Investigations of Advance Heat Transfer Through Heat Pipe

By

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DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

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Investigations of Advance Heat Transfer Through Heat Pipe

Major Project Report

Submitted in fulfillment of the requirements

For the Degree of

Master of Technology in Mechanical Engineering

(Thermal Engineering)

By

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Undertaking for originality of the work

I, Parashar Vyas, Roll No.11MMET18, give undertaking that the Major Project entitled "**Investigations of Advance Heat Transfer Through Heat Pipe** "submitted by me, towards the fulfillment of requirements for degree of Master technology in Mechanical Engineering(Thermal Engineering) of Nirma university, Ahmedabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

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This is certify that the Major Project entitled "**Investigations of Advance Heat Transfer Through Heat Pipe**" submitted by **Mr Parashar K Vyas** (Roll no :11MMET18), towards the fulfillment of the requirements for Master of Technology in mechanical engineering (Thermal Engineering) of Institute of Technology, Nirma University, Ahmedabad is the record of work carried out by his under our supervision and guidance. In our opinion, the submitted work has reached a level required for being accepted for examination. The result embodied in this major project, to the best of my knowledge, haven't been submitted to any other university or institution for award of any degree.

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Abstract

In present work, heat pipe has been developed & tested considering it's application in waste heat recovery. Ample amount heat is wasted in industrial processes at different temperature is included in the literature. For recovery of these waste heat, presently shell & tube and fin & tubes heat exchangers are in use. To increase the heat recovery efficiently for low temperature gradient, heat pipe can be a alternative solution.

In present study heat pipe was developed and tested at 120^0C evaporator temperature. Parametric study was carried out by using different wall material, length of the pipe, wick material, wick layer and orientation of pipe to evaluate heat flux, system efficiency, thermal resistance and overall heat transfer coefficient. Based on property analysis water, methanol, ethanol and pantene can be used as working fluid. Comparing liquid thermal conductivity, latent heat, specific heat, working temperature range, toxicity and cost the water is most suitable as a working fluid.

The research is based on the fundamental principle of heat pipe technology, which includes the design and experimental results on the vapor, liquid 2-phase flow and heat transfer, heat transfer limits, heat transfer enhancement with heat pipes and the material compatibility. This research concludes the high efficient heat pipe (heat & mass transfer equipment) by theoretical selection and experimental procedure. As per as dimension is concern, no correlations are available in any of the literature to decide dimensions and particular area for waste heat recovery system.

Key Words: waste heat recovery, carbon-steel water heat pipe, waste heat boiler, air preheater, energy conservation, heat generators, compatibility, wick.

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Nomenclature

ΔP_l	:	adiabatic section pressure drop
θ	:	angle (Θ)
A_w	:	area of wick (m^2)
ΔP_c	:	condenser pressure drop
ρ	:	density (kg / m^3)
d_v	:	diameter of vapor core (m)
d	:	diameter of wick wire (m)
l_{eff}	:	effective length (m)
Q	:	energy input (W)
k_{eq}	:	equivalent thermal conductivity (W / K m)
ΔP_v	:	evaporator pressure drop
d_o	:	external diameter of heat pipe (m)
ΔP_g	:	gravitational pressure drop
R	:	heat pipe resistance (m^2K/W)
U	:	heat transfer coefficient (W / m^2K)
d_i	:	internal diameter of heat pipe (m)
kJ/hr	:	kilojoules per hour
$kJ/m2$	:	kilojoules per sq. meter
L_c	:	length of condenser section (m)
L_e	:	length of evaporator section (m)
LPH	:	liter per hour
m	:	mass flow rate (kg / hr)
K	:	permeability (m^2)
σ	:	surface tension (N / m)
C_p	:	specific heat (kJ / kg ^{0}C)
$\triangle T$	:	temperature difference(^{0}C)
K	:	thermal conductivity (W / m^0C)
k_t	:	thermal conductivity of the heat pipe wall (W / K m)
R_{ct}	:	thermal resistance of the heat pipe wall at the condenser (K / W)
R_{et}	:	thermal resistance of the heat pipe wall at evaporator (K / W)
R_{cw}	:	thermal resistance of the heat pipe wick at the condenser (K / W)
R_{ew}	:	thermal resistance of the heat pipe wick at the evaporator (K / W)
μ	:	viscosity (Ns / m^2)
ϵ	:	volume Fraction of solid face (0.314)
T_{wo}	:	water outlet temperature (^{0}C)

Abbreviation

kJ	:	kilo joule
LPH	:	loop heat pipe
mm	:	Millimeter
NCG	:	non condensable gas
THP	:	thermosyphone heat pipe
W	:	watt
WHP	:	wick heat pipe

Chapter 1

Introduction

The transportation of heat plays an important role in many areas of modern technology. For example almost all the electrical devices including solid state electronic components generate heat as a useless by product, which must be removed from the immediate environment and disposed of in some suitable heat sink. Therefore a major research effort has been mounted to and more efficient means of moving heat from one place to another. Heat pipes are devices that can transfer large amount of heat with small temperature differences between evaporator i.e. heat source and condenser i.e. heat sink. This property can be seen better when a heat pipe is compared with aluminum or a copper rod. Assume that it is required to transfer Q=20W of thermal energy over a distance of L=0.5m in a device ¢=1.27cm in diameter. As shown in Figure 1.1, the temperature difference required to transfer this heat will be as below

