466
310	1113.81	584.48	11.02	0.1175	0.1022	1424	0.0212	0.452
315	1090.55	576.44	12.67	0.1121	0.1039	1636.8	0.0198	0.438
320	1066.30	568.19	14.51	0.1069	0.1056	1872.8	0.0183	0.425
325	1040.94	559.69	16.58	0.1020	0.1074	2133.5	0.0169	0.411
330	1014.52	550.92	18.89	0.0973	0.1093	2420.5	0.0155	0.398
335	986.90	541.85	21.49	0.0928	0.1112	2735.5	0.0142	0.385
340	957.93	532.44	24.40	0.0886	0.1133	3080.2	0.0129	0.372
345	927.46	522.63	27.66	0.0844	0.1155	3456.5	0.0116	0.358
350	895.20	512.38	31.33	0.0804	0.1179	3866	0.0104	0.345
355	861.07	501.62	35.48	0.0766	0.1206	4310.8	0.0092	0.332
360	824.77	490.26	40.19	0.0728	0.1235	4792.9	0.0080	0.319
365	785.85	478.2	45.56	0.0691	0.1268	5314.4	0.0069	0.306
370	743.94	465.29	51.73	0.0655	0.1305	5877.8	0.0058	0.293
375	698.30	451.34	58.90	0.0619	0.1350	6485.4	0.0048	0.280
380	647.98	436.07	67.37	0.0583	0.1403	7140.2	0.0038	0.267
385	591.75	419.08	77.56	0.0547	0.1468	7845.3	0.0029	0.253
390	527.20	399.62	90.22	0.0509	0.1553	8604.5	0.0020	0.240
395	449.60	376.2	106.81	0.0468	0.1670	9422.3	0.0012	0.226

c) Mercury (Peterson, G.P., 1994, *An Introduction to Heat Pipes;Modeling, Testing and Application",* John Wiley & Sons.

Temp. (K)	Latent Heat of Vap. (kj/kg)	Liq. Density (kg/m^3)	Vapor Density (kg/m^3)	Liq. Viscosity x 10^3 (Ns/m^2)	Vapor Viscosity x 10^{-4} (Ns/m^2)	Vapor Pressure (Pa)	Surface Tension (N/m)	Liquid Conductivity (W/mK)
423	308.8	13230	0.010	0.0011	3.9	1000	0.045	9.99
473	306.3	13112	0.305	0.0010	4.3	9500	0.043	10.61
523	303.8	12995	0.600	0.0010	4.8	18000	0.042	11.23
573	301.8	12880	1.73	0.0009	5.3	44000	0.040	11.73
623	298.9	12763	4.45	0.0009	6.1	116000	0.038	12.18
673	296.3	12656	8.75	0.0009	6.6	242000	0.037	12.58
723	293.8	12508	16.8	0.0008	7.0	492000	0.036	12.96
773	291.3	12308	28.6	0.0008	7.5	886000	0.034	13.31
823	288.8	12154	44.92	0.0008	8.1	1503000	0.033	13.62
873	286.3	12054	65.75	0.0008	8.7	2377000	0.032	13.87
923	283.5	11962	94.39	0.0008	9.5	3495000	0.030	14.15
973	280.3	11881	132	0.0008	10.2	4897500	0.029	14.47
1023	277	11800	170	0.0008	11	6300000	0.028	14.8

