Thermodynamic Analysis of Vapour Compression Refrigeration System Using a Heat Pipe

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In partial fulfilment of the requirement for the degree of

Master of Technology

In

Thermal Engineering

Submitted by

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CERTIFICATE

This is to certify that the M. Tech. dissertation entitled **“Thermodynamic Analysis of Vapour Compression Refrigeration System Using a Heat Pipe”** submitted by Ankit Dwivedi (2K13/THE/06) in partial fulfilment of the requirement for the award of the degree of Master of Technology, Delhi Technological University (Formerly Delhi College of Engineering)in Thermal Engineering,under my guidance and supervision.

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DECLARATION

I, Ankit Dwivedi, hereby declare that the work entitled **"Thermodynamic Analysis of Vapour Compression Refrigeration System Using a Heat Pipe"** has been carried out by me under the guidance of Prof. (Dr.)R.S. Mishra, at Delhi Technological University, Delhi.This dissertation is part of partial fulfilment of requirement for the degree of M.Tech in Thermal Engineering. This is the original work and has not been submitted for any other degree in any other university.

Ankit Dwivedi

2k13/THE/06

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ABSTRACT

Heat pipe was developed by NASA for space applications, and this device can be used in simple refrigeration systems, which will have a great impact on the size of the system. Several working fluids can be used in the heat pipe along with the different available material in the designs of the pipe. In this research work several parameters of a heat pipe are observed varying the design conditions alongside the working fluid in the heat pipe which are Water, Ammonia and Ethanol analysed on software. The design conditions are kept as it would be in a commonly used refrigeration system. The capacity of the system is varied and analysis is done. The capacity can be enhanced without much change in the size. Also the temperature of working becomes more or less constant. Water turns out to be a very effective working fluid in a heat pipe with the view of refrigeration when the temperature of working increases. Range of Figure of merits for water is 25000(kW/m^2) where as HTF is of the range of 45(W-m).

CHAPTER 1

INTRODUCTION

1.1 History and Definition of refrigeration

Refrigeration may be characterized as the procedure of accomplishing and keeping up a temperature below ambience state, the point being to cool an item or a space to the required temperature. A stand out amongst the most essential uses of refrigeration’s been the conservation of varieties of perishable sustenance items by keeping them at low temperatures. The refrigeration systems are likewise utilized broadly for giving comfort to people by being one of the methods for aerating and cooling. Cooling alludes to the treatment of air in order to at the same time control its temperature, purity, odour, air movement, relative humidity and course, as needed by tenants, a procedure, or items in the space. The subject of refrigeration and ventilating has advanced out of human requirement for sustenance and solace, and its history goes back to hundreds of years.

In early times refrigeration was accomplished by simple means, ie. the through ice or evaporative cooling. In early days, ice was either:

- Transported from hilly areas,
- Gathered in the winters and put away in ice houses for being used summer or,
- Made amid the nights by the cooling of water by radiation to stratosphere.

Refrigeration as it is known nowadays is developed by counterfeit means. In spite of the fact that it is exceptionally hard to make an unmistakable outline in the middle of natural and artificial refrigeration, it is for the most part concurred that the historical backdrop of counterfeit refrigeration started in 1755, when the Scottish pioneer William Cullen made the first refrigerating machine, which could generate a little amount of ice in his research centre. Taking into account the working rule, refrigeration frameworks can be delegated vapour compression machines, vapour absorption machines, gas cycle machines and so on.

Refrigerant may be a substance that goes about as a cooling medium by removing heat from another body or substance. Under this generalised definition, numerous substances may be called as refrigerants, e.g. ice, cool water, icy air and so on. In the close cycle vapour compression and air cycle refrigeration systems the refrigerant is a working liquid that experiences cyclic changes. In a thermoelectric framework the present conveying electrons may be dealt with as a refrigerant.

All the substances utilized as a part of the early phases of refrigeration experienced one issue or other. The issues were mainly the security issues, ie. poisonous quality, combustibility, high working capacities and so on. Accordingly vast scale commercialization of refrigeration systems was obstructed.

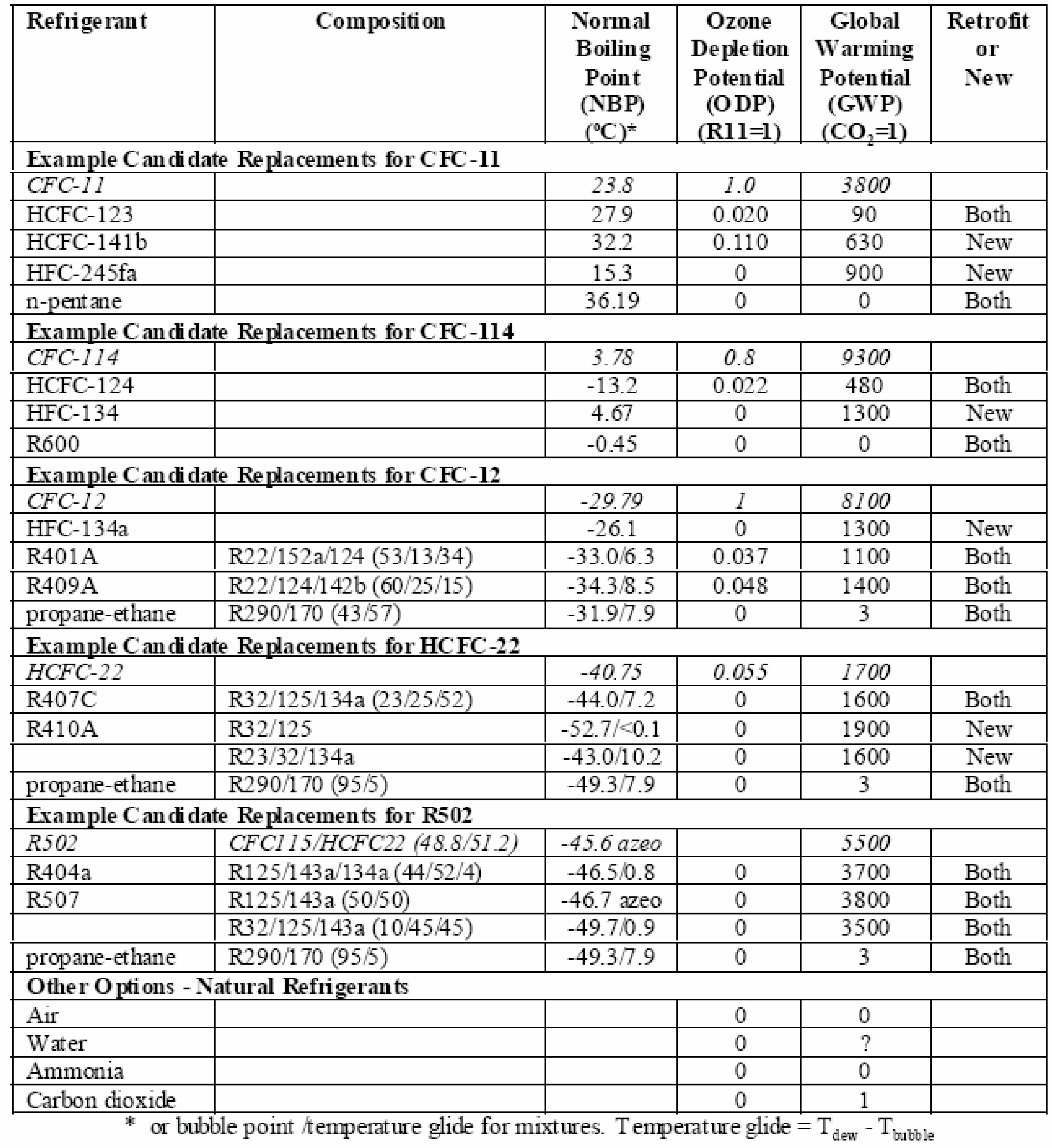
The advent of CFCs and related compounds has upset the field of refrigeration and cooling. The vast majority of the issues connected with right on time refrigerants, for example, harmfulness, combustibility, and material inconsistency were wiped out totally. Likewise, Freons are exceedingly steady compounds. Also, by astutely controlling the structure an entire scope of refrigerants most appropriate for a specific application could be acquired. Just Ammonia among the more established refrigerants survived the Freon enchantment. The chloro-flouro-carbons appreciated complete mastery for around fifty years, until the Ozone Layer Depletion issue was raised via Rowland and Molina in 1974.

All in all the non-ODS manufactured refrigerants, for example, HFC-134a have high a dangerous global warming potential (GWP), subsequently they confront an unverifiable future. Since the a worldwide temperature alteration effect of a refrigerant additionally relies on upon the vitality effectiveness of the framework utilizing the refrigerant (aberrant impact), the proficiency issue has get to be critical in the outline of new refrigeration frameworks. In spite of the fact that the issues of ozone layer exhaustion and an unnatural weather change has prompted a few issues, they have likewise had advantageous impacts of making individuals understand the significance of natural benevolence of advancements. It is normal that with the more noteworthy mindfulness more mindful plans will rise which will eventually advantage the entire humankind.

1.2 Applications:

- Food processing, preservation and distribution
- Chemical and process industries
- Comfort air-conditioning

1.3 Refrigerants:



Refrigerant	Composition	Normal Boiling Point (NBP) (°C)*	Ozone Depletion Potential (ODP) (R11=1)	Global Warming Potential (GWP) (CO_2=1)	Retrofit or New
Example Candidate Replacements for CFC-11					
CFC-11		*23.8*	*1.0*	*3800*	
HCFC-123		27.9	0.020	90	Both
HCFC-141b		32.2	0.110	630	New
HFC-245fa		15.3	0	900	New
n-pentane		36.19	0	0	Both
Example Candidate Replacements for CFC-114					
CFC-114		*3.78*	*0.8*	*9300*	
HCFC-124		-13.2	0.022	480	Both
HFC-134		4.67	0	1300	New
R600		-0.45	0	0	Both
Example Candidate Replacements for CFC-12					
CFC-12		*-29.79*	*1*	*8100*	
HFC-134a		-26.1	0	1300	New
R401A	R22/152a/124 (53/13/34)	-33.0/6.3	0.037	1100	Both
R409A	R22/124/142b (60/25/15)	-34.3/8.5	0.048	1400	Both
propane-ethane	R290/170 (43/57)	-31.9/7.9	0	3	Both
Example Candidate Replacements for HCFC-22					
HCFC-22		*-40.75*	*0.055*	*1700*	
R407C	R32/125/134a (23/25/52)	-44.0/7.2	0	1600	Both
R410A	R32/125	-52.7/<0.1	0	1900	New
	R23/32/134a	-43.0/10.2	0	1600	New
propane-ethane	R290/170 (95/5)	-49.3/7.9	0	3	Both
Example Candidate Replacements for R502					
R502	*CFC115/HCFC22 (48.8/51.2)*	*-45.6 azeo*		*5500*	
R404a	R125/143a/134a (44/52/4)	-46.5/0.8	0	3700	Both
R507	R125/143a (50/50)	-46.7 azeo	0	3800	Both
	R32/125/143a (10/45/45)	-49.7/0.9	0	3500	Both
propane-ethane	R290/170 (95/5)	-49.3/7.9	0	3	Both
Other Options - Natural Refrigerants					
Air			0	0	
Water			0	?	
Ammonia			0	0	
Carbon dioxide			0	1	

* or bubble point /temperature glide for mixtures. Temperature glide = T_{dew} - T_{bubble}

Fig 1.1: List of Refrigerants with their ODP and GWP (ref. NPTEL)

The figure shows different refrigerants with their ozone depletion potential (ODP) and global warming potential (GWP). This table also shows which of the new refrigerants will be able to replace which conventional Freon.

1.4 Heat Pipes:

1.4.1 History

Heat Pipes were created particularly for space applications amid the mid 60' by the NASA. One principle issue in space applications was to transport the temperature from within to the outside, on the grounds that the warmth conduction in a vacuum is exceptionally restricted. Henceforth there was a need to create a quick and viable approach to transport heat, without having the impact of gravity power. The thought behind is to make a flow field which transports heat starting with one spot then onto the next by mode of convection, in light of the fact that convective heat exchange is much quicker than conduction.

These days, heat pipes are utilized in many applications, where one has restricted space and the need of a high heat flux. Obviously, it is still being used in space applications, yet it is likewise utilized as a part of heat exchange frameworks, cooling of PCs, cell telephones and cooling of solar based collectors.

1.4.2 Principle of Working:

The fundamental of heat pipes is dependent of evaporation and condensation. At the hot side, the working liquid is evaporated and at the cool side it condenses. As every material has different properties, it’s required to choose the set of material properly.

At the source the cool fluid is evaporated, the hot vapour stream is a while later transported to the sink where the vapour condenses again and is transported back to the source. The issue of this procedure is the space utilization; consequently it was important to build up a compacter approach to transport heat with the indicated procedure. The thought of a Heat Pipe is presently to incorporate the complete convective transport in one channel, where the vapour stream is in the centre of the pipe and the floe happens to be on the outside of the barrel. Heat exchange is so efficient that fundamentally it is a direct result of the low heat resistance because of the convective stream, as explained earlier. This low heat resistance is because of little effective length of heat exchange through strong porous wick walls.

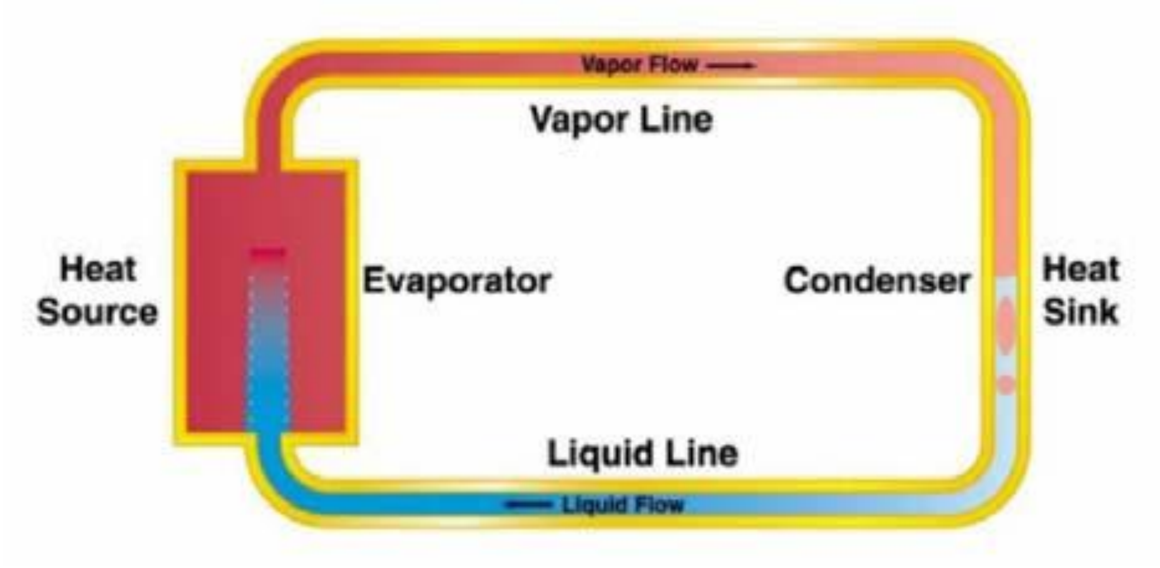


Fig 1.2:Circular process of a heat pipe[6]

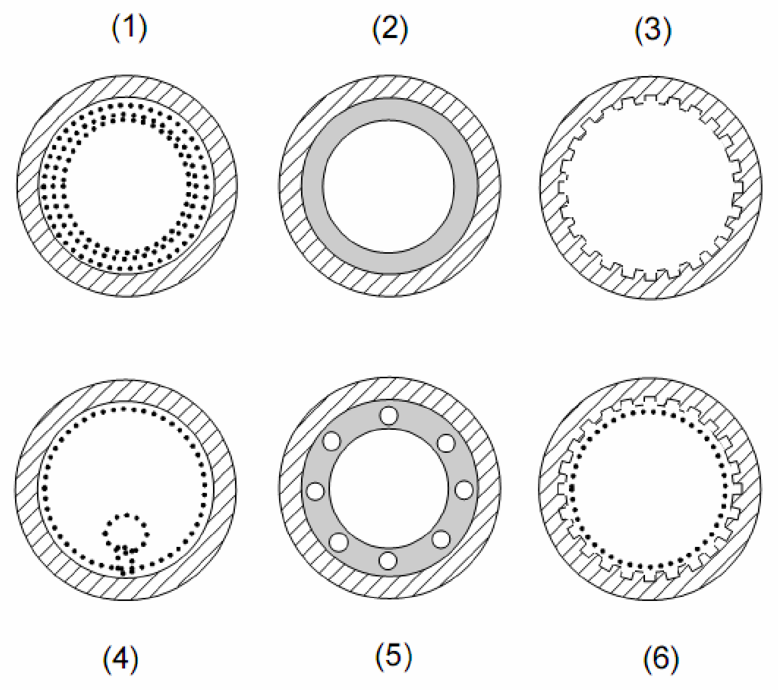


Fig 1.3: Different structures of the capillary layer [6]

Where :

1. Net structure
2. Sinter structure
3. Open channel structure

4-6. Combined structures.