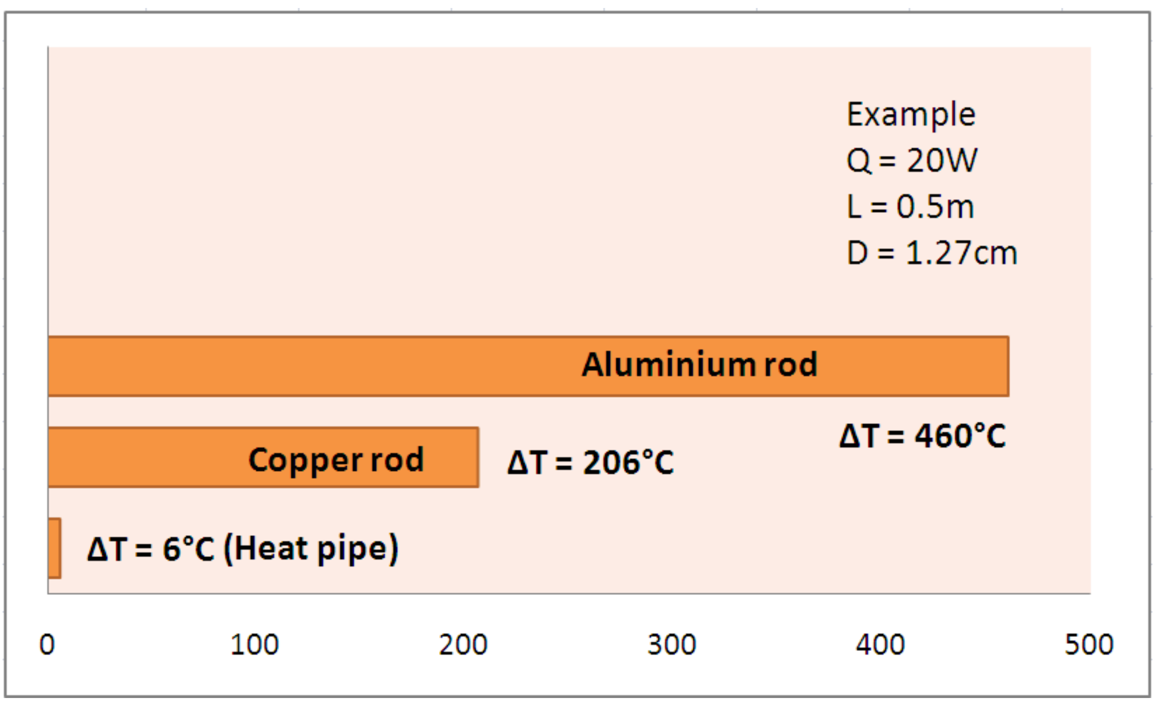


Figure 1.1: Comparison of heat pipe and solid conductors [1]

Solid aluminum rod	=	460 °C
Solid copper rod	=	206 °C
Simple copper-water heat pipe with a screen mesh wick	=	6 °C

Heat pipes can carry higher wattage with less temperature gradient as discussed above, thus they are used to remove heat from these components and carry to the heat sink [1].

Thermodynamic cycle

Thermodynamic cycle of heat pipe & working procedure is shown in Figure 1.2

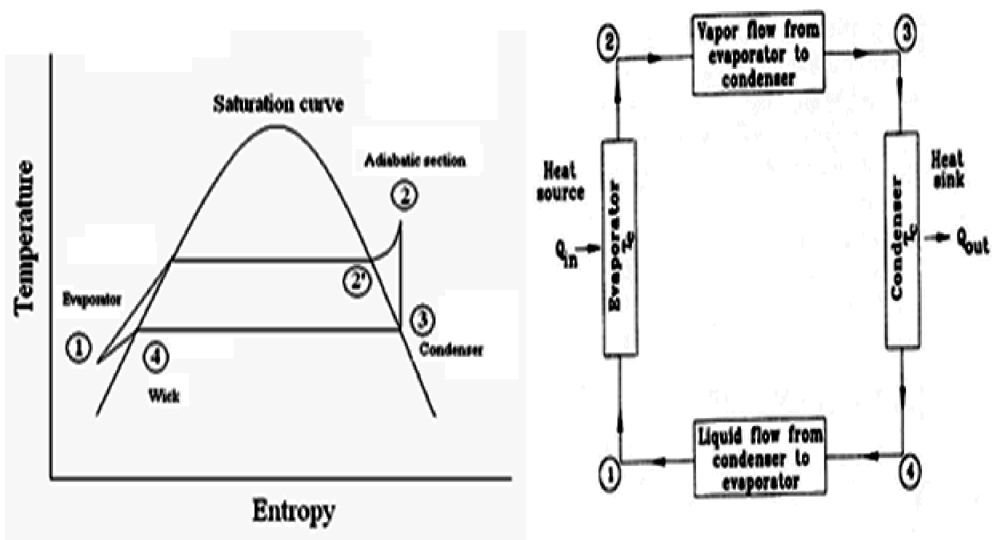


Figure 1.2: Thermodynamic cycle of heat pipe [1]

Details of processes involved are as under:

- 1-2 : Heat applied to evaporator through external sources, vaporizes working fluid to a saturated state (2') or superheated state (2) of vapor.
- 2-3 : Vapor pressure drives vapor through adiabatic section to condenser.
- 3-4 : Vapors condenses, releasing heat to a heat sink.
- 4-1 : Capillary pressure created by menisci in wick pumps condensed fluid into evaporator section.

1.1 Heat pipe overview

A heat pipe is a simple device that can quickly transfer heat from one point to another. They are often referred as a "superconductors" of heat as they possess an extra ordinary heat transfer capacity.

1.1.1 Heat pipe construction and working principle

The simple diagram of heat pipe is shown in Figure 1.3. It can be observed that the heat pipe is a device having three main sections

1. A sealed container (pipe wall and end caps)
2. A wicking structure
3. Small amount of working fluid

The length of the heat pipe is divided into three sections as shown in Figure 1.3.

Evaporator section - The region where the external heat source is connected to the heat pipe.

Adiabatic (transport) section - The region where the heat pipe is externally insulated and no heat transfer to or from the heat pipe, it is mainly a transport section for the vapour and liquid.

Condenser section - The region from where the heat is released to heat sink.

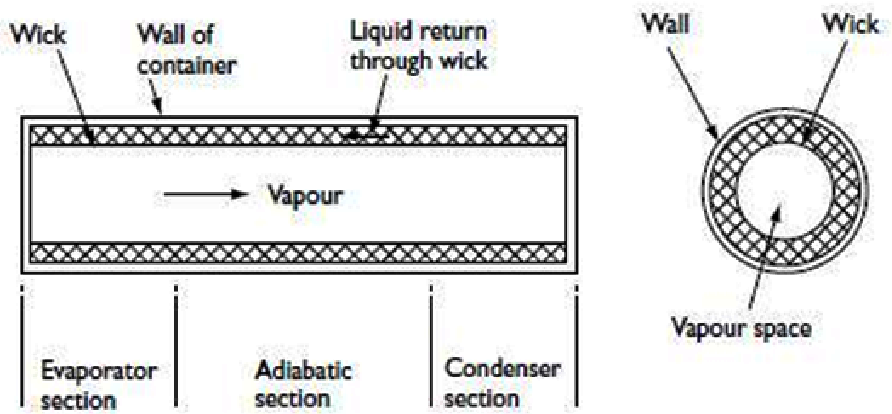


Figure 1.3: Heat pipe constructions [2]

A heat pipe may have multiple heat sources or sinks with or without adiabatic sections depending on a specific applications and design.

Heat applied to the evaporator section is conducted through the pipe wall and wick structure, where it vaporizes the working fluid. The resulting vapour pressure drives the vapour through the adiabatic section to the condenser, where the vapour condenses, releasing its latent heat of vaporization to the provided heat sink. This condensate then returns to the evaporator through the wick by capillary action. Therefore, the heat pipe can continuously transport the latent heat of vaporization from the evaporator to the condenser section. This process will continue as long as there is a sufficient capillary pressure to drive the condensate back to the evaporator. A flowchart representing the cyclic operation of heat pipe is shown in Figure 1.4.