2. The Matlab Code

a) Main routine;

```
functionlimits(mercury1,mercury2,mercury3,mercury4,mercury5,mercury
6,mercury7,mercury8,ammonia1,ammonia2,ammonia3,ammonia4,ammonia5,am
monia6,ammonia7,ammonia8,water1,water2,water3,water4,water5,water6,
water7,water8,wicktype,T,waterprop,fluid,ammoniaprop,mercuryprop,N,
d,dv,fi,La,Levap,Lcon,dw,rce,keff,t,rn,kw,Lay,gama,Rvap,K,por,Ng,w,
dep,fReg,rh_g)

global lambda
global roliq
global rovap
global nuliq
global nuv
global Pvap
global sigma
global kl
global T
global Rvap
global gama
global capillarywater
global entrainmentwater
global boilwater
global viscouswater
global sonicwater
```

```
global capillaryammonia
global entrainmentammonia
global boilammonia
global viscousammonia
global sonicammonia
global capillarymercury
global entrainmentmercury
global boilmercury
global viscousmercury
global sonicmercury
global limit

if fluid==1
        lambda=water1(T)*1000;
        roliq=water2(T);
        rovap=water3(T)
        nuliq=water4(T);
        nuv=water5(T)/1000000;
        Pvap=water6(T);
        sigma=water7(T)
        kl=water8(T);
        if wicktype==4
                wirescreen(Lay,N,dw,t,d,kl,kw);
                sonicwater=sonic_limit(fluid,dv,T,lambda,rovap);

capillarywater=capillary_limit(rce,K,sigma,N,roliq,d,dv,fi,La,Levap
,Lcon,nuv,rovap,lambda,nuliq,dw,gama,Rvap,T,t);

entrainmentwater=entrainment_limit(d,dv,sigma,rovap,lambda,t);

boilwater=boiling_limit(N,dv,Levap,keff,T,d,t,lambda,rovap,sigma,rn
);

viscouswater=viscous_limit(dv,lambda,Pvap,nuliq,Levap,rovap);

        else if wicktype==2
                grooves(w,dep,Ng,kl,kw,t,d,fReg);

sonicwater=sonic_limitgrooves(fluid,dv,T,rovap,lambda);

capillarywater=capillary_limitgrooves(rce,K,sigma,Ng,roliq,d,dv,fi,
La,Levap,Lcon,nuv,rovap,lambda,nuliq,dw,gama,Rvap,T,t,w,dep,rh_g);

entrainmentwater=entrainment_limitgrooves(d,dv,sigma,rovap,lambda,t
,rh_g);

boilwater=boiling_limitgrooves(Ng,dv,Levap,keff,T,d,t,lambda,rovap,
sigma,rn,w,rce,dep);

viscouswater=viscous_limitgrooves(dv,lambda,Pvap,nuliq,Levap,rovap)
;
         end;
         end;
    q=capillarywater;
    limit='capillary';
    if q>entrainmentwater
        q=entrainmentwater;
```

```
            limit='entrainment';
        end;
        if q>viscouswater
            q=viscouswater;
            limit='viscous';
        end;
        if q>sonicwater
            q=sonicwater;
            limit='sonic';
        end;
          if q>boilwater
            q=boilwater;
            limit='boiling';
        end;
        limit
        q

    else if fluid==2
            for i=1:200
                lambda=ammonia1(T)*1000;
                roliq=ammonia2(T);
                rovap=ammonia3(T);
                rovap=ammoniaprop(i,4);
                nuliq=ammonia4(T);
                nuv=ammonia5(T)/10000;
                Pvap=ammonia6(T)*1000000;
                sigma=ammonia7(T);
                kl=ammonia8(T);
                if wicktype==4
                       wirescreen(Lay,N,dw,t,d,kl,kw);

sonicammonia=sonic_limit(fluid,dv,T,lambda,rovap);

capillaryammonia=capillary_limit(rce,K,sigma,N,roliq,d,dv,fi,La,Lev