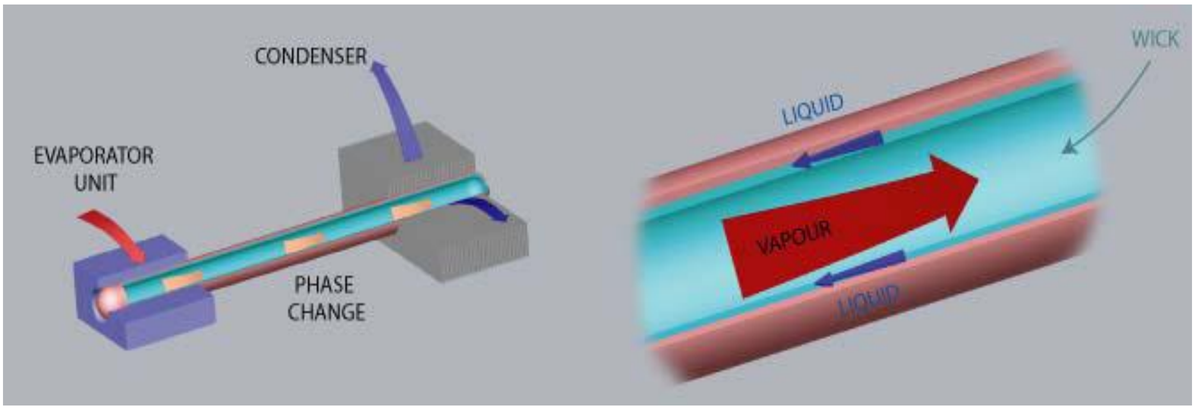


Fig 1.4: Concept of Heat Pipe [6]

Fig 1.5: Definition of the effective length[6]

3.1.3 Key Features of heat pipe

- Very high heat transfer capacity in a simple system.
- Allows several Heat Transport passes, avoids a single point breakdown.
- Smaller units, no bulky pressure vessel.
- Valves, pumps or compressors are not required.
- Can be started cold, no need of preheat.
- High temperatures allow very high efficiency operation, almost constant temperature process.

1.4.4 Limitations

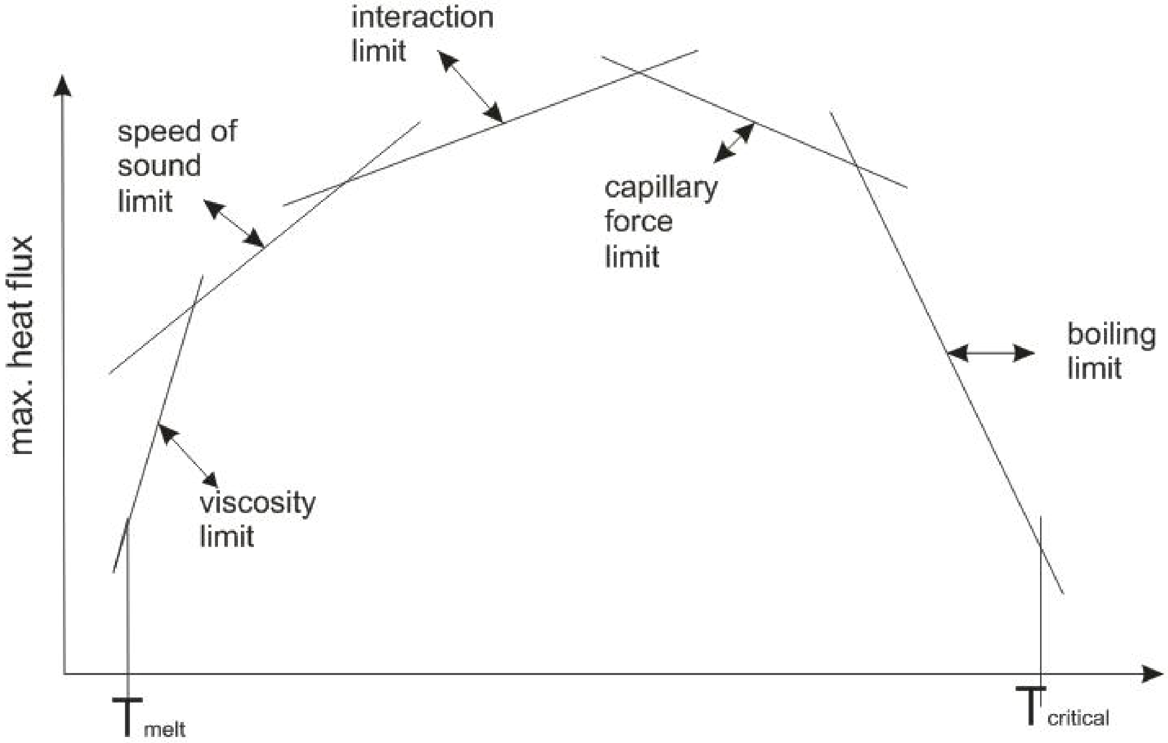


Fig 1.6: Usage limitations of a heat pipe [6]

Melting temperature: One can't utilize a heat pipe below the melting temperature.

Viscosity Limit: The viscosity of the liquid is too high for being transported at low temperatures and low pressures.

Sonic Limit: It is critical for high temperature, where the vapour could conceivably achieve the velocity of sound while leaving the source.

Capillary limit: When the capillary force is definitely not fulfilling the required force transport the fluid

Interaction limit: This breaking point is associated with open channels, where the vapor can be diverted by the vapour, because of high speed contacts.

Boiling cut off: The fluid forms bubbles which close the capillaries, it's not an issue for open channel structures

Critical temperature: Beyond the critical temperature the concept of vapour and liquid seizes to exist.

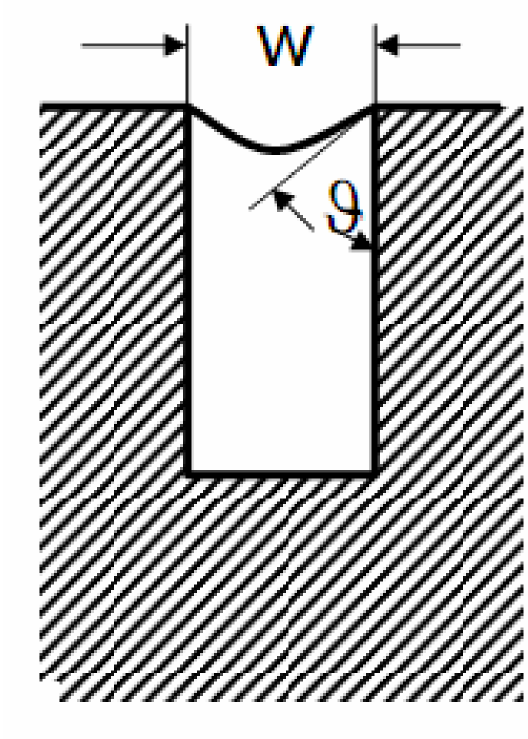


Fig 1.7: Pore Radius [6]

CHAPTER 2

LITERATURE REVIEW

To understand the importance of a heat pipe in the applications of refrigeration the following research papers have been studied and different aspects of research work have been explored.

[1] ***P. Yeunyongkul et al.[2010]*** conducted set of experiments for which the results of closed loop oscillating heat pipe (CLOHP) condenser was connected to vapor compression refrigeration. Two cons of split sort ventilation system were considered. Firstly, the extensive weight drop was happened in the condenser on the grounds that refrigerant streams inside little copper tube. This weight drop causes higher compressor force, bringing about diminishing of the Coefficient of Performance (COP). Also, a great deal of squandered warmth misfortunes to encompassing following the refrigerant needs to consolidate in the wake of going through condenser. To recuperate the drop in weight heat from the gathering process, the exploration went ahead to utilize CLOHP in the ribbon of routine split sort ventilation system. 12,500 Btu/hr of refrigerating limit, while the refrigerant and working liquid was R22 and R123, individually. On contrasting the test information and the routine one it was found that the COP of the last was more prominent than the COP of the CLOHP condenser, yet the vitality proficiency rating (ERR) of the customary condenser was lower than the CLOHP condenser. The drop in the weight of the refrigerant line of the CLOHP condenser was lesser than the traditional one. Finally, an increment of 3 °C in water temperature which assimilates heat from the condenser area of CLOHP was acquired for usage

[2] ***Fabian Korn et al. [2012]*** performed several vital experiments on heat pipes to establish it to be one of the most effective procedures to transport thermal energy from one point to another, mostly used for cooling. It is based on a combination of conduction and convective heat transfer, what makes it to a complex heat transfer problem. In this report the main working principal and most important possibilities to calculate a heat pipe will be shown. All in all it is necessary to understand all the basic theories of heat and mass transfer to understand the working principle of a heat pipe. On a first look a heat pipe seemed to be a very easy tool to transport energy, but if one looks closer, it is a very complex heat and mass transfer process which takes place in a heat pipe. First of all one has convective heat transfer in the adiabatic transport range, and one has convection through porous materials also. The second major point is mass transfer due to vaporization and condensation, also through porous media. Furthermore there are capillary effects, pressure effects and heat conduction effects involved, which creates a complex structure of heat transfer, where a lot of knowledge is involved. And all of these points can be treated as one problem, from this follows that a complete understanding of all involved processes needs more time and space than it is available for this project report.

[3]*Sameer Khandekar et al. [2010]* performed experiments on the global thermal performance modeling of Pulsating Heat Pipes (PHPs) requires local, spati-o-temporally coupled, flow and heat transfer information during the characteristic, self-sustained thermally driven oscillating Taylor bubble flow, under different operating conditions. Local hydrodynamic characteristics such as velocities, lengths, shapes and profiles of bubbles and slugs, their dynamic contact angles, thickness of the liquid film that surrounds the bubbles, enhanced mixing/ flow circulation within the liquid slugs and net pressure drop along the flow, etc., are needed to predict local heat transfer and thus, the global thermal performance. In this paper, we systematically review the experimental, theoretical/analytical, and modeling methodologies to predict these hydrodynamic properties in unidirectional two-phase Taylor bubble flows, in the context of Pulsating Heat Pipes. Indeed, there is little literature available for oscillating Taylor bubbles flows. In view of the state-of-the-art, we therefore recommend some directions and perspectives for furthering research on understanding and modeling PHPs. Understanding the local hydrodynamics of oscillating Taylor bubble flows forms an important element in the building block of the mathematical description of the flow in a PHP. Eventually, the wettability of the liquid-solid combination has profound effects on the overall transport behavior. Needless to say, while the issue of hydrodynamics and resulting thermal behavior are intrinsically linked, in this article we have focused our attention only on reviewing the hydrodynamics of Taylor bubble flows. The fundamental understanding of the wettability and contact line motion are also reviewed. The related heat transfer implications necessarily require another detailed complementary review, which will be addressed in the near future. Their review clearly exhibits the fact that many hydrodynamic aspects of steady unidirectional Taylor bubble flow, such as the thickness of the liquid film that surrounds the bubbles, bubble velocity, bubble and slug lengths, mixing and flow circulation in the liquid slugs, as well as pressure drop during Taylor flow, are quite well understood. On the other hand, oscillating Taylor bubble flows require immediate attention. Unless the nuances of oscillatory confined bubbles in small capillaries is well discerned, the net pressure drop correlations for a PHP cannot be exhaustively constructed. The challenging issues are high inertia, effect of wettability, contribution of surface waves, film thickness dynamics and bubble breakage and merger due to flow instability.

[4]*Jozef Hužvár, Patrik Nemec et al. [2007]* used heat pipe, observed its basic principles and operating limits. High temperature heat pipes were evaluated for use in energy conversion applications such as fuel cells, gas turbine re-combustors, and Stirling cycle heat sources, with the resurgence of space nuclear power, additional applications include reactor heat removal elements and radiator elements. In the temperature range between 500 and 1000 °C, heat pipes can offer the favorable features of passive, reliable operation, effective thermal coupling between non-contacting fluid streams, and modest cost. Long operating life and reliable performance are critical requirements for these applications. Using the analysis techniques for each limitation independently, the heat transport capacity as a function of the mean operating temperature can be determined. This procedure yields a heat pipe performance region that is shown on graph 1 and graph 2. As shown, the separate performance limits define an operational range represented by the region bounded by the combination of the individual limits. In effect, this operational range defines the region or combination of temperatures and maximum transport capacities at which the heat pipe will function. Limitations of the maximum heat input that may be transported by a heat pipe can be divided in two primary categories: limits that result in heat pipe and limits that do not. For the limitations resulting in heat pipe failure, all are characterized by insufficient liquid flow to the evaporator for a given heat input, thus resulting in dry out of the evaporator wick structure. However, limitations not resulting in heat pipe failure that the heat pipe operating at temperature for an increase in heat input.

[5] *R.Z. Wanget al. [2008]* added heat pipes in adsorption water chiller or ice maker initials. His work showed that the adsorption refrigerators are very efficient. The firstly a small scale silica gel–water adsorption water chiller with cooling power rated as 10 kW; the system could be powered by 60–100°C hot water, a cooling COP = 0.4 has been achieved when driven by 85°C hot water. This adsorption chiller has been used for solar powered air conditioner and also as the chiller for CCHP system. Secondly a silica gel–water adsorption room air conditioner powered by 80°C hot water. The system was very compact and they suggested for potential applications of micro CCHP system based on fuel cells. The system has a COP of over 0.3 and cooling power of about 1 kW. The third example is the use of split heat pipes to heat or cool the adsorber for making ice in fishing boats. The application of these technologies avoids the corrosion of adsorber at the heating phase by exhausted gases and at the cooling phase by seawater, and also has the advantage of high heat transfer performance. With such arrangement and careful considerations of the arrangement of wicks in heat pipes, and also the use of composite adsorbent (calcium chloride and activated carbon)-ammonia adsorption pair, the system test has shown the specific refrigeration power for more than 730 W/kg at 15 °C. Adsorption refrigeration systems integrated with heat pipes are quite efficient for real applications. The research work proved that heat pipes could be used as heat exchangers for adsorbers, evaporators or condensers. Proper designing may help to simplify the adsorption refrigeration system, make the system cost lower, and solve the problems of corrosions, etc. Based upon various types of heat-pipe design, adsorption water chiller, adsorption room air conditioner and adsorption ice maker for fishing boats have been successfully demonstrated. Some of adsorption the refrigeration systems have been even commercialized.