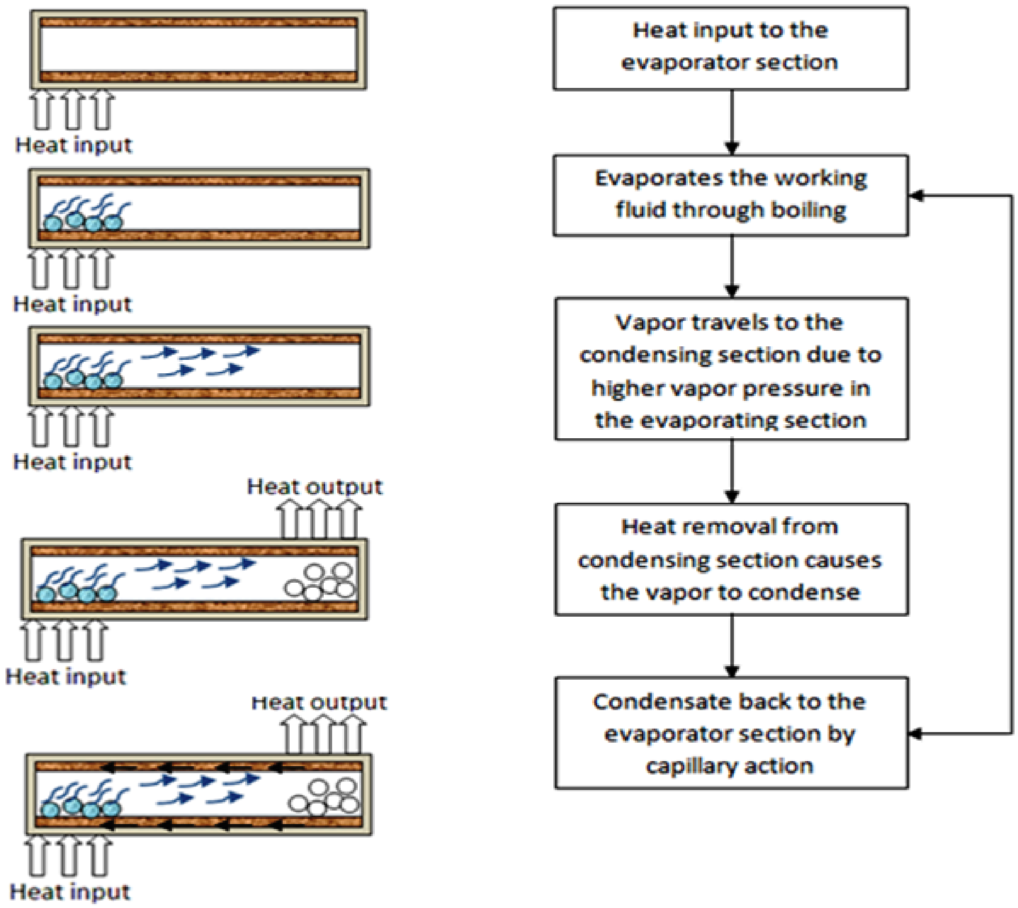


Figure 1.4: Flow chart of heat pipe working [3]

1.1.2 Advantages of heat pipe

- Heat pipes are totally passive heat transfer systems which having no moving parts to wear out. Capillary pumping in the wick is generated by the heat transfer process and requires no other power or moving parts to pump the condensate working fluid. They require no electrical energy to operate hence no power consumption.
- They are very reliable, maintenance free, noise free equipment. Size of the heat pipe is very compact, which makes them the ideal choice for space-constrained applications.
- A heat pipe rated at 10W can carry that heat current with a certain temperature drop ΔT, but ΔT is found to be more or less independent of the length of the heat pipe. Thus, heat pipes are remarkable not only for their very low thermal resistance R (small ΔT) but also for the highly unusual property that R is roughly independent of length. The advantage of using a heat pipe rather than a rod of metal therefore increases with increasing length.
- They offer the design engineer low cost packaging and flexibility because they can be manufactured in a number of shapes and sizes.

- Heat pipes have enormously more heat transfer capability than other methods on a weight and size basis.
- Heat pipe operates satisfactorily in a zero gravity environment.

1.1.3 Disadvantages of heat pipe

- Heat pipes must be tuned to particular cooling conditions. The choice of pipe material, size and coolant all have an effect on the optimal temperatures in which heat pipes work.
- When a heat pipe is heated above a certain temperature, all of the working fluid in the heat pipe will vaporize and condensation process will cease to occur; in such conditions, the heat pipe's thermal conductivity is effectively reduced to heat conduction properties of its solid metal casing. As the most of the heat pipes are constructed with a metal of higher conductivity, an overheated heat pipe will generally continue to conduct heat at around heat $1/80^{\text{th}}$ original conductivity [2]. In addition, below a certain temperature, the working fluid will not undergo phase change and the thermal conductivity will be reduced to that of the metal casing. One of the key criteria for selecting the working fluid is the desired operational temperature range of the application. The lower temperature limit occurs few degrees above the freezing point of the working fluid.
- Most manufacturers cannot fabricate the traditional heat pipes with less than 3mm diameter because of advanced fabrication techniques required [3].

1.1.4 Applications of heat pipe

- Heat pipes are used in a wide range of products like air-conditioners, refrigerators, heat exchangers, transistors, capacitors, etc. Heat pipes are also used in laptops to reduce the working temperature for better efficiency. Their application in the field of cryogenics is very significant, especially in the development of space technology.
- Electronics cooling : smaller size and high performance components cause high heat fluxes and high heat dissipation demands. Used to cool transistors and high density semiconductors.
- Aerospace cool : satellite's electronics components, solar array, as well as shuttle leading edge during reentry, for transferring heat to a radiator over large distances from areas where there are equipment with high heat dissipation, for spreading heat over a structural panel to minimize thermal gradients over it.
- Heat exchangers : power industries use heat pipe heat exchangers as air heaters on boilers.
- Other applications : production tools, medicine, human body temperature control, engines and automotive industry.

Chapter 2

Literature Review

2.1 Types of heat pipe

2.1.1 Classification based upon their conductance

2.1.1.1 Fixed conductance

The fixed conductance heat pipe is a device of very high thermal conductance with no fixed operating temperature. Its temperature rises or falls according to variations in the heat source or heat sink.