ap,Lcon,nuv,rovap,lambda,nuliq,dw,gama,Rvap,T,t);

entrainmentammonia=entrainment_limit(d,dv,sigma,rovap,lambda,t);

boilammonia(i,1)=boiling_limit(N,dv,Levap,keff,T,d,t,lambda,rovap,s
igma,rn);

viscousammonia=viscous_limit(dv,lambda,Pvap,nuliq,Levap,rovap);
                 else if wicktype==2
                       grooves(w,dep,Ng,kl,kw,t,d,fReg);

sonicammonia=sonic_limitgrooves(fluid,dv,T,rovap,lambda);

capillaryammonia=capillary_limitgrooves(rce,K,sigma,Ng,roliq,d,dv,f
i,La,Levap,Lcon,nuv,rovap,lambda,nuliq,dw,gama,Rvap,T,t,w,dep,rh_g)
;

entrainmentammonia=entrainment_limitgrooves(d,dv,sigma,rovap,lambda
,t,rh_g);

boilammonia=boiling_limitgrooves(Ng,dv,Levap,keff,T,d,t,lambda,rova
p,sigma,rn,w,rce,dep);
```

```
viscousammonia=viscous_limitgrooves(dv,lambda,Pvap,nuliq,Levap,rova
p);
             end;
             end;
          end;
        q=capillaryammonia;
        limit='capillary';
        if q>entrainmentammonia
            q=entrainmentwammonia;
            limit='entrainment';
        end;
        if q>viscousammonia
            q=viscousammonia;
            limit='viscous';
        end;
        if q>sonicammonia
            q=sonicammonia;
            limit='sonic';
        end;
        if q>boilammonia
            q=boilammonia;
            limit='boiling';
        end;
        limit
        q

else if fluid==3;
            lambda=mercury1(T)*1000;
            roliq=mercury2(T);
            rovap=mercury3(T);
            nuliq=mercury4(T);
            nuv=mercury5(T)/1000000;
            Pvap=mercury6(T);
            sigma=mercury7(T);
            kl=mercury8(T);
            if wicktype==4
                wirescreen(Lay,N,dw,t,d,kl,kw);
                sonicmercury=sonic_limit(fluid,dv,T,lambda,rovap);

capillarymercury=capillary_limit(rce,K,sigma,N,roliq,d,dv,fi,La,Lev
ap,Lcon,nuv,rovap,lambda,nuliq,dw,gama,Rvap,T,t);

entrainmentmercury=entrainment_limit(d,dv,sigma,rovap,lambda,t);

boilmercury=boiling_limit(N,dv,Levap,keff,T,d,t,lambda,rovap,sigma,
rn);

viscousmercury=viscous_limit(dv,lambda,Pvap,nuliq,Levap,rovap);
            else if wicktype==2
                      grooves(w,dep,Ng,kl,kw,t,d,fReg);

sonicmercury=sonic_limitgrooves(fluid,dv,T,rovap,lambda);

capillarymercury=capillary_limitgrooves(rce,K,sigma,Ng,roliq,d,dv,f
i,La,Levap,Lcon,nuv,rovap,lambda,nuliq,dw,gama,Rvap,T,t,w,dep,rh_g)
;
```

```
entrainmentmercury=entrainment_limitgrooves(d,dv,sigma,rovap,lambda
,t,rh_g);

boilmercury=boiling_limitgrooves(Ng,dv,Levap,keff,T,d,t,lambda,rova
p,sigma,rn,w,rce,dep);

viscousmercury=viscous_limitgrooves(dv,lambda,Pvap,nuliq,Levap,rova
p);
            end;
            end;
        q=capillarymercury;
        limit='capillary';
        if q>entrainmentmercury
            q=entrainmentmercury;
            limit='entrainment';
        end;
        if q>viscouswater
            q=viscousmercury;
            limit='viscous';
        end;
        if q>sonicmercury
            q=sonicmercury;
            limit='sonic';
        end;
        if q>boilmercury
            q=boilmercury;
            limit='boiling';
        end;
        limit
        q
        end;
    end;
end;
```