[6] ***Pracha Yeunyongkul et al.[2009]*** aimed at experimentally investigating the application of a closed loop oscillating heat pipe (CLOHP) as the condenser for a vapor compression refrigeration system. Split type air conditioner for residential use has two major disadvantages. First, it has a large pressure drop in the condenser caused by the flow of refrigerant inside a very small tube which affects compressor power and results in a decrease in the coefficient of performance (COP). Second, a large amount of heat is rejected to the surroundings since the refrigerant has to condense after passing through the condenser. To decrease pressure drop and recover heat rejection from the condensing process, this study considered using CLOHP instead of the conventional condenser in the split type air conditioner. The refrigeration capacity was set at 12,500 Btu/h (3.663 kW) with R22 as the refrigerant. The simulation of CLOHP condenser for the establishment of the optimum size of the vapor compression refrigeration system was performed using the herm-o-economical method. For the optimum system size, it was found that water as the working fluid provided the highest net savings. The optimum size of the system with water as the working fluid consists of a 0.08 meter of evaporator section length, a 0.1 meter of condenser section length, pipe with an inner diameter of 2.03 millimeter, and 250 turns. Therefore, these sizes were selected to construct the CLOHP condenser. The experimental results were obtained and compared with the conventional condenser. It was found that COP of CLOHP condenser with a heat load of 800 W was decreased by about 32.4% but the pressure drop of the CLOHP condenser was lower than that of the conventional condenser by about 91.2%.In addition, the energy efficiency rating of the CLOHP condenser was higher than the conventional condenser by about 13.4%. Finally, the outlet temperature of the cooling water which recovers heat from the condenser section of CLOHP, was increased by about 3°C. The same trend was also observed for the heat loads of 900 W and 1,000 W. They concluded that COPs of the conventional system with the heat loads of 800, 900 and 1,000 W were 4.92, 4.79 and 4.51, respectively while the COP of the CLOHP condenser system with the same heat loads were 3.28, 3.13 and 3.14, respectively. The COP of the CLOHP condenser system was obviously lower than the conventional system. The pressure drops of the conventional condenser with the same heat loads were 83.5, 82.7 and 69.7 kPa, respectively while the pressure drops of the CLOHP condenser were 6.89, 6.90 and 6.13 kPa, respectively. Therefore, the CLOHP condenser can definitely reduce the pressure drop when compare with the conventional system .The outlet temperature of the cooling water which recovers heat from the condenser section of CLOHP, was increased by about 3 °C showing the possibility of recovering heat for future utilization .To improve the COP, heat rejection rate and increase the outlet temperature of the cooling water which recover heat from the condenser section of CLOHP, in our future work will change the working fluid from water with other working fluids which have higher thermal efficiency and are environmentally safe, for example R134a.

[7] ***R. Rajashree et al. [1990]*** went through a numerical analysis of an unsteady, viscous, laminar, incompressible, two dimensional heat and mass transfer, in the vapour gas region of gas loaded circular heat pipe . The governing equations of motion were solved using Dufort-Frankel finite difference scheme with appropriate initial and boundary conditions. Successive over relaxation technique has been adopted to solve Poisson equation. The analysis is restricted to the vapour space of the cylindrical heat pipe with evaporator, adiabatic and condenser sections and the qualitative behaviors of the start-up transient in the vapour gas region has been presented with sodium as vapour for different Reynolds Number. The study demonstrated the qualitative behavior of heat and mass transfer can be studied with the above method with precision.

2.2 Conclusions from the literature review and research gap identified:

In the case of heat pipe an analysis based on multiple easily available working fluid, with different working conditions can be done. For an example liquid Sodium is not a vital fluid to be used for common refrigerating plants, where fluids as methanol, ethanol, water ammonia etc can make the use of heat pipe in the main stream. Also the main parameters of a heat pipe such as cross-sectional area of the wick, Heat Transfer Factor etc can be analysed on the basis of varying design variables such as operating temperature, pore radius, wick permeability, wick material, maximum heat to be transported etc . Along with being used in the VCR system it can be used in an ejector type refrigerating unit and vapour absorption and Electrolux and may enhance the performance of these systems.

In the case of observing the changes brought by a regenerative heat exchanger and a flash gas removing chamber along with two stage compression without inter-cooling with eco-friendly refrigerants a comparison with Simple VCR cycle can be observed and same for the VARS cycle can be done.

2.3 Analysis Conducted:

Variations of the heat pipe variables with design variables are tabulated and plotted as well. Three working fluids Water, Ammonia and Ethanol are chosen. To choose out of several eco-friendly refrigerants a quantitative analysis has been done in a generalised fashion.

2.4 Objectives:

- To analyse the parameters of the Heat Pipe and observe its variation with the desired conditions that are going to be present while the working of the refrigeration system ie. the Refrigeration Cpacity (RC) in Tonnage of Refrigeration (TR) , Condenser Temperature of the cycle T_c.
- To analyse the scope of three different working fluids for the heat pipe ie. Water, Ammonia and Ethanol and their suitability for being used in refrigeration.
- To analyse the sizes of evaporator , condenser and heat pipe for same desired output.

CHAPTER 3

PROBLEM FORMULATION

3.1 Equations Used in Heat Pipe:

Figure of Merit (MF_h):

$MF_h=\rho_l\ \sigma\ h_{fg}/\mu_l$

where:

ρ_l is the density of liquid in heat pipe in kg/m^3,

σ is the surface tension of the fluid in N/m,

h_{fg} is the latent heat of vaporization at heat pipe temperature in kJ/kg, and

μ_l is the viscosity of the liquid in Pa-s.

Maximum Heat transfer (Q_{max}) (Capillary Limitation):

$Q_{max}= (MF_h)(A_w k_{hp}/L_{eff})\{(2/r_p)-(\rho_l g\ L_{eff}\sin\phi\ /\ \sigma)\}$

where:

MF_h is the Figure of Merit in kW/m^2,

A_w is the cross sectional area of wick in m^2,

k_{hp} is the Permeability of the wick material in m^2,

L_{eff} is the effective length of the heat pipe in m,

r_p is the pore radius in m,

g is the acceleration due to gravity in m/s^2, and

Φ is the inclination angle in degrees.

Heat Transfer Factor (HTF)

$$HTF=(2^* MF_h^* A_w^* k_{hp}/r_p)$$

The symbols have usual predefined meaning.

3.2 The Problem Statement:

The commonly used VCRS cycle has its evaporator temperature at 268 K and condenser temperature at 318 K. Refrigerants such as R407C, R410A and R600a keeping R134a as a reference. The system has to have a capacity of 10 TR increasing upto50 TR at the evaporator temperature also the evaporator temperature will be observed from 253K to 273K. Considering the surrounding situation of the country the condenser temperature will be varied from 313K to 328K.

In the advanced study the condenser used in the VCRS will be replaced by a Heat pipe of a chosen wick material 30 mesh screens. The effect of the size change will be observed with the application of heat pipe. Also three different fluids for heat pipe namely Water, Ammonia and Ethanol will be checked for their merits and what effect would they have on the design of heat pipe. The heat taken by the condenser will now be absorbed by the heat pipe and a direct comparison for the size of these two components would be sought after.

The length of tubes in the evaporator and condenser and the effective length of the heat pipe are kept 1 meter in length.

3.3 The Selection of the working fluid in the heat pipe:

The Fluids used in the Heat Pipe are chosen on the basis of their merit:

1. Water
2. Ammonia
3. Ethanol

Also these fluids are very easily available. These also show the best Figure of Merit to be used in heat pipe applications. These fluids are cheap and non toxic in nature. Ethanol is flammable so it’s used must be strictly observed.

The latent heat of vaporisation and condensation is also a very big deciding factor for the fluid in heat pipe. The working liquid must have great thermo-physical properties at the predetermined operational temperature and weight. Working liquid must have high wettability, surface tension and other attractive thermo-physical properties incorporating a high fluid thermal conductivity, high inert enthalpy of vaporization, low fluid viscosity, and a low vapour viscosity. In this manner, the working liquid must be decided to consider this working temperature, additionally its concoction similarity with the holder and wick materials.

CHAPTER 4

RESULTS AND DICSUSSIONS

4.1 Analysis of the Heat Pipe:

As discussed in the earlier section the performance of a heat pipe can be analysed on the basis of a few important factors namely:

- Heat Transfer Factor (HTF),(W-m)
- Pore radius (r_p), (m)
- Effective Length (L_{eff}),(m)
- Figure of Merit of the Fluids. (MF_h), (kW/m^2)
- Permeability of the Wick Material (k_{hp}) (m^2)
- Angle of Inclination (ϕ),(degrees)
- Cross-sectional Area of Wick (A_w),(m^2)

The fluids used for the analysis purpose are Ammonia, Water and Ethanol as discussed earlier.

To begin the Figure of Merit would be analysed:

4.1.1 Figure of Merit (MF_h)

The figure of merit is the ratio of the product of density, surface tension and latent heat of evaporation to the dynamic viscosity of the fluid at saturated liquid state.

$MF_h=\rho_l*\sigma_l*h_{fg}/\mu_l$

As it can be seen clearly the Figure of Merit is the ratio of the thermo-physical properties it is dependant only on the temperature of working. So its variation is observed with the variation of temperature between the working limits of the condenser of the system ie. 40°C to 55°C.

The analysis will show which one of the three fluids will work better at higher temperatures with a higher capacity of absorbing heat, with smallest size.

The tables show the variation of the Figure of Merit with respect to the temperature of working of the heat pipe T_c.

T_c(K)	MF_h(Water)
313	22667
314.7	23197
316.3	23722
318	24244
319.7	24762
321.3	25276
323	25785
324.7	26289
326.3	26788
328	27282

MF_h(Ammonia)	T_c(K)
104326	313
100378	314.7
96529	316.3
92767	318
89095	319.7
85512	321.3
82017	323
78608	324.7
75284	326.3
72045	328

T_c(K)	MF_h(Ethanol)
313	1515
314.7	1538
316.3	1560
318	1582
319.7	1603
321.3	1625
323	1645
324.7	1665
326.3	1685
328	1704

Table 4.1: Parametric Table for Variation of Figure of Merit v/s Working Temperature

The following plots between the Figure of Merit and condenser temperature depict the changes with temperature in a more elaborate manner.