2.1.1.2 Variable conductance

In many heat pipe applications a specific operating temperature range is desired along the certain portion of the pipe even though source and sink conditions are changing. In those cases it becomes necessary to actively or passively control the heat pipe so that it maintains the desired operating temperature range. Temperature control is obtained by varying one or several of the conductance that make up the heat pipe's overall thermal conductance [2].

2.1.2 Classification based upon their application and operation

2.1.2.1 Two-phase closed thermosyphon heat pipe

A two-phase closed thermosyphon is a gravity-assisted wick less heat pipe. The condenser section is located above the evaporator so that the condensate is returned by gravity. The performance of the thermosyphon is mainly depends on the volume of the working fluid. The maximum rate of heat transfer increases with the amount of the working fluid up to a certain value.

2.1.2.2 Flat plate heat pipe

A flat plate heat pipe (Figure 2.1) is capillary driven, and has a rectangular shape with a small aspect ratio. Additional wick blocks between the evaporator and condenser aid in condensate return, especially when the condenser is below the evaporator in a gravity field [2].

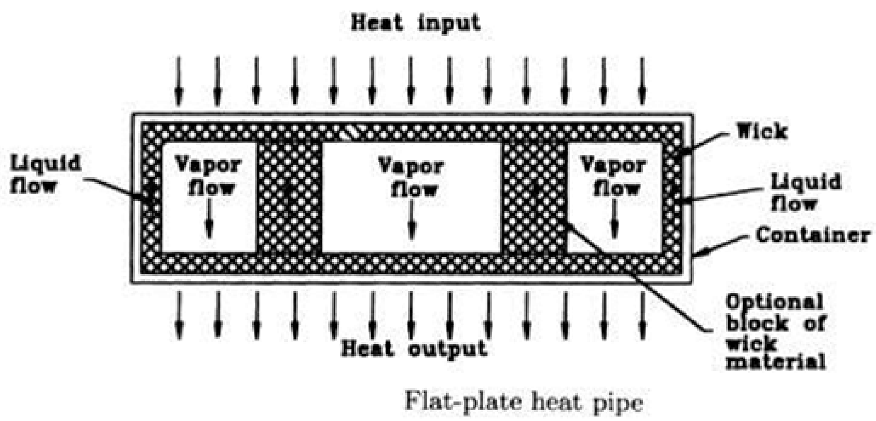


Figure 2.1: Flat plate heat pipe [2]

2.1.2.3 Capillary-driven heat pipe

The capillary-driven heat pipe consists of a sealed container, in which a wick is placed in the inner radius of the pipe wall. The purpose of the wick is to provide a capillary-driven pump for returning the condensate to the evaporator section.

2.1.2.4 Annular heat pipe

The annular heat pipe is similar to the conventional capillary-driven heat pipe except that the cross section of the vapor space is annular instead of circular. This enables the designer to place wick material both on the inside of the outer pipe and on the outside of the inner pipe. In this manner, the surface area for heat input and output and hence capillary limit can be increased significantly without increasing the outer diameter of the heat pipe.

2.1.2.5 Rotating heat pipe

Rotating heat pipes are in two configurations. First, the heat pipe can be in the shape of a circular cylinder with or without an axial taper, which rotates either about its own axis of symmetry or it revolves off-axis. Secondly, the heat pipe can be manufactured in the shape of disk, where two parallel disks are joined together at the outer and inner radius to form the vapor space. Condensate is returning to the evaporator by centrifugal force. The rotating heat pipe is shown in Figure 2.2

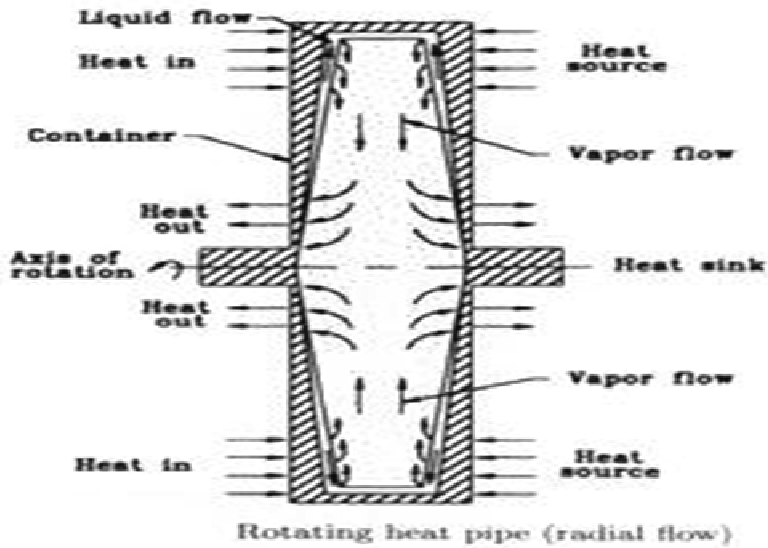


Figure 2.2: Rotating heat pipe [2]

2.1.2.6 Mono-groove heat pipe

The mono-groove heat pipe (Figure 2.3) utilizes an extruded cross section which separates the vapor space from the liquid return path.

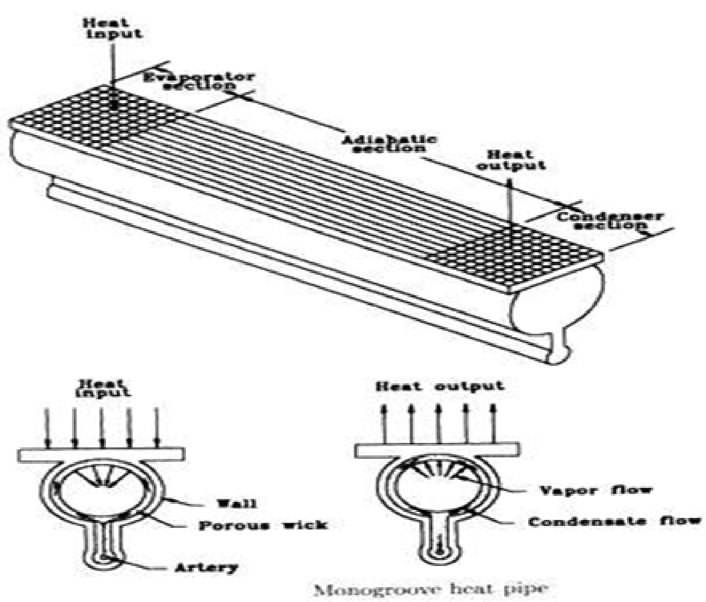


Figure 2.3: Mono-groove heat pipe [2]

2.1.2.7 Leading edge heat pipe

The leading edge heat pipe has been proposed to cool the leading edges of the future hypersonic aircraft and reentry vehicles. The heat pipe will cover the leading edges of the wings and engine nacelles, where the aerodynamic heating is most intense.