b) Wire Screen Mesh Properties Sub-Routine;

```
function wirescreen(Lay,N,dw,t,d,kl,kw)

global por;
global K;
global dv;
global keff;
global rce;

NN=N/0.0254;
rce=1/(2*NN);
dv=(d-2*t)-2*(2*Lay*dw);
por=1-(1.05*3.14*NN*dw/4);
K=dw*dw*por*por*por/(122*(1-por)*(1-por));
keff=kl*((kl+kw)-((1-por)*(kl-kw)))/((kl+kw)+((1-por)*(kl-kw)));
```

c) Open Grooves Sub-Routine

```
function grooves(w,dep,Ng,kl,kw,t,d,fReg)
global por
global K
global keff
global rce
global rh_g
global dv
dv=d-2*t;
wf=(3.14*dv/Ng)-w;
if wf<0
    q=0
end;
rce=w;
rh_g=(2*w*dep)/(w+2*dep);
por=w/(w+wf);
a=w/dep;
K=(2*por*rh_g*rh_g)/(fReg(w/dep));
keff=((wf*kl*kw*dep)+w*kl*(0.185*wf*kw+dep*kl))/((w+wf)*(0.185*wf*k
w+dep*kl));
```

d) Sub-Routines of Limits for Wire Screen Mesh Wick Structure:

```
function[qb]=boiling_limit(N,dv,Levap,keff,T,d,t,lambda,rovap,sigma
,rn);

NN=N/0.0254;
a=2*3.14*Levap*keff*T;
rvap=dv/2;
Pcap=2*sigma*(2*NN);
rin=d/2-t;
log(rin/rvap);
b=lambda*rovap*log(rin/rvap);
c=((2*sigma/rn)-Pcap);
qb=(a/b)*c;
if qb<0
   qb=0;
end


function[q]
=capillary_limit(rce,K,sigma,N,roliq,d,dv,fi,La,Levap,Lcon,nuv,rova
p,lambda,nuliq,dw,gama,Rvap,T,t)
L=La+(Levap+Lcon)/2;

if (Levap==0)&&(Lcon==0)
    L=La;
end;
```

```
C=1;
fRev=16;
fi=fi*pi/180;
g=9.81;
rhv=dv/2;
Av=3.14*dv*dv/4;
Awick=3.14*((d-2*t)*(d-2*t)-dv*dv)/4;
a=(2*sigma/rce)-roliq*g*(dv*cos(fi)+L*sin(fi));
b=(C*fRev*nuv)/(2*rhv*rhv*Av*rovap*lambda);
x=nuliq/(K*Awick*lambda*roliq);
q=a/((b+x)*L);

Re=(2*rhv*q)/(Av*nuv*lambda);
Mach=(q*0.5)/(Av*rovap*lambda*Rvap*T*gama);

if Re<=2300 && Mach<=0.2
    fRev=16;
    C=1;
else if Re<=2300 && Mach>0.2
        fRev=16;
        C=(1+((gama-1)/2)*Mach*Mach)^0.5;
    else if Re>2300 && Mach<=0.2
            fRev=0.038;
            C=((2*rhv*q)/(Av*lambda*nuv))^0.75;
        else if Re>2300 && Mach>0.2
                fRev=0.038;
                C=((1+((gama-
1)/2)*Mach*Mach)^0.5)*(((2*rhv*q)/(Av*lambda*nuv))^0.75);
            end
        end
    end
end

fRev
C

b=(C*fRev*nuv)/(2*rhv*rhv*Av*rovap*lambda);
q=a/((b+x)*L);
if q<0
    q=0;
end


function[q] =entrainment_limit(d,dv,sigma,rovap,lambda,t);

di=d-2*t;
Av=3.14*dv*dv/4;
Aw=3.14*(di*di-dv*dv)/4;
rhw=Aw/(3.14*(dv+(d-2*t)));
a=((sigma*rovap)/(2*rhw)).^0.5;
q=Av*lambda*a;
if q<0
    q=0;
end
```

```
function[q] =sonic_limit(fluid,dv,T,rovap,lambda);
global gama;
global M;
global Rvap;

if fluid==1;
    gama=1.33;
    M=18;
else if fluid==2;
        gama=1.33;
        M=17;
    else if fluid==3;
            gama=1.67;
            M=200;
        end;
    end;
end;

Av=3.14*dv*dv/4;
Runi=8.314e3;
Rvap=Runi/M;
a=((gama*Rvap*T)/(2*(gama+1))).^0.5;
q=Av*rovap*lambda*a;
if q<0
    q=0;
end


function[q]=viscous_limit(dv,lambda,Pvap,nuliq,Levap,rovap);
Av=3.14*dv*dv/4;
a=Av*dv*dv*lambda*Pvap*rovap;
b=64*nuliq*Levap;
q=a/b;
if q<0
    q=0;
end
```