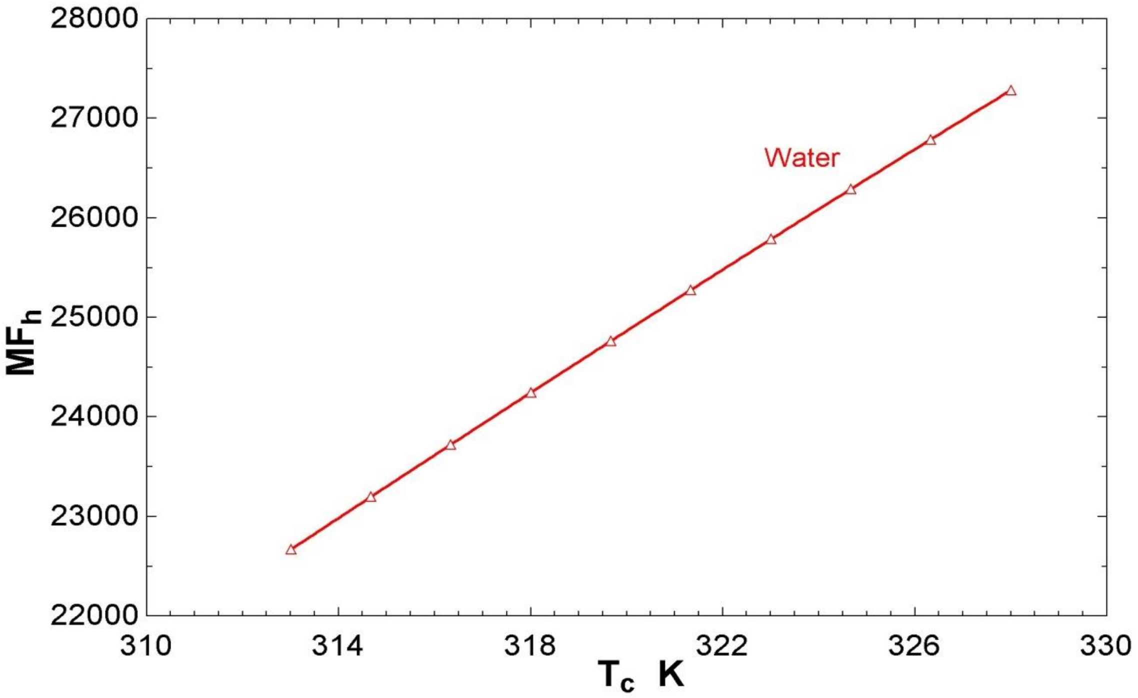


Fig 4.1: Variation Figure of Merit and condenser temperature for Water

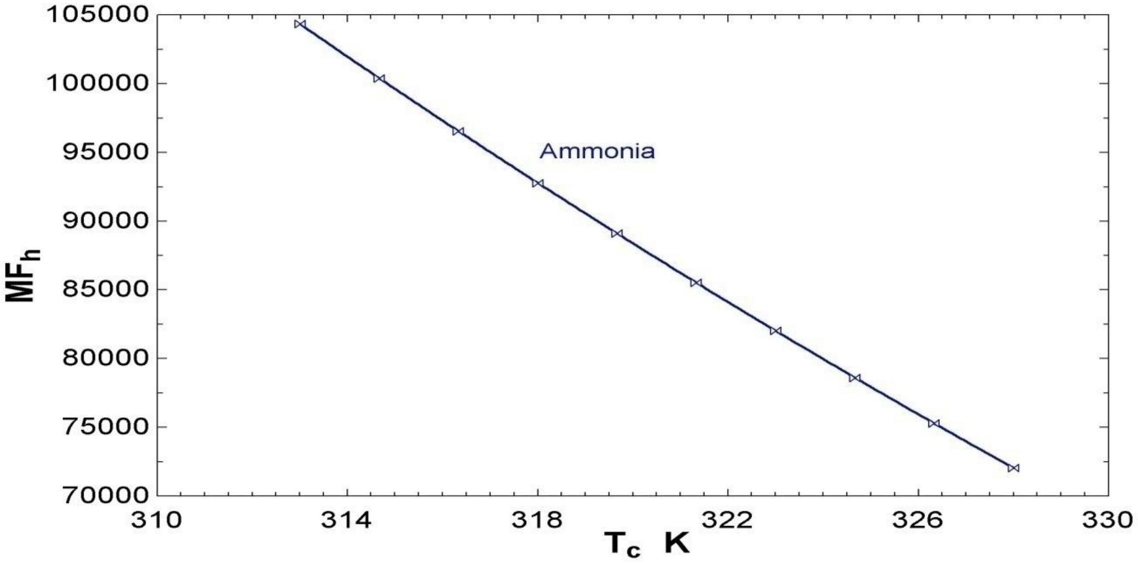


Fig 4.2: Variation of Figure of Merit and condenser temperature for Ammonia

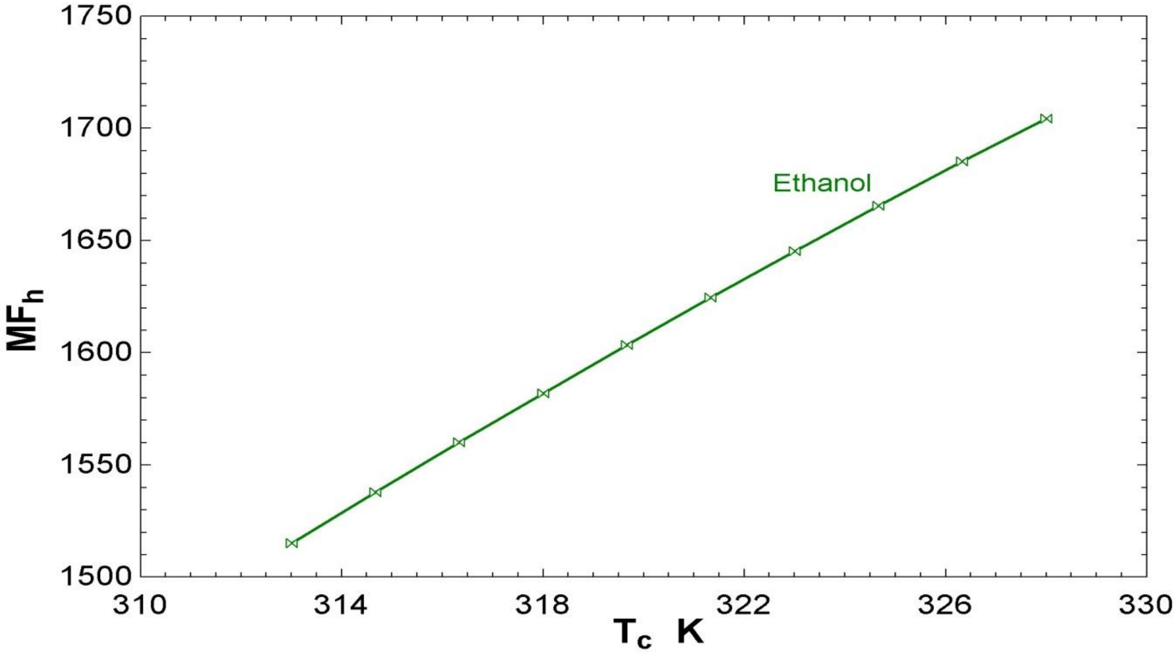


Fig 4.3: Variation of Figure of Merit and condenser temperature for Ethanol

The Variation is least and the value is smallest for Ethanol while its maximum for Ammonia, the values are medium for Water. Depending upon the applications if high heat transfer is going to be at higher temperatures water is the most suitable fluid for heat pipe. While ammonia has high merits at relatively lower temperatures.

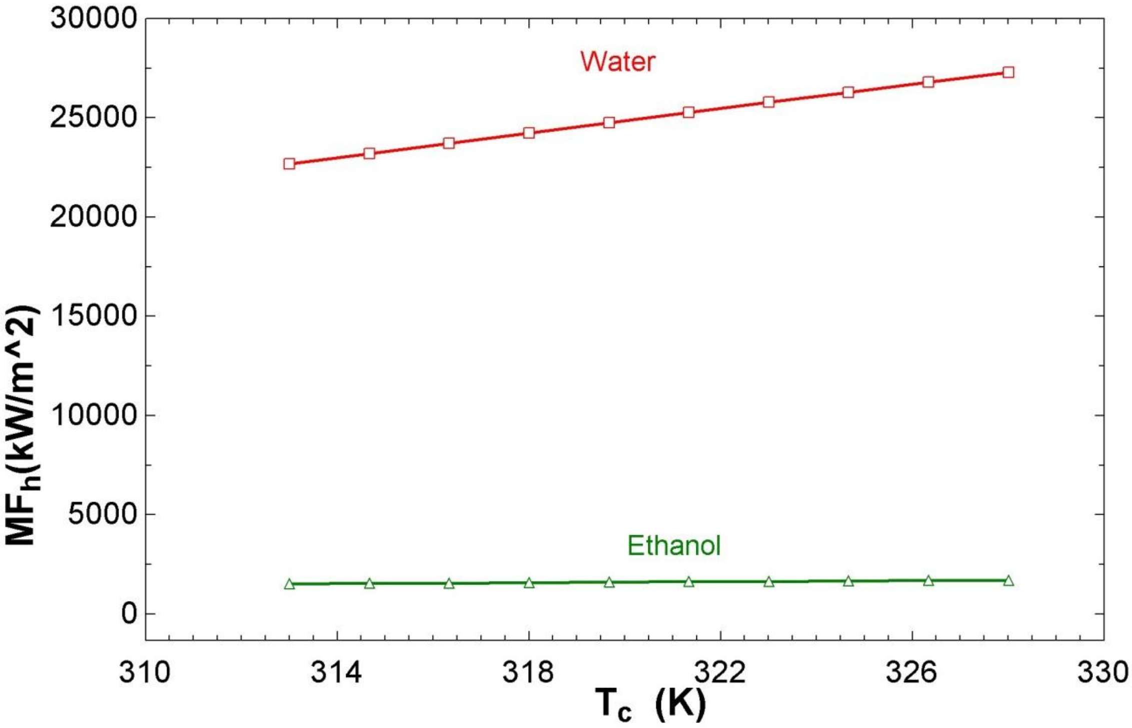


Fig 4.4: Variation of Figure of Merit and Heat pipe temperature for Water and Ethanol

Figure shows an overlay plot between Figure of Merit and Heat pipe temperature for Water and Ethanol. As already shown in the tables the least Merit is related to the Ethanol while Water has substantially large value.

4.1.2 Wick Area, Figure of Merit and Operating Temperature.

In another analysis the variation in wick area (A_w) is observed with the operating temperature (T_c) and the Figure of Merit (MF_h). The type of the plots obtained is 3-D and it’s taken between the wick area and the operating temperature keeping the Figure of Merit as the contour variable. First the parametric tables are obtained are presented.

$A_w(m^2)$	$T_c(K)$	$MF_h(kW/m^2)$
0.02182	313	22667
0.02158	314.1	23008
0.02135	315.1	23347
0.02113	316.2	23685
0.02092	317.3	24021
0.02071	318.4	24356
0.02051	319.4	24689
0.02032	320.5	25020
0.02013	321.6	25349
0.01995	322.6	25677
0.01978	323.7	26002
0.01961	324.8	26325
0.01945	325.9	26646
0.0193	326.9	26965
0.01915	328	27282

A_w (m^2)	MF_h (kW/m^2)	T_c (K)
0.01792	104326	313
0.01942	101780	314.1
0.02115	99271	315.1
0.02315	96801	316.2
0.02549	94368	317.3
0.02827	91973	318.4
0.03161	89614	319.4
0.03572	87293	320.5
0.04087	85008	321.6
0.04751	82759	322.6
0.05639	80545	323.7
0.06887	78368	324.8
0.08765	76225	325.9
0.1191	74118	326.9
0.1823	72045	328

Table 4.2: Variation in Wick Area vs. Operating Temperature and Figure of Merit

$A_w(m^2)$	$T_c(K)$	$MF_h(kW/m^2)$
0.7299	313	1515
0.728	314.1	1530
0.7263	315.1	1544
0.7248	316.2	1558
0.7233	317.3	1573
0.722	318.4	1587
0.7209	319.4	1600
0.7199	320.5	1614
0.719	321.6	1628
0.7183	322.6	1641
0.7177	323.7	1654
0.7173	324.8	1667
0.7169	325.9	1680
0.7167	326.9	1692
0.7167	328	1704

Table 4.2: Variation in Wick Area vs. Operating Temperature and Figure of Merit

The parametric tables correspond to the values for Water, Ammonia and Ethanol respectively.

On the basis of the parametric tables the following plots are obtained.

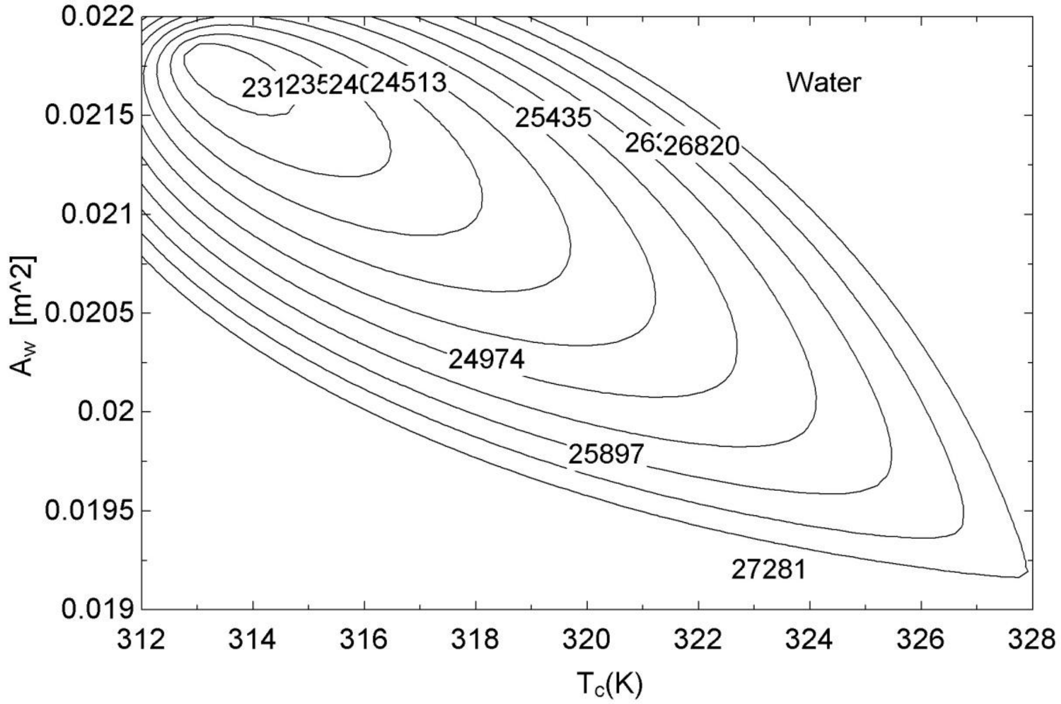


Fig 15: Variation of Wick Area with operating temperature and Figure of Merit of heat pipe for Water

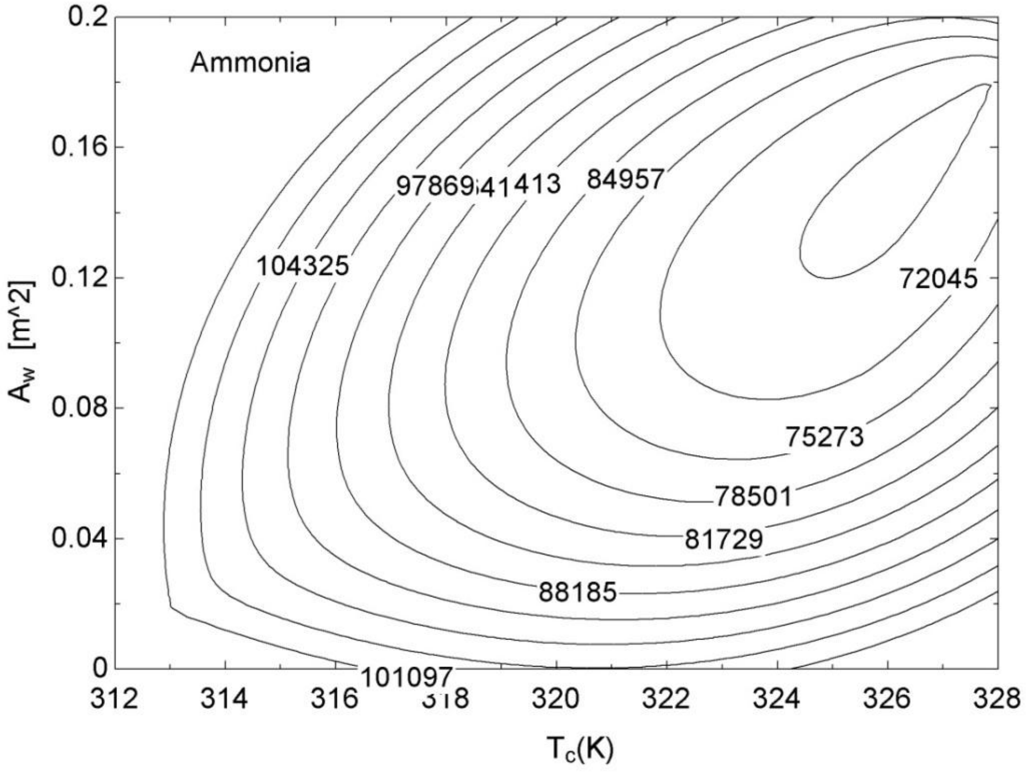


Fig 4.6: Variation of Wick Area , operating temperature and Figure of Merit of heat pipe for Ammonia

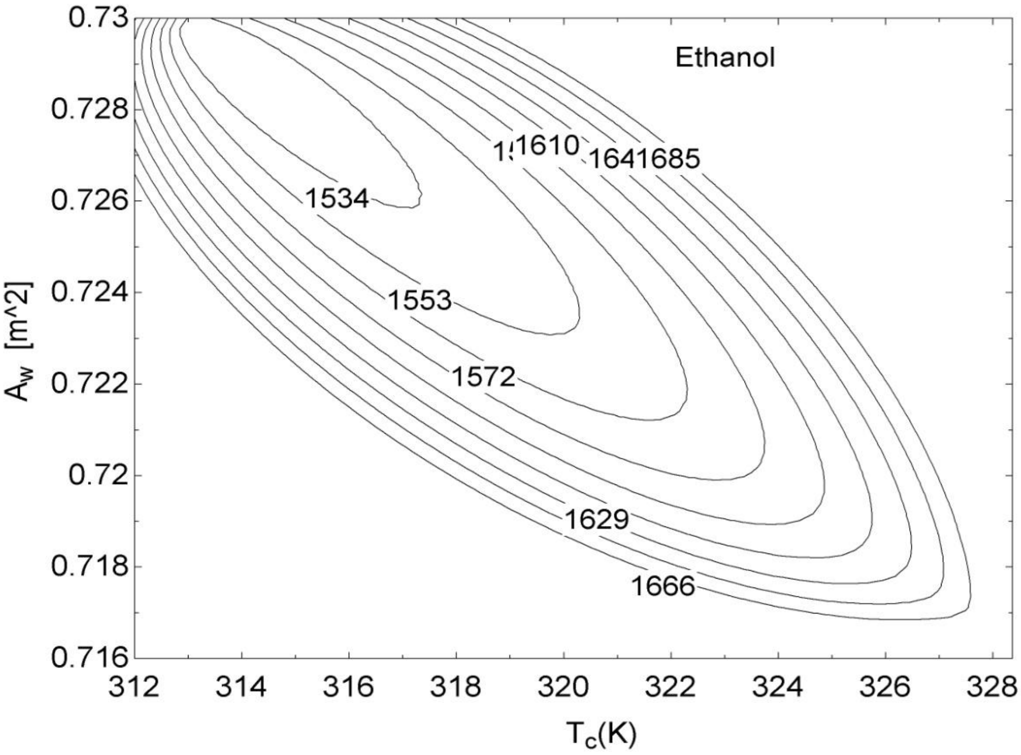


Fig 4.7: Variation of Wick Area with Operating Temperature and Figure of Merit of heat pipe for Ethanol

It can be observed that at higher temperatures Water is providing the lowest possible area where as the lowest area at lower temperatures is required if Ammonia is taken the working fluid.