2.1.2.8 Micro heat pipes

Micro heat pipes are small heat pipes that are non circular and use angled corners as liquid arteries. The micro grooved heat pipe is shown in Figure 2.3

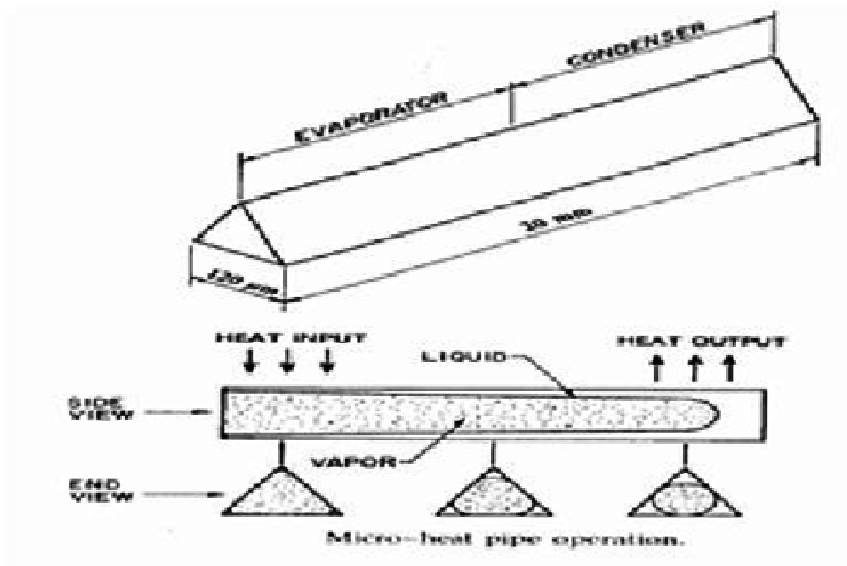


Figure 2.4: Micro heat pipe [2]

2.1.2.9 Variable conductance heat pipe

Variable conductance heat pipe (Figure 2.5) is as the same as the capillary driven heat pipe or the thermosyphon except that a non-condensing gas is introduced into the vapor space. During the operation, this gas is swept down the length of the heat pipe by the working fluid vapor to the condenser section. Since the condensation of the working fluid does not take place where the non-condensing gas is present, a portion of the condenser is blocked from transferring heat to the heat sink. If the heat input to the evaporator section is increased, the vapor gas temperatures both increase, which leads to the compression of the inert gas.

This in turn increases the amount of the condenser surface area available to transfer the heat. This phenomena result in variable conductance heat pipe is being able to maintain nearly constant evaporator temperature regardless of the heat input.

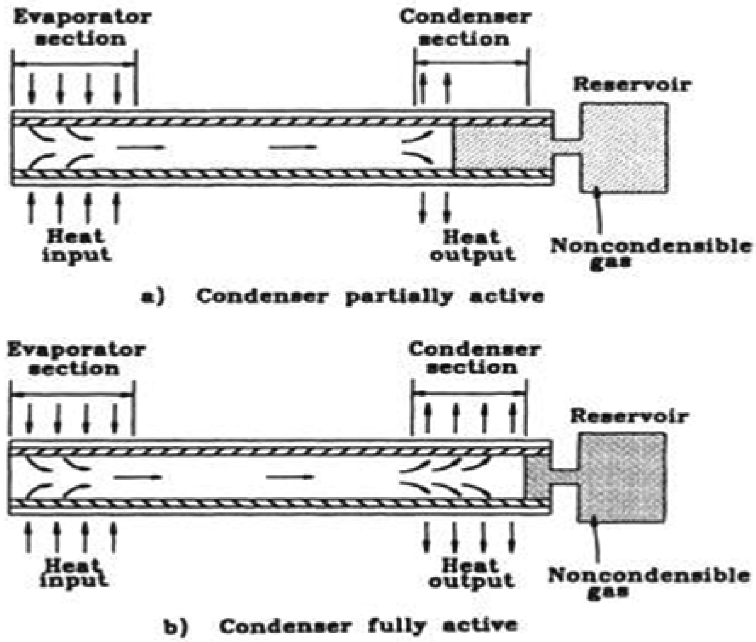


Figure 2.5: Variable conductance heat pipe [2]

2.1.3 Classification based upon their capillary wick structures

Generally the heat pipes can be divided into two categories based upon their wick structures:

- Homogeneous wicked heat pipe
- Composite wicked heat pipes
- Homogeneous wicks have the benefit of being relatively simple to design, manufacture and install.

Composite wicks, however, can significantly increase the capillary limit of the heat pipe, but have the drawback of high manufacturing costs.

2.1.3.1 Homogeneous wicks

Homogeneous wicks are typically comprised of a single material and are distributed uniformly along the axial length of the heat pipe. Homogeneous wicks are widely used because of its reliability, good start-up under load characteristics flexibility of application and lower cost. Some of the commonly used wick structures are shown in Figure2.6 [2][4].

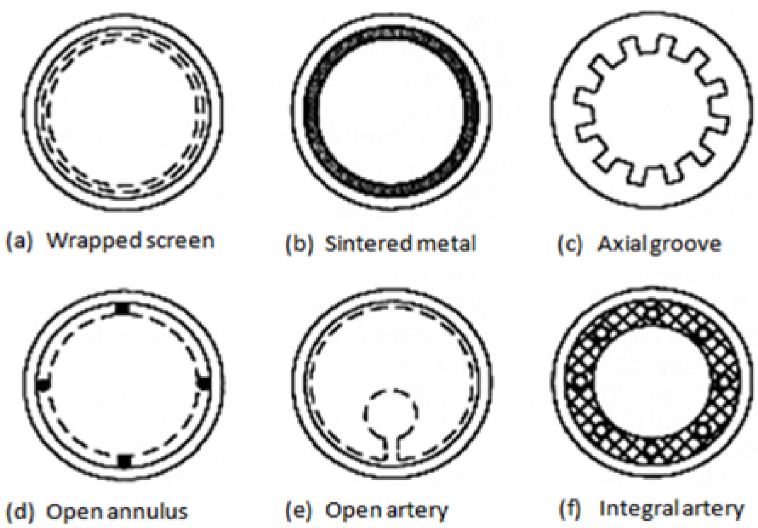


Figure 2.6: Homogeneous wick structures [2]

2.1.3.2 Screen wire mesh

The screen wick is probably simplest and most common type of wick structure, it consists of a metal or cloth fabric which is wrapped around a mandrel and inserted into the heat pipe. After the placement, the mandrel is removed, leaving the wick held by tensioned wrapped screen, in this case of a metal fabric. For a cloth fabric, a spring may be inserted into heat pipe to hold the wick against inside of the pipe wall. The capillary pressure generated by a screen wick is determined by the size of the rectangular pores between the individual threads. The permeability is determined by the number of wraps and looseness of the wraps, which creates annular gaps through which the condensate can flow [4].