e) Sub-Routines of Limits for Open Groove Wick Structure:

```
function[qb]
=boiling_limitgrooves(Ng,dv,Levap,keff,T,d,t,lambda,rovap,sigma,rn,
w,rce,dep);
a=2*3.14*Levap*keff*T;
rvap=dv/2;
Pcap=2*sigma/rce;
rin=d/2-t+dep;
rvap;
rin/rvap;
log(rin/rvap);
b=lambda*rovap*log(rin/rvap);
c=((2*sigma/rn)-Pcap);
qb=(a/b)*c;
if qb<0
   qb=0;
end
```

```
function[q]
=capillary_limitgrooves(rce,K,sigma,Ng,roliq,d,dv,fi,La,Levap,Lcon,
nuv,rovap,lambda,nuliq,gama,Rvap,T,t,w,dep,rh_g)
L=La+(Levap+Lcon)/2;

if (Levap==0)&&(Lcon==0)
    L=La;
end;

C=1;
fRev=16;
fi=fi*pi/180;
g=9.81;
Av=3.14*dv*dv/4;
rhv=dv/2;
Agrooves=w*dep*Ng;%%area os grooves
a=(2*sigma/rce)-roliq*g*(dv*cos(fi)+L*sin(fi));
b=(C*fRev*nuv)/(2*rhv*rhv*Av*rovap*lambda);
x=nuliq/(K*Agrooves*lambda*roliq);
q=a/((b+x)*L);


Re=(2*rhv*q)/(Av*nuv*lambda);
Mach=(q*0.5)/(Av*rovap*lambda*Rvap*T*gama);

if Re<=2300 && Mach<=0.2
    fRev=16;
    C=1;
else if Re<=2300 && Mach>0.2
        fRev=16;
        C=(1+((gama-1)/2)*Mach*Mach)^0.5;
    else if Re>2300 && Mach<=0.2
            fRev=0.038;
            C=((2*rhv*q)/(Av*lambda*nuv))^0.75;
        else if Re>2300 && Mach>0.2
                fRev=0.038;
                C=((1+((gama-
1)/2)*Mach*Mach)^0.5)*(((2*rhv*q)/(Av*lambda*nuv))^0.75);
            end
        end
    end
end

b=(C*fRev*nuv)/(2*rhv*rhv*Av*rovap*lambda);
q=a/((b+x)*L);

if q<0
    q=0;
end
```

```
function[q]
=entrainment_limitgrooves(d,dv,sigma,rovap,lambda,t,rh_g);
Av=3.14*dv*dv/4;
a=((sigma*rovap)/(2*rh_g)).^0.5;
q=Av*lambda*a;
if q<0
    q=0;
end


function[q] =sonic_limitgrooves(fluid,dv,T,rovap,lambda);
global gama;
global M;
global Rvap;

if fluid==1;
    gama=1.33;
    M=18;
else if fluid==2;
        gama=1.33;
        M=17;
    else if fluid==3;
            gama=1.67;
            M=200;
        end;
    end;
end;

Av=3.14*dv*dv/4;
Runi=8.314e3;
Rvap=Runi/M;
a=((gama*Rvap*T)/(2*(gama+1))).^0.5;
q=Av*rovap*lambda*a;
if q<0
    q=0;
end


function[q]=viscous_limitgrooves(dv,lambda,Pvap,nuliq,Levap,rovap);
Av=3.14*dv*dv/4;
a=Av*dv*dv*lambda*Pvap*rovap;
b=64*nuliq*Levap;
q=a/b;
if q<0
    q=0;
end
```

3. Interpolation of the Properties

The thermophysical properties of the fluids are calculated in the program for the temperatures where the properties are not defined. Using the defined properties, a curve fit function is applied for all thermophysical properties of all three fluids. The curves in the degrees of 6 are fitted to the functions of properties. Interpolation is done using these fitted curves and the properties are calculated for the temperatures where the properties are not defined.