Ethanol on the other hand shows the similar variations as water but requires more wick area than both the other fluids.

The figure of merits has a big effect for deciding the wick area. More the figure of merits lesser will be the area.

This can be seen in another set of plots which are over lay of two different working fluids. A 2-D plot has been developed between the wick area and the condenser temperature at which the heat pipe is working.

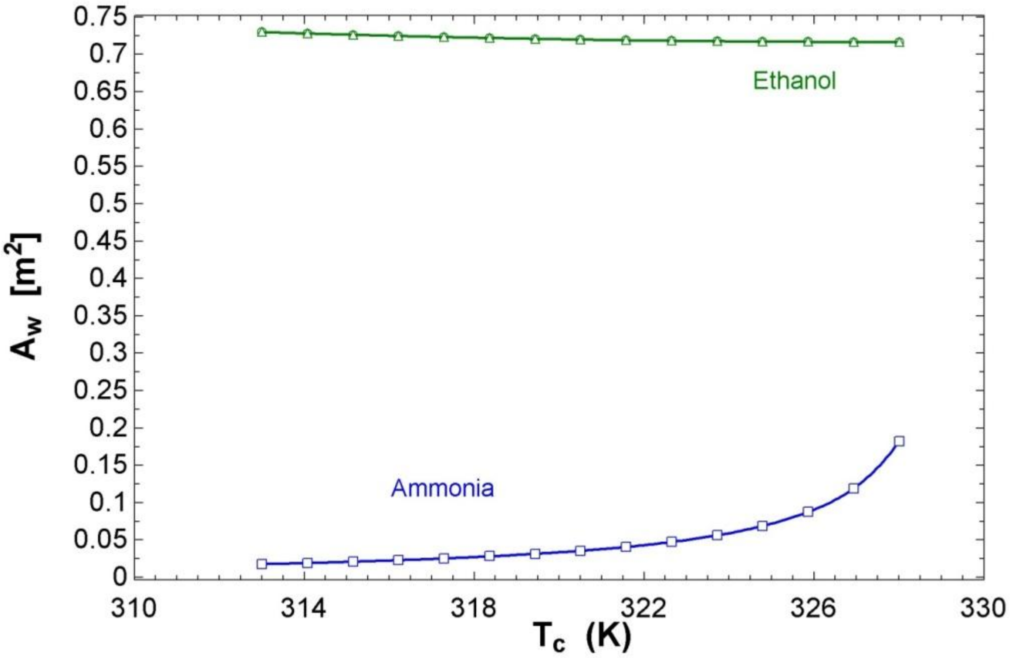


Fig 4.8: Variation of Wick Area with operating temperature of heat pipe for Ethanol and Ammonia

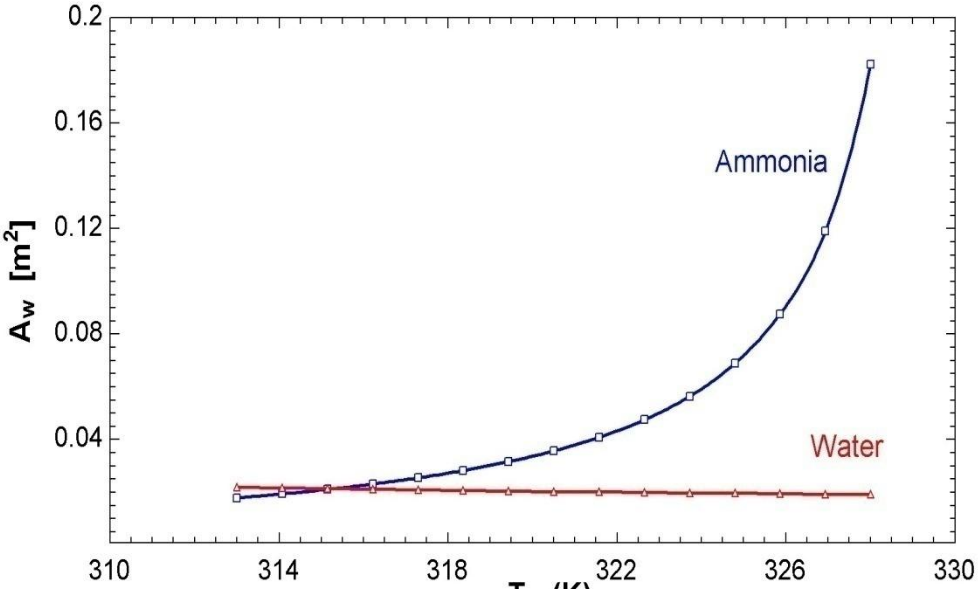


Fig 4.9: Variation of Wick Area with operating temperature of heat pipe for Ammonia and Water

The difference between the wick area required for Ethanol and Ammonia can be observed very easily, whereas the same required for Ammonia and water are comparable. Size is affected by the wick area required per unit effective length.

4.1.3 Wick Area and Refrigerating Capacity:

The Refrigerating capacity is a very important parameter in any refrigeration plant. The capacity must be enhanced, size and the power consumption must be reduced to obtain a desired result. Hence the wick area with respect to the maximum heat absorbed by the heat pipe(Q_{max}) and the Refrigerating capacity is studied. The parametric tables are as follows:

$A_w(m^2)$	Q_{max}(kW)	RC(TR)
0.02078	42.02	10
0.02671	54.03	12.86
0.03265	66.04	15.71
0.03859	78.04	18.57
0.04452	90.05	21.43
0.05046	102.1	24.29
0.0564	114.1	27.14
0.06233	126.1	30
0.06827	138.1	32.86
0.07421	150.1	35.71
0.08014	162.1	38.57
0.08608	174.1	41.43
0.09201	186.1	44.29
0.09795	198.1	47.14
0.1039	210.1	50

$A_w(m^2)$	RC(TR)	Q_{max}(kW)
0.02729	10	42.02
0.03508	12.86	54.03
0.04288	15.71	66.04
0.05067	18.57	78.04
0.05847	21.43	90.05
0.06626	24.29	102.1
0.07406	27.14	114.1
0.08186	30	126.1
0.08965	32.86	138.1
0.09745	35.71	150.1
0.1052	38.57	162.1
0.113	41.43	174.1
0.1208	44.29	186.1
0.1286	47.14	198.1
0.1364	50	210.1

Table 4.3: Variation in wick area, heat transfer and the refrigerating capacity

$A_w(m^2)$	RC(TR)	Q_{max}(kW)
0.7225	10	42.02
0.9289	12.86	54.03
1.135	15.71	66.04
1.342	18.57	78.04
1.548	21.43	90.05
1.755	24.29	102.1
1.961	27.14	114.1
2.167	30	126.1
2.374	32.86	138.1
2.58	35.71	150.1
2.787	38.57	162.1
2.993	41.43	174.1
3.199	44.29	186.1
3.406	47.14	198.1
3.612	50	210.1

Table 4.3: Variation in wick area, heat transfer and the refrigerating capacity

The tables show the relation between the wick area, heat transfer and the refrigerating capacity. Based on these tables generated plots are as follows:

First 3-D plot between the wick area and the refrigerating capacity is presented keeping the heat transfer as the contour variable.

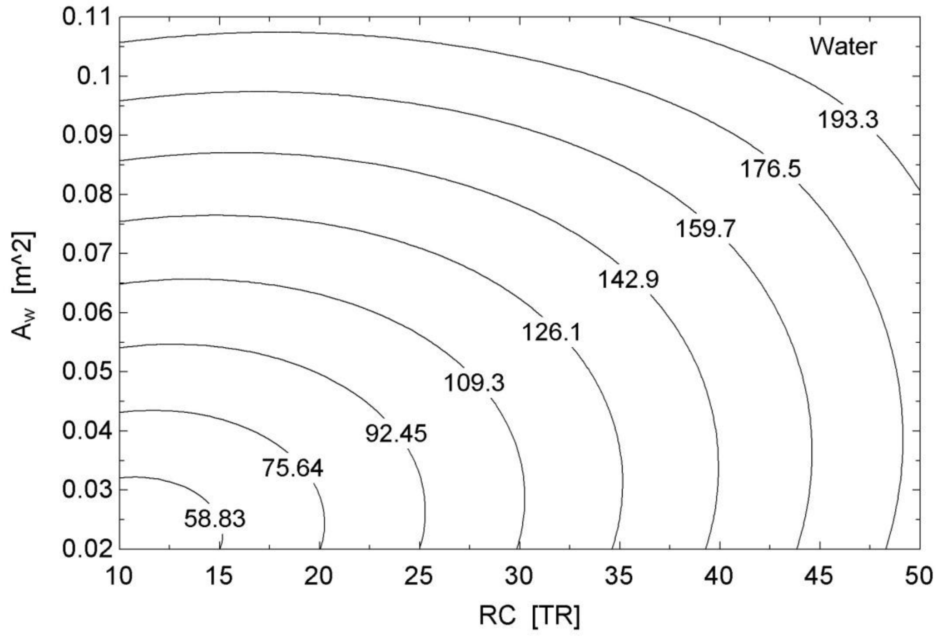


Fig 4.10: Plot between Wick area and the Refrigerating Capacity for Water

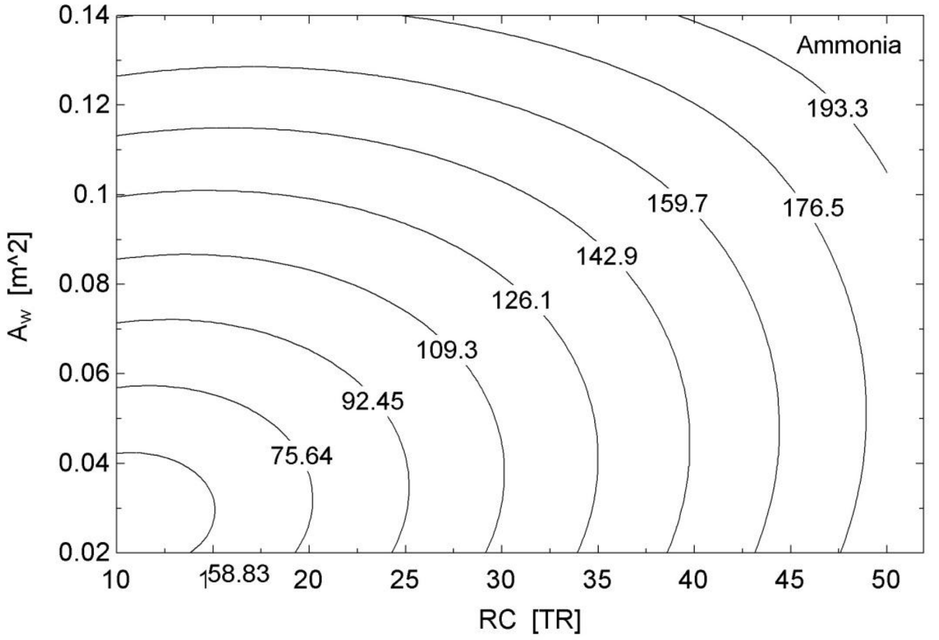


Fig 4.11: Plot between Wick area and the Refrigerating Capacity for Ammonia

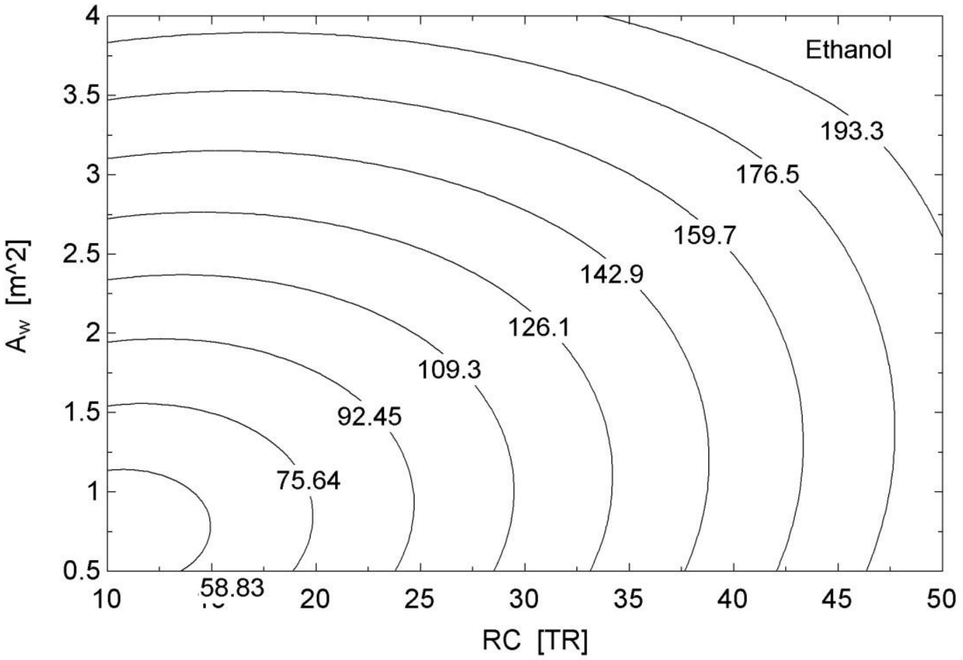


Fig 4.12: Plot between Wick area and the Refrigerating Capacity for Ethanol

As it was expected the wick area is increasing with the capacity in all the three cases.

Another over lay plot for Ammonia and Water is presented.

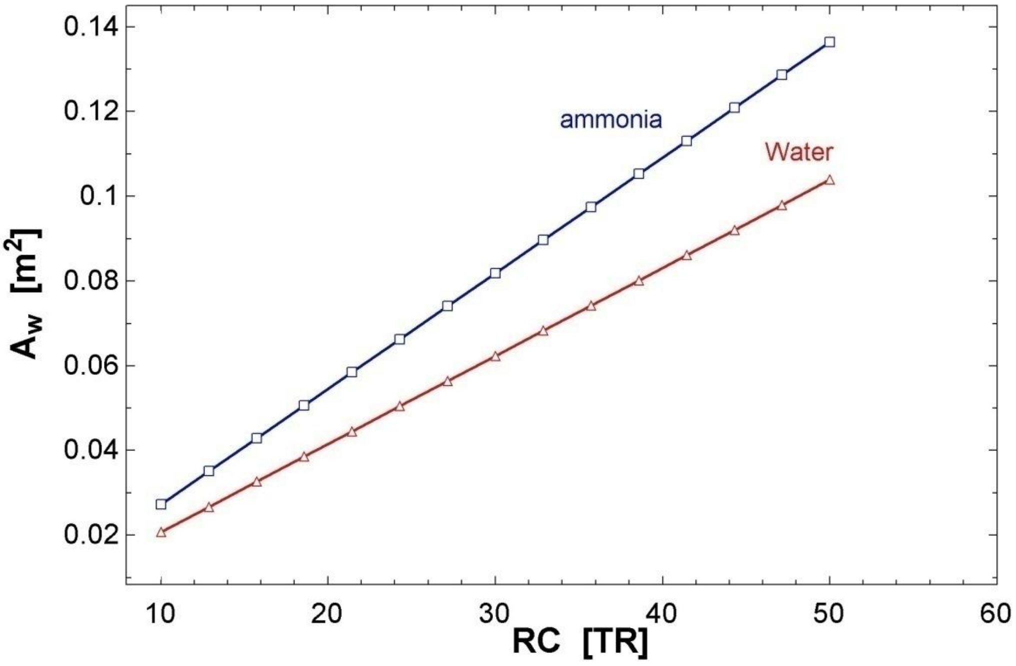


Fig 4.13: Variation of Wick area with refrigerating capacity for Ammonia and Water

The wick areas for Ammonia and Water are comparable and close enough. But water proves out to be better in this analysis. The separate variation for Ethanol is also shown.