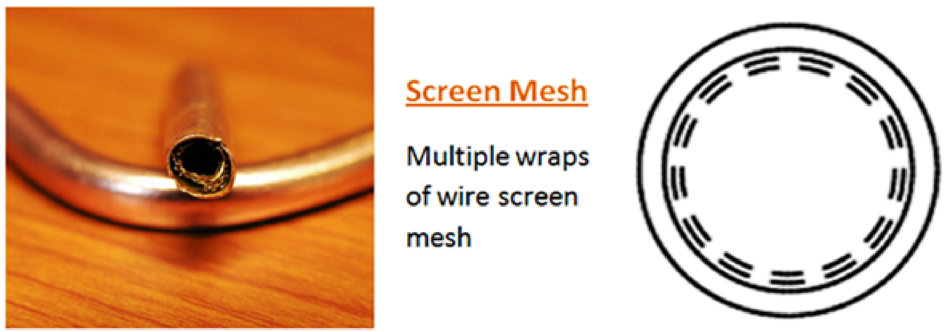


Figure 2.7: Constructional detail of screen wire mesh heat pipe [4]

2.1.3.3 Sintered metal powder

Sintered metal wicks are manufactured by packing tiny metal particles in the powder form between the inner heat pipe wall and a mandrel. This assembly is then heated until the metal spheres are sintered to each other and to the inner wall of the heat pipe. Special materials are used for mandrel so that it can be removed, leaving an open vapor space. This type of wick is obviously more difficult to manufacture compared to the simple screen wick. Since a metal powder is sintered, the effective thermal conductivity is much higher than a comparable screen wick due to poor thermal contact between the screens wraps. These wicks have low permeability due to the small pore sizes and the relatively tortuous path the liquid must follow. These wicks, therefore, are most suitable for applications where the heat transport capability is not too restrictive. The pore size distribution, porosity and permeability are vital parameters that affect fluid transport through the wick and hence, the overall performance of the heat pipe [5]. Porosity has a great influence on the maximum heat transfer rate. An increase in porosity leads to increase in the maximum heat transfer rate. The mean powder diameter is also an important design parameter. Capillary radius of curvature is mainly a function of the mean powder diameter. The heat transfer rate of the heat pipe with a larger powder diameter is indeed higher than that with a smaller one.

Figure 2.8: Constructional detail of sintered metal powder heat pipe [4]

2.1.3.4 Axial groove

Axial groove wicks are formed by the extrusion or broaching of grooves into inner radius of the pipe. Several different types of grooves have been designed like rectangular, triangular, trapezoidal etc. trapezoidal and rectangular grooves are most common grooves used in different heat pipe wicks. The performance of the axial groove is excellent. Since the size of the grooves are large compared to screen or sintered metal wick, the capillary pumping pressure is quite small [4].

Figure 2.9: Constructional detail of axial grooved heat pipe [4]

2.1.4 Classification based upon their working fluid temperature ranges

A first consideration in the identification of a suitable working fluid is the operating vapor temperature range. Each heat pipe application has a particular temperature range in which the heat pipe needs to operate. The working fluid is responsible for the transport of heat through the heat pipe. The choice of working fluid is dictated by several considerations, including operating temperature, latent heat of vaporization, liquid viscosity, toxicity, chemical compatibility with container material, wicking system design and performance requirements. The highest performance from a heat pipe is obtained by utilizing a working fluid that has a high surface tension (σ), a high latent heat (L), and a low liquid viscosity (µl). These fluid properties are contained in the parameter M, Figure of Merit or the Liquid Transport Factor. The working fluid's ability to transport heat can be judged by liquid transport factor or figure of Merit, M. Higher will be the M, better will be the heat transport capability [4].

Heat pipes can be classified according to its working fluid temperature ranges are as below:

2.1.4.1 Cryogenic heat pipes

Cryogenic heat pipe operate between 4 to 200 K. Typical working fluids include helium, argon, oxygen and krypton.

2.1.4.2 Low temperature heat pipes

The low temperature is from 200 to 550 K. Most heat pipe applications fall within this range. Commonly used fluids are ammonia, acetone, the freon compounds, and water.

2.1.4.3 Medium temperature heat pipes

The working fluids in the medium temperature range heat pipe i.e. 550 to 750K, are Mercury and sulfur are working fluids.

2.1.4.4 High temperature heat pipes

The high temperature range heat pipe is considering as 750 K and above. Sodium, lithium, silver and a sodium-potassium compounds are often used in the high temperature range.

2.2 Heat transfer limitations

There are various parameters that put limitations and constraints on the steady and transient performance of heat pipes. The rate of heat transport through a heat pipe is subjected to a number of operating limits. Physical phenomena that might limit heat transport in heat pipes are due to capillary, sonic, entrainment, boiling, frozen start up, continuum effect, vapor pressure and condenser effects.

The heat transfer limitations can be of any of the limitations depending upon the size and shape of the pipe, working fluid, wick structure and operating temperature. The lowest limit among the all constraints defines the maximum heat transport limitation of a heat pipe at a given temperature [6].