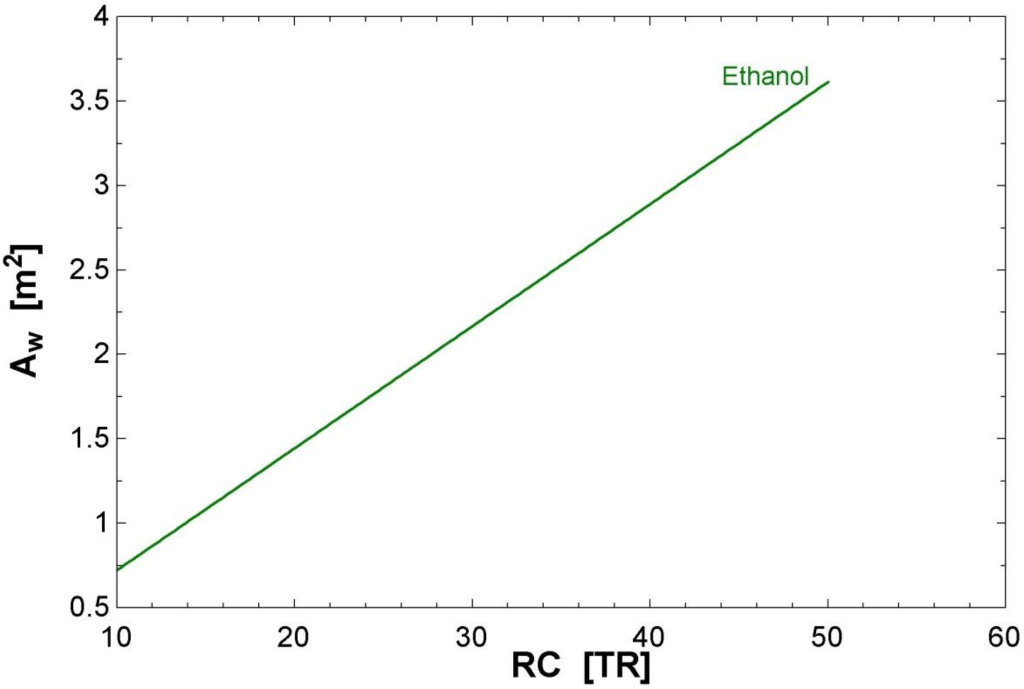


Fig 4.14: Variation of Wick area with refrigerating capacity for Ethanol

As it can be seen easily Water is giving a lesser area of heat pipe for the same capacity of the plant. The capacity can be increased and further analysis cab be done.

4.1.4 Wick Area, Figure of Merit, Heat Transfer Factor and Heat Pipe Temperature:

Heat transfer factor is another factor in the analysis which will affect the size of the heat pipe. It depends on permeability, pore radius, Figure of Merit and the wick area. Its unit is W-m, and plays an important in the designing of a heat pipe. Parametric tables between these variables are as follows:

The table respectively shows the variation for Water, Ammonia and Ethanol.

$A_w(m^2)$	HTF(W-m)	MF_h (kW/m^2)	$T_c(K)$
0.01645	43.36	22667	313
0.01627	43.52	23008	314.1
0.01609	43.68	23347	315.1
0.01592	43.84	23685	316.2
0.01575	43.99	24021	317.3
0.01559	44.15	24356	318.4
0.01543	44.31	24689	319.4
0.01528	44.47	25020	320.5
0.01514	44.63	25349	321.6
0.015	44.78	25677	322.6
0.01486	44.94	26002	323.7
0.01473	45.1	26325	324.8
0.01461	45.26	26646	325.9
0.01448	45.42	26965	326.9
0.01437	45.58	27282	328

Table 4.4: Table of Heat Transfer Factor, Wick Area and Figure of Merit

$A_w(m^2)$	MF_h (kW/m^2)	HTF(W-m)	$T_c(K)$
0.00394	104326	47.78	313
0.00406	101780	48.04	314.1
0.00419	99271	48.31	315.1
0.00432	96801	48.57	316.2
0.00445	94368	48.84	317.3
0.00459	91973	49.12	318.4
0.00474	89614	49.4	319.4
0.0049	87293	49.68	320.5
0.00506	85008	49.97	321.6
0.00522	82759	50.26	322.6
0.0054	80545	50.56	323.7
0.00558	78368	50.86	324.8
0.00577	76225	51.17	325.9
0.00597	74118	51.48	326.9
0.00618	72045	51.8	328

Table 4.4: Table of Heat Transfer Factor, Wick Area and Figure of Merit

$A_w(m^2)$	HTF(W-m)	MF_h (kW/m^2)	T_c(K)
0.2645	46.6	1515	313
0.263	46.78	1530	314.1
0.2615	46.96	1544	315.1
0.2601	47.13	1558	316.2
0.2587	47.31	1573	317.3
0.2574	47.49	1587	318.4
0.2561	47.66	1600	319.4
0.2549	47.84	1614	320.5
0.2537	48.02	1628	321.6
0.2526	48.2	1641	322.6
0.2515	48.38	1654	323.7
0.2505	48.55	1667	324.8
0.2495	48.73	1680	325.9
0.2486	48.91	1692	326.9
0.2477	49.09	1704	328

Table 4.4: Table of Heat Transfer Factor, Wick Area and Figure of Merit

Based on these tables plots are generated. First a 3-D plot is developed between the HTF and MF_h keeping the operating temperature as the contour variable.

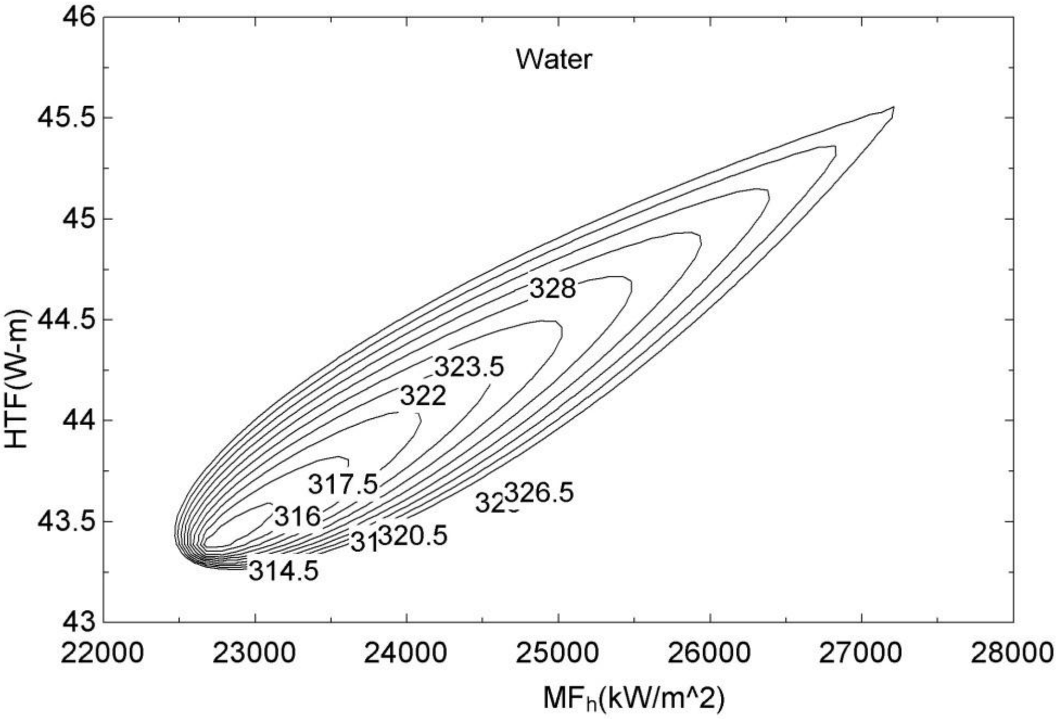


Fig 4.15: Plot between HTF and Figure of Merit for Water

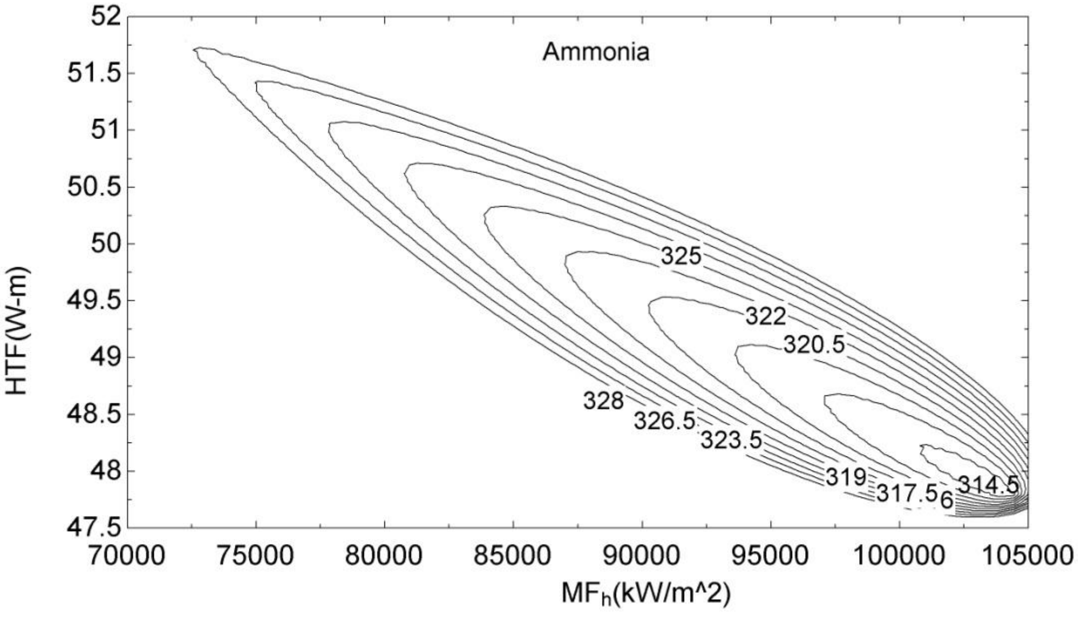


Fig 4.16: Plot between HTF and Figure of Merit for Ammonia

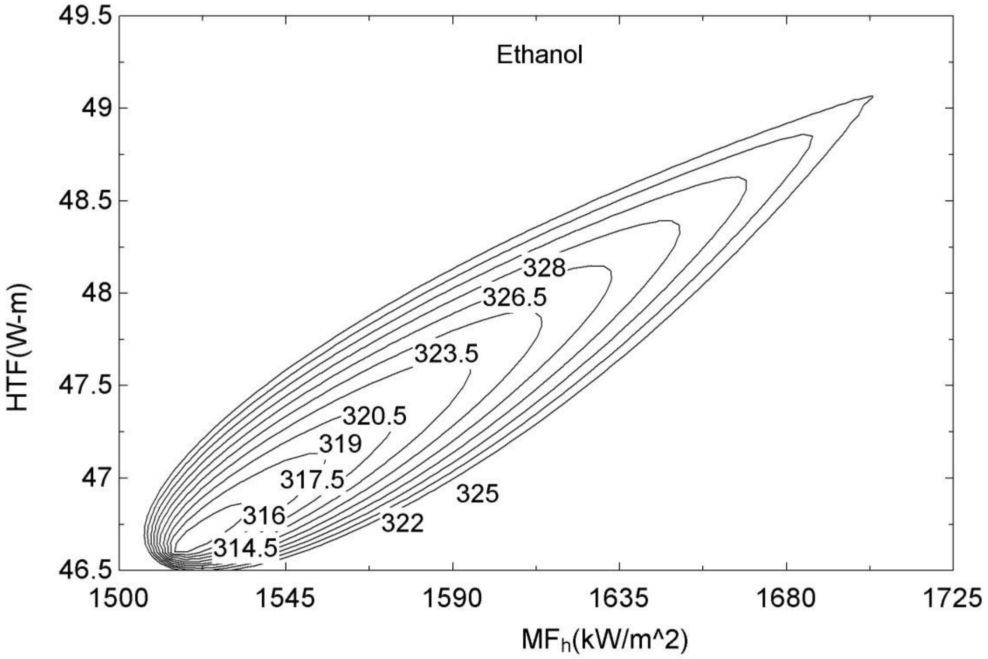


Fig 4.17: Plot between HTF and Figure of Merit for Ethanol

4.1.5 Wick Area and Inclination Angle of Heat Pipe:

The inclination angle of the heat pipe also affects the wick area required in the heat pipe. The inclination angle (ϕ) is measured in degrees. The parametric tables are presented as follows:

$A_w(m^2)$	Φ(degree)
0.01491	0
0.01568	0.1053
0.01655	0.2105
0.01751	0.3158
0.01859	0.4211
0.01982	0.5263
0.02122	0.6316
0.02283	0.7368
0.0247	0.8421
0.02691	0.9474
0.02956	1.053
0.03278	1.158
0.03679	1.263
0.04192	1.368
0.04871	1.474
0.05812	1.579
0.07205	1.684
0.09474	1.789
0.1383	1.895
0.2559	2

$A_w(m^2)$	Φ(degree)
0.0039	0
0.00459	0.1053
0.00557	0.2105
0.0071	0.3158
0.00978	0.4211
0.01571	0.5263
0.03989	0.6316
-0.0739	0.7368
-0.0192	0.8421
-0.011	0.9474
-0.0077	1.053
-0.006	1.158
-0.0048	1.263
-0.0041	1.368
-0.0035	1.474
-0.0031	1.579
-0.0028	1.684
-0.0025	1.789
-0.0023	1.895
-0.0021	2

$A_w(m^2)$	Φ(degree)
0.2284	0
0.2596	0.1053
0.3006	0.2105
0.3569	0.3158
0.4392	0.4211
0.5709	0.5263
0.8152	0.6316
1.425	0.7368
5.667	0.8421
-2.869	0.9474
-1.145	1.053
-0.715	1.158
-0.5198	1.263
-0.4084	1.368
-0.3363	1.474
-0.2858	1.579
-0.2485	1.684
-0.2199	1.789
-0.1971	1.895
-0.1786	2

Table 4.5: Wick Area and Inclination Angle of Heat Pipe

These tables show the variations of wick area with the inclination angle for Water, Ammonia and Ethanol respectively. The angle has been varied from 0 to 2 degree.

Plots are developed on this basis which is following on the next page. First the variation is taken for Ethanol, the other one shows the overlay plot for Water and ammonia.

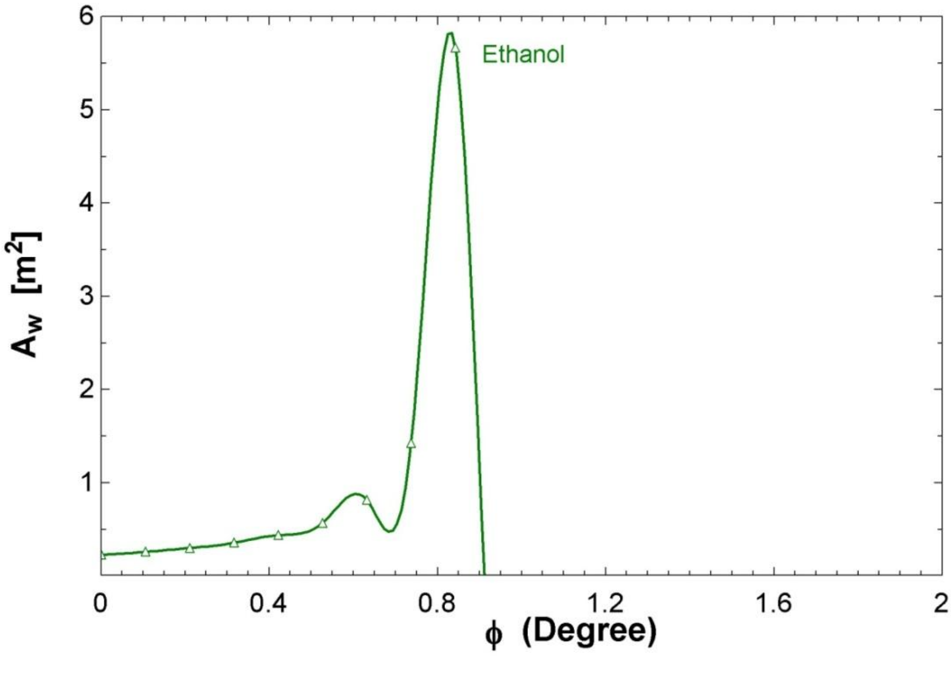


Fig 4.18: Variation of Wick Area with Inclination Angle for Ethanol

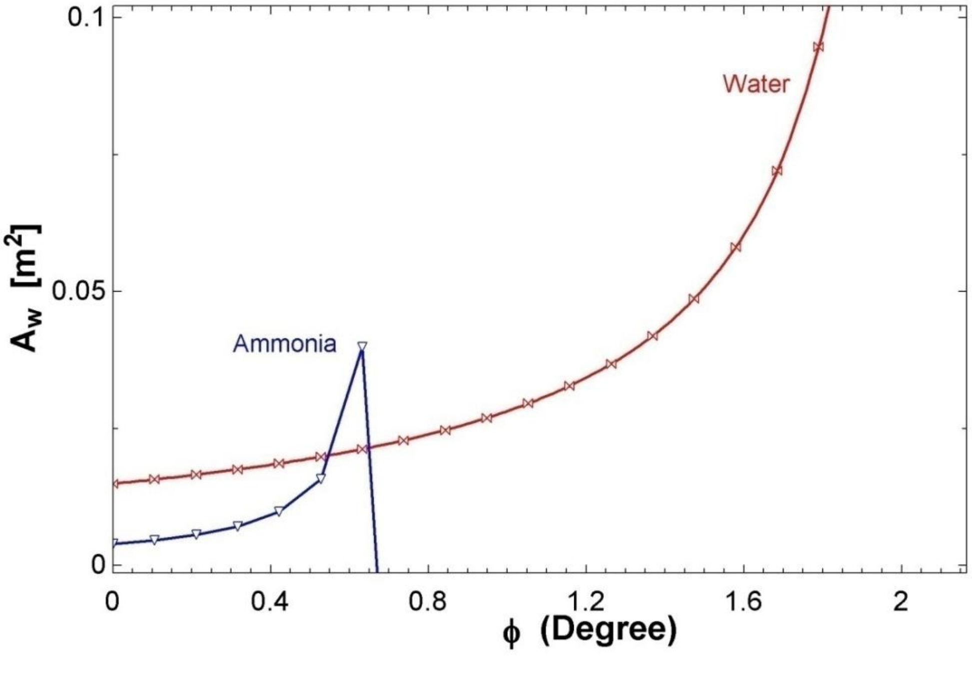


Fig 4.19: Variation of Wick Area with Inclination Angle for Ammonia and Water

4.1.6 Wick Area and Heat Transfer Factor:

The overlay plot for Water, Ammonia and Ethanol is prepared on the basis of the previous parametric tables:

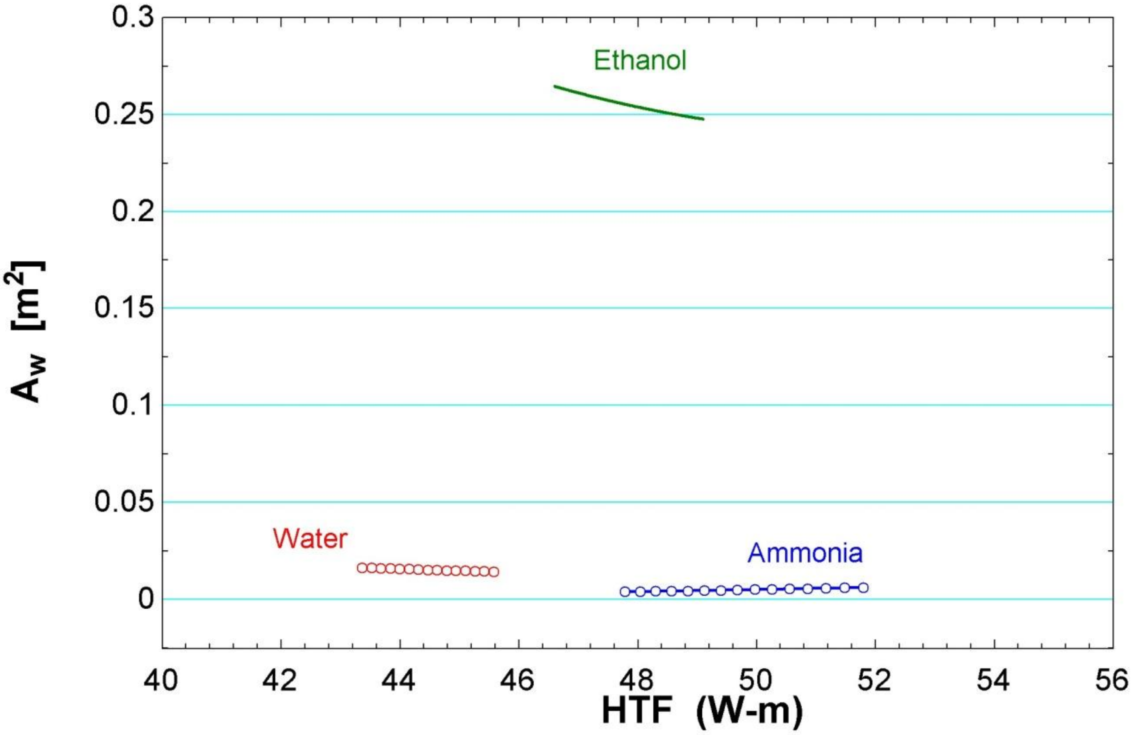


Fig 4.20: Overlay Plot for Water, Ammonia and Ethanol

As it can be seen the area required for Ethanol is maximum and minimum Ammonia. The Heat Transfer Factor is largest for the Ammonia and smallest for Water.

4.1.7 Wick Area and Pore Radius:

Another analysis is conducted to obtain the effect of pore radius r_p on the wick area. Parametric tables are presented as follows. The tables show respectively the relation between the A_w and r_p Water, Ammonia and Ethanol. The widest range of acceptable inclination angles is obtained for water, where as for ammonia area tends to negative for higher values of angle.

$A_w(m^2)$	HTF(W-m)	$r_p(m)$
0.0145	43.95	0.0004
0.02455	45.28	0.00066
0.03522	46.7	0.00091
0.04659	48.21	0.00117
0.05871	49.82	0.00143
0.07167	51.54	0.00169
0.08556	53.38	0.00194
0.1005	55.36	0.0022
0.1165	57.5	0.00246
0.1339	59.8	0.00271
0.1527	62.3	0.00297
0.1731	65.01	0.00323
0.1955	67.97	0.00349
0.2199	71.22	0.00374
0.2468	74.79	0.004

$A_w(m^2)$	HTF(W-m)	$r_p(m)$
0.00418	48.46	0.0004
0.00762	53.76	0.00066
0.0119	60.36	0.00091
0.01738	68.8	0.00117
0.02464	79.99	0.00143
0.03472	95.53	0.00169
0.04966	118.6	0.00194
0.07409	156.2	0.0022
0.1213	228.9	0.00246
0.2506	428.2	0.00271
2.119	3308	0.00297
-0.4021	-577.7	0.00323
-0.1997	-265.7	0.00349
-0.1392	-172.5	0.00374
-0.1101	-127.7	0.004

Table 4.6: Parametric Tables for Variation of Wick Area and Pore radius

$A_w(m^2)$	HTF(W-m)	$r_p(m)$
0.2377	47.01	0.0004
0.4228	50.88	0.00066
0.6411	55.46	0.00091
0.9026	60.94	0.00117
1.221	67.63	0.00143
1.619	75.96	0.00169
2.128	86.63	0.00194
2.803	100.8	0.0022
3.743	120.5	0.00246
5.139	149.8	0.00271
7.431	197.8	0.00297
11.89	291.2	0.00323
24.32	551.9	0.00349
248.8	5257	0.00374
-35.32	-698.5	0.004

Table 4.6: Parametric Tables for Variation of Wick Area and Pore radius

Plots are developed on the basis of these analysis between A_w & r_p and HTF& $r_{p.}$