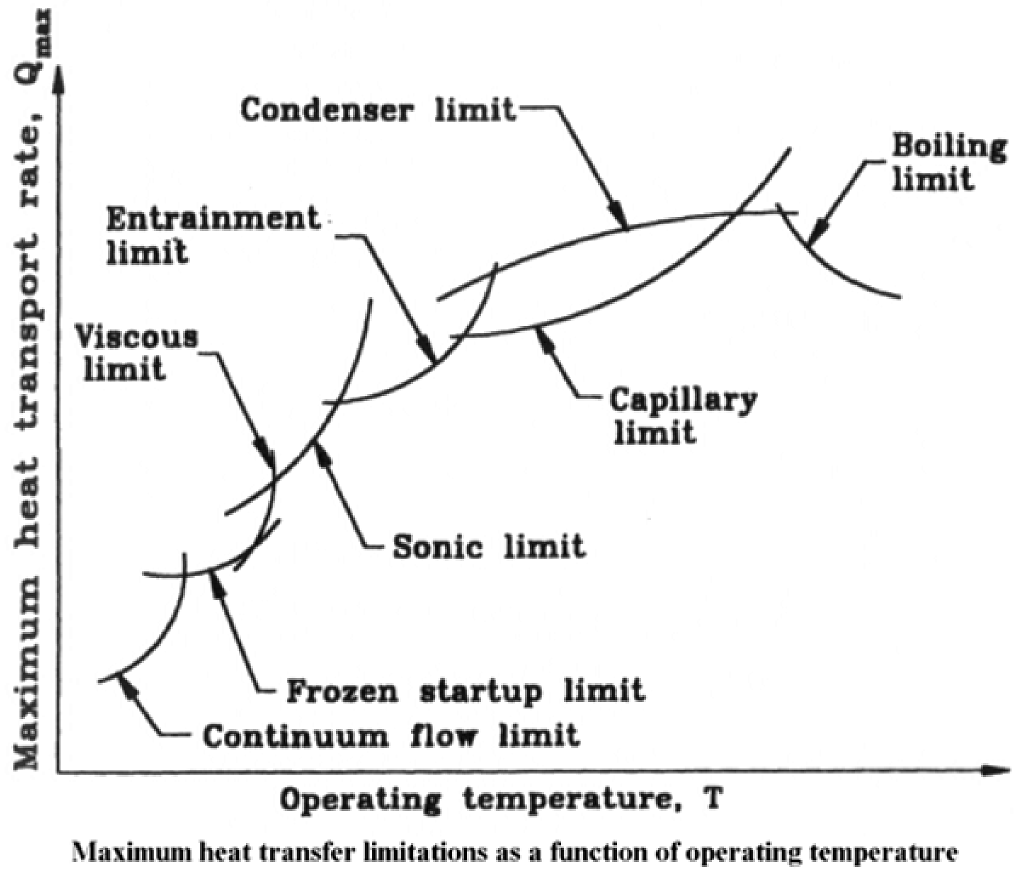


Figure 2.10: Heat transfer limitations [6]

2.2.1 Capillary limit

In order to maintain the pressure difference between liquid and vapor, the interface separating them must be curved. Any curved liquid-vapor interface creates a pressure difference. When the required inter-facial pressure exceeds the capillary pressure that the wick can sustain, the pumping rate is no longer sufficient to supply enough liquid to the evaporation sites. Consequently, more liquid is evaporated than replenished and the local dry-out of the wick occurs. The ability of a particular capillary structure to provide the circulation for a given working fluid is limited. The limit is usually called capillary limitations or hydrodynamic limitation. The capillary limit is the most commonly encountered limitation in the operation of low temperature heat pipes.

It occurs when the capillary pumping rate is not sufficient to provide enough liquid to the evaporator section. This is due to the fact that the sum of liquid and vapor pressure drops exceed the maximum capillary pressure that wick can sustain. The maximum capillary pressure for a given wick structure is depends upon the physical properties of the wick and the working fluid. Any attempt to increase the heat transfer above the capillary limit will cause dry-out in the evaporator section where a sudden increase in wall temperature along the evaporator section takes place [4, 6]. The necessary condition to prevent dry-out from the evaporator section is given by,

$$\Delta P_c \geq \Delta P_l + \Delta P_v + \Delta P_g \qquad 2.1$$

where,

ΔP_c: Condenser pressure drop

ΔP_l: Adiabatic section pressure drop

ΔP_v: Evaporator pressure drop

ΔP_g: Gravitational pressure drop

2.2.2 Sonic limit

The evaporator and condenser sections of a heat pipe represent a vapor flow channel with mass addition and extraction due to evaporation and condensation. The vapor flow velocity increases along the evaporator and reached a maximum at the end of the evaporator section. The vapor velocity exceeds the local speed of sound. This may choked the flow is called as sonic limit. This limit generally occurs during heat pipe start up. Vapor flow is choked when heat pipe operates near the freezing point where vapor pressures and densities are very low. It is associated with high vapor velocities with lower density fluid. When sonic limit exceeds, it does not represent a serious failure as does the capillary limit. The rate of heat transfer will not increase by decreasing the condenser temperature under the choked conditions. Therefore, when the sonic limit is reached further increase in heat transfer rate can be realized only when the evaporator temperature increases. Operation of heat pipes with a heat rate close to or at sonic limit results in a significant axial temperature drop along the heat pipe [6].

2.2.3 Boiling limit

If the radial heat flux in the evaporator section becomes too high, the liquid in the evaporator wick boils and the wall temperature becomes excessively high. The vapor bubbles that form in the wick prevent the liquid from wetting the pipe wall, which causes hot spots. If the boiling is severe it dries out the wick in the evaporator which is defined as boiling limit. High local heat fluxes that cause the nucleate boiling and interrupts the liquid flow in evaporator. However, under a low or moderate radial heat flux, low intensity stable boiling is possible without causing dry-out condition. It should be noted that the boiling limitation is a radial heat flux limitation rather than axial heat flux limitation. This limit is also related to evaporator surface area [6].

2.2.4 Entrainment limit

A shear force exists at the liquid-vapor interface since the vapor and liquid move in opposite directions. At high relative velocities, droplets of liquid can be torn from wick surface and entrained into the vapour flowing toward the condenser section. It will affect flow of liquid towards the evaporator. High velocity vapor flow strips and entrain liquid droplets thereby impeding the liquid flow to evaporator. It occurs at high loads and near freeze point. If the entrainment becomes too high, the evaporators will dry-out. The heat transfer rate at which this occurs is called the entrainment limit. Entrainment can be detected by the sounds made by the droplets striking the condenser end of the heat pipe. The entrainment limit is often associated with low or moderate temperature heat pipe with small diameter, or high temperature heat pipes when the heat input at the evaporator is high [6].

2.2.5 Vapor pressure limit or viscous limit

At low operating temperatures, viscous forces may dominant for the vapor flow down the heat pipe. For a long heat pipe, the vapor pressure at the condenser section may reduce very much. The heat transport of the heat pipe may be limited under this condition. The vapor pressure limit (viscous limit) is encountered when a heat pipe operates at temperature below its normal operating range such as during start up from frozen state. In this case, the vapor pressure is very small; with the condenser end cap pressure is nearly zero [6].

2.2.6 Frozen start-up limit

During the start up process from the frozen state, the active length of the heat pipe is less than the total length and the distance to which the liquid has to travel in the wick is less than that required for steady state operation. Therefore, the capillary limit will usually not occur during the start up process if the heat input is not very high and is not applied suddenly. However, for heat pipes with initially frozen working fluid, if the melting temperature working fluid and heat capacities of the heat pipe container and wick are very high, and latent heat of vaporization and cross sectional area of the wick are small, a frozen start up limit may occur under this condition due to freezing out of vapor from the evaporation zone to the adiabatic or condenser zone.

Under the normal working conditions, the working fluid in the wick structure of a heat pipe is in the liquid state and flows from the condenser section to the evaporator section due to capillary pumping action in the heat pipe. However, if a heat pipe is started from the ambient temperature, the working fluid in the wick structure may be in the solid state, depending on the specific working fluid used. For low or medium temperature heat pipes, the working fluid is usually in the liquid state at the ambient temperature. For high temperature heat pipes on the other hand, the working fluid in the heat pipe is usually in the solid state at ambient temperature, due to the high melting temperature of the working fluid. Therefore, frozen start up is a routine occurrence during the high temperature heat pipe operation [4].

2.2.7 Continuum flow limit

For heat pipe with very low operating temperatures, especially when the dimension of the heat pipe is very small such as micro heat pipe, the vapor flow in the heat pipe may be in the free molecular condition or rarefied condition. The heat transport capability under this condition is limited, which is called as vapor continuum limit. The vapor flow in the heat pipe core is usually in the continuum state for conventional heat pipes. However, as the size of the heat pipe decreases, the vapor in the heat pipe may lose its continuum characteristics. The heat transport capability of a heat pipe operating under non continuum vapor flow conditions is very limited and the large temperature gradient exists along the heat pipe length. As a result, the heat pipe will lose its advantages as an effective heat transfer device. This condition must be checked for the miniature or micro heat pipes, whose dimensions may be extremely small. This continuum criterion is usually expressed in terms of the Knudson number,

$\mathrm{Kn} = \lambda/\mathrm{D}$ $\mathrm{Kn} \leq 0.01$ for continuum vapor flow

$\mathrm{Kn} > 0.01$ for Rarefied or Free Molecular flow

Where, λ is the mean free path of the vapor molecule and D is the minimum dimension of the vapor flow passage [4].

2.3 Waste heat recovery

Waste heat is heat, which is generated in a process by way of fuel combustion or chemical reaction, and then "dumped" into the environment even though it could still be reused for some useful and economic purpose. The essential quality of heat is not the amount but rather its "value". The strategy of how to recover this heat depends in part on the temperature of the waste heat gases and the economics involved.

Large quantity of hot flue gases is generated from boilers, kilns, ovens and furnaces. If some of this waste heat could be recovered, a considerable amount of primary fuel could be saved. The energy lost in waste gases cannot be fully recovered.

2.3.1 Classification and application

In considering the potential for heat recovery, it is useful to note all the possibilities, and grade the waste heat in terms of potential value as shown in the following Table 2.1.

Table 2.1: Waste source and quality[7]

Sr. No.	Source	Quality
1	Heat in flue gases	The higher the temperature, the greater the potential value for heat recovery
2	Heat in vapour streams	As above but when condensed latent heat also recoverable
3	Convective and radiant heat lost from exterior of equipment	Low grade : if collected may be used for space heating or air preheats
4	Heat losses in cooling water	Low grade : useful gains if heat is exchanged with incoming fresh water
5	Heat losses in providing chilled water or in the disposal of chilled water	a) High grade : if it can be utilized to reduce demand for refrigeration b) Low grade : if refrigeration unit used as a form of heat pump
6	Heat stored in products leaving the process	Quality depends upon temperature
7	Heat in gaseous and liquid effluents leaving process	Poor if heavily contaminated and thus requiring alloy heat exchanger

2.3.1.1 High temperature heat recovery

Table 2.2 gives temperatures of waste gases from industrial process equipment in the high temperature range. All of these results from direct fuel fired processes.

Table 2.2: Typical waste heat temperature at high temperature range from various sources [7]

Types of Device	Temperature, (°C)
Nickel refining furnace	1370 - 1650
Aluminum refining furnace	650 - 760
Zinc refining furnace	760 - 1100
Copper refining furnace	760 - 815
Steel heating furnaces	925 - 1050
Copper furnace	900 - 1100
Open hearth furnace	650 - 700
Cement kiln (Dry process)	620 - 730
Glass melting furnace	1000 - 1550
Hydrogen plants	650 - 1000
Solid waste incinerators	650 - 1000
Fume incinerators	650 - 1450

2.3.1.2 Medium temperature heat recovery

Table 2.3 gives the temperatures of waste gases from process equipment in the medium temperature range. Most of the waste heat in this temperature range comes from the exhaust of directly fired process units.

Table 2.3: Typical waste heat temperature at medium temperature range from various sources [7]

Type of Device	Temperature, (°C)
Steam boiler exhausts	230 - 480
Gas turbine exhausts	370 - 540
Reciprocating engine exhausts	315 - 600
Reciprocating engine exhausts (turbo charged)	230 - 370
Heat treating furnaces	425 - 650
Drying and baking ovens	230 - 600
Catalytic crackers	230 - 600
Annealing furnace cooling systems	425 - 650

2.3.1.3 Low temperature heat recovery

Table 2.4 lists some heat sources in the low temperature range. In this range it is usually not practical to extract work from the source, though steam production may not be completely excluded if there is a need for low-pressure steam. Low temperature waste heat may be useful in a supplementary way for preheating purposes.

Table 2.4: Typical waste heat temperature at low temperature range from various sources [7]

Source	Temperature, (^{o}C)
Process steam condensate	55 - 88
Cooling water from: Furnace doors	32 - 55
Bearings	32 - 88
Welding machines	32 - 88
Injection molding machines	32 - 88
Annealing furnaces	66 - 230
Forming dies	27 - 88
Air compressors	27 - 50
Pumps	27 - 88
Internal combustion engines	66 - 120
Air conditioning and refrigeration condensers	32 - 43
Liquid still condensers	32 - 88
Drying, baking and curing ovens	93 - 230
Hot processed liquids	32 - 232
Hot processed solids	93 - 232

2.4 Benefits of waste heat recovery

Benefits of 'Waste Heat Recovery' can be broadly classified in two categories:

2.4.1 Direct benefits

Recovery of waste heat has a direct effect on the efficiency of the process. This is reflected by reduction in the utility consumption & costs, and process cost [7].

2.4.2 Indirect benefits:

2.4.2.1 Reduction in pollution

A number of toxic combustible wastes such as carbon monoxide gas, sour gas, carbon black off gases, oil sludge, Acrylonitrile and other plastic chemicals etc, releasing to atmosphere if/when burnt in the incinerators serves dual purpose i.e. recovers heat and reduces the environmental pollution levels [7].

2.4.2.2 Reduction in equipment sizes

Waste heat recovery reduces the fuel consumption, which leads to reduction in the flue gas produced. This results in reduction in equipment sizes of all flue gas handling equipments such as fans, stacks, ducts, burners, etc.

2.4.2.3 Reduction in auxiliary energy consumption

Reduction in equipment sizes gives additional benefits in the form of reduction in auxiliary energy consumption like electricity for fans, pumps etc.

2.5 Literature review

Chaudhry et