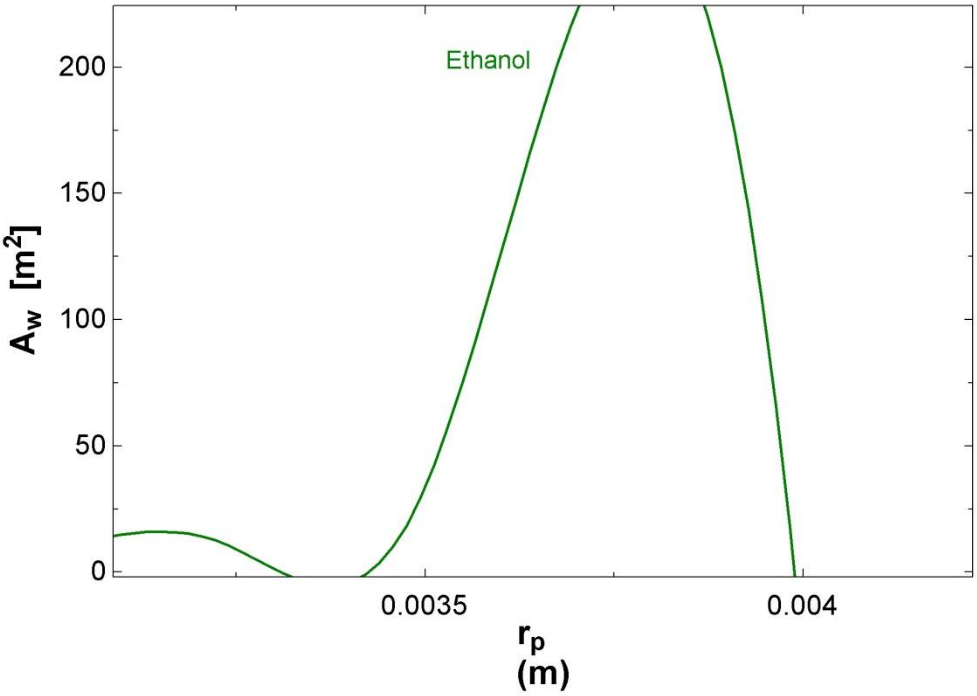


Fig 4.21: Plot between Wick Area and Pour Radius for Ethanol

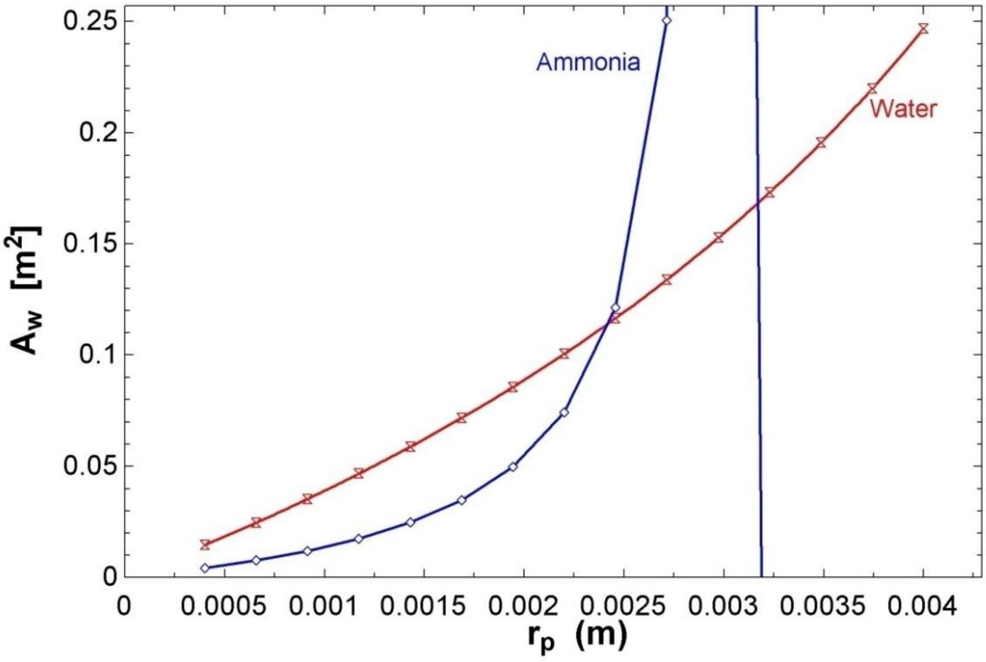


Fig 4.22: Plot between Wick Area and Pour Radius for Ammonia and Water

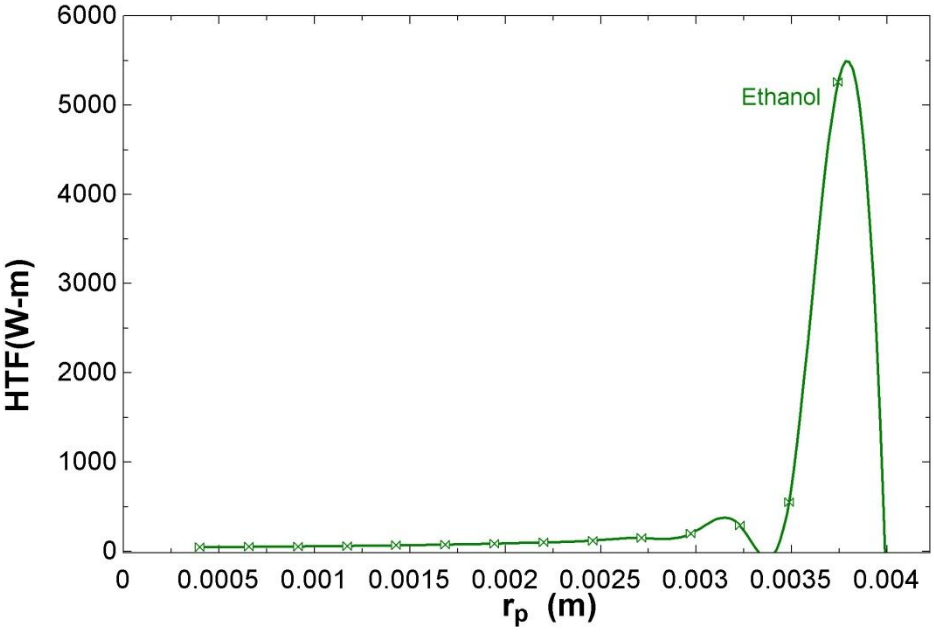


Fig 4.23: Plot between HTF and Pour Radius for Ethanol

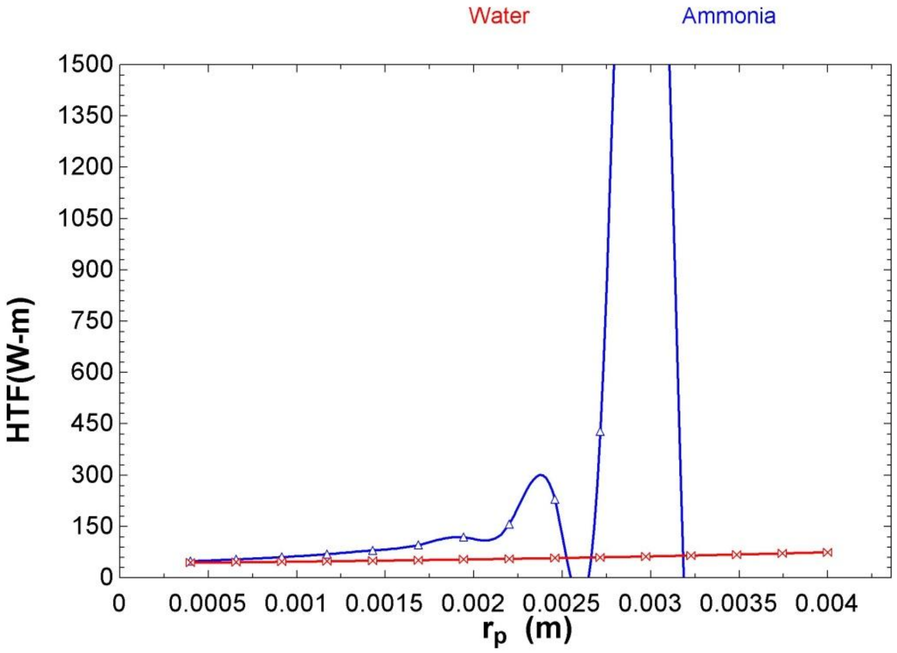


Fig 4.24: Plot between HTF and Pour Radius for Ammonia and Water

The first of the plots is drawn for Ethanol, and the other is drawn as an overlay plot for Water and Ammonia.

4.1.8 Comparison between The Cross Sectional Area of the Wick And Condenser Area:

The heat pipe in this analysis replaces the conventional condenser which is deemed to be one of the essential parts of the refrigeration cycle. Be it a VCRS cycle or a VARS cycle condenser has always been there. The main advantage of a heat pipe is that it can be observed that its size will not be very huge even if the capacity of the system has to be enhanced. In this analysis Parametric Tables are prepared for condenser area, wick area, evaporator area with varying operating temperatures and refrigerating capacity and maximum heat transferred in heat pipe and condenser. The purpose is to show the difference in the condenser size, evaporator size and wick area of a heat pipe all the three having unit length.

$A_w(m^2)$	$A_{ic}(m^2)$	$T_c(K)$
0.2645	13.37	313
0.263	9.764	314.1
0.2615	7.901	315.1
0.2601	6.705	316.2
0.2587	5.854	317.3
0.2574	5.211	318.4
0.2561	4.706	319.4
0.2549	4.297	320.5
0.2537	3.958	321.6
0.2526	3.672	322.6
0.2515	3.427	323.7
0.2505	3.215	324.8
0.2495	3.03	325.9
0.2486	2.866	326.9
0.2477	2.721	328

Table 4.7: Table for Condenser Area, Wick Area and Operating Temperature

$A_{ic}(m^2)$	$A_w(m^2)$	$Q_{max}(kW)$
5.147	0.2454	40
5.699	0.2717	44.29
6.25	0.298	48.57
6.802	0.3243	52.86
7.353	0.3506	57.14
7.905	0.3769	61.43
8.456	0.4032	65.71
9.008	0.4295	70
9.559	0.4558	74.29
10.11	0.4821	78.57
10.66	0.5084	82.86
11.21	0.5347	87.14
11.76	0.561	91.43
12.32	0.5873	95.71
12.87	0.6136	100

Table 4.8: Table for Heat capacity, Condenser Area and Wick Area

Here A_{ic} stands for the inner side area of the condenser. Q_{max} is the maximum heat extracted by the condenser and heat pipe separately.

Plots developed on the basis of these parametric tables are as follows. Also the comparison of Evaporator area with wick are and condenser area with increasing capacity is plotted.

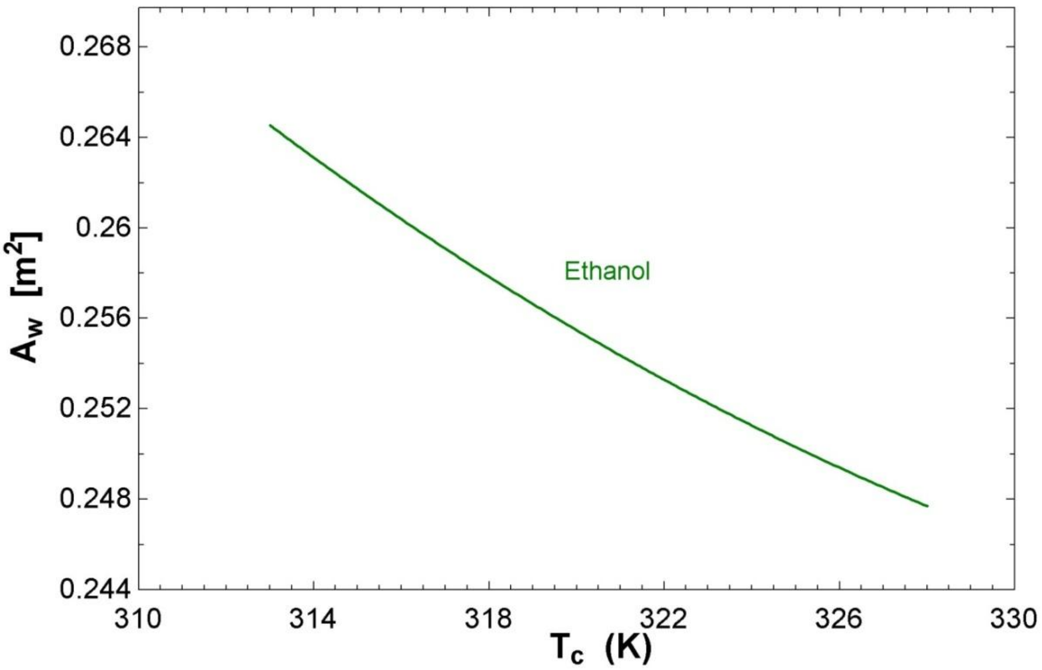


Fig 4.25: Variation of Wick Area with condenser temperature for Ethanol

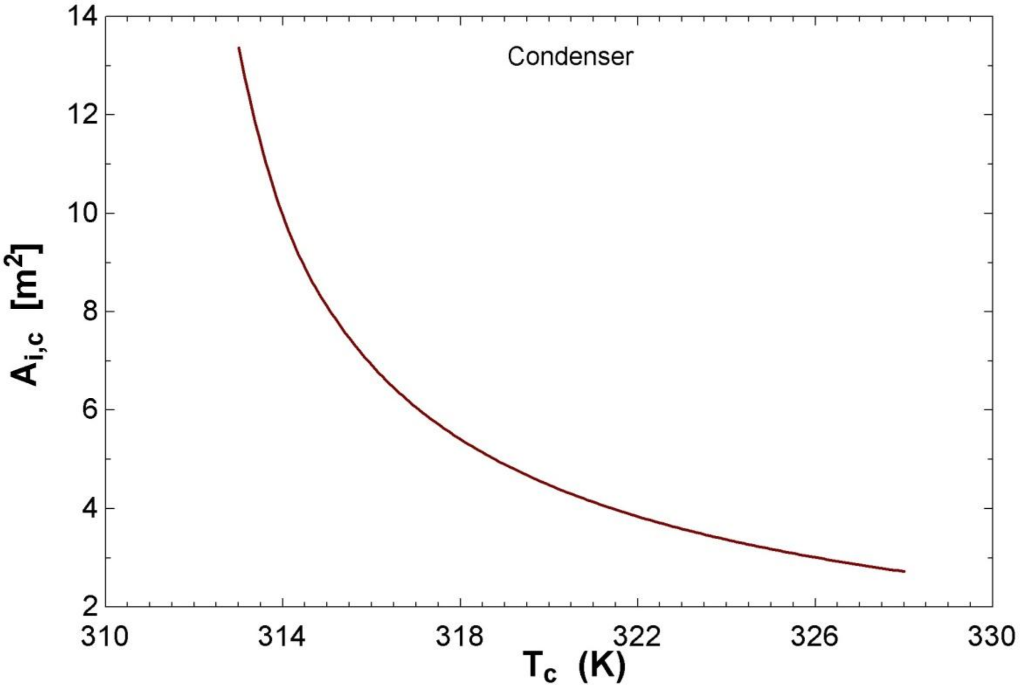


Fig 4.26: Variation of condenser area with operating temperature

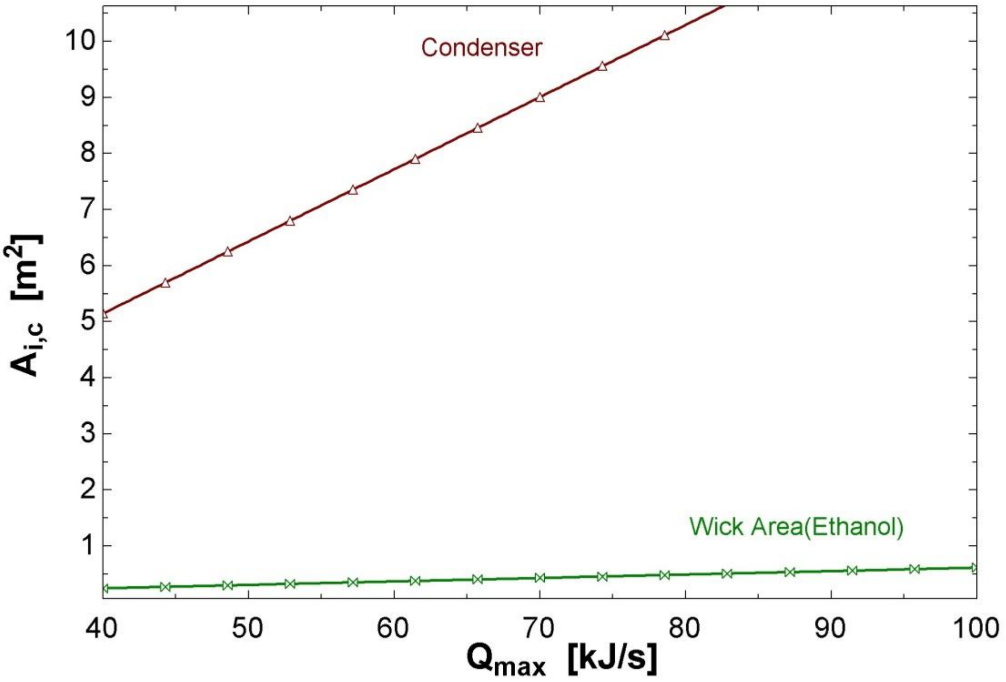


Fig 4.27: Variation of Condenser area and wick area with heat capacity

The Cross-Sectional area of the wick is very small when compared to the heat transfer area of the condenser which result reduced size and capacity of the plant will also will be high.

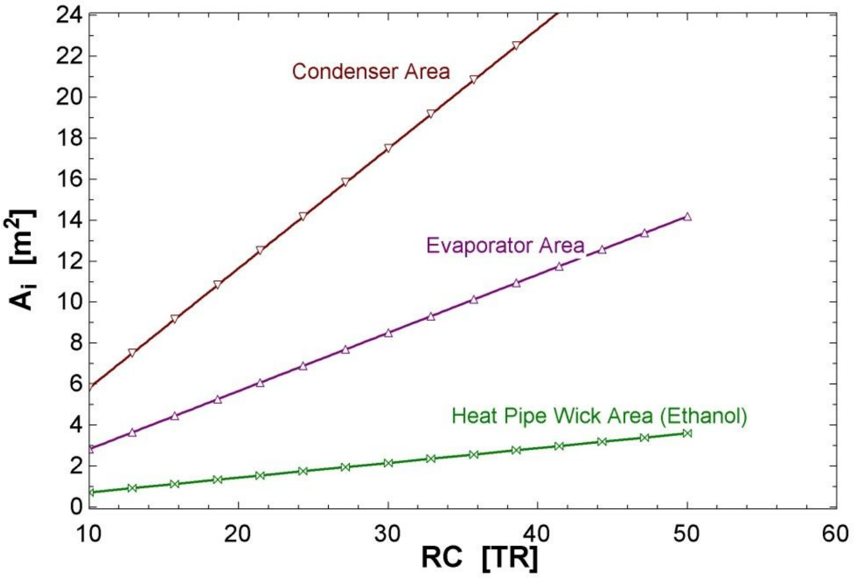


Fig 4.28: Variation of condenser, evaporator and wick area with refrigerating capacity

Figure shows the inner condenser area, inner evaporator area and wick cross-sectional area.

CHAPTER 5

CONCLUSIONS

From the results of the above research work the following conclusions can be made:

- The Figure of Merit of the working fluids Ammonia has the largest and Ethanol has the least merit. But Water shows the best possible results for the operating temperature of the heat pipe.
- As the temperature of heat pipe raises the merit of Water and Ethanol show increase and for Ammonia it decreases. As the Figure of Merit increases, the required cross-sectional area of the wick decreases. Expected variations have been observed between working temperature and wick cross-sectional area ie. with increase in temperature the area increases. For Ammonia it has a sharp rise as the temperature decreases and for Water and Ethanol the area is almost constant.
- As the capacity of refrigeration is increased maximum wick area is required by Ethanol and the least is for water. The same variations can be observed between the Heat transfer Factor and Wick Cross-sectional area.
- Angle of inclination also has an interesting effect on the on the area of the wick, so the inclination has to be chosen to optimize the heat transfer, the wick area and the cost of the heat pipe.
- For the range of temperature we are dealing with maximum heat transfer factor has been obtained for Ammonia, and the area will be the maximum for the Ethanol. With increase in the pore radius the area first increases then decreases after a certain limit, hence it has to be chosen optimally. The heat transfer factor also shows a similar variation with the pore radius.
- With increasing capacity the wick area increases slightly when compared to the increase in a conventional condenser. The low wick area will render a small overall area of heat pipe.

  In Future work mixture of different working fluids can be experimented with, along with different nano-particles. Also there is a scope of optimization for different types of heat pipe working on the refrigerating conditions. Different materials for the construction of the heat pipe will also be studied for their suitability for being used in refrigeration plants.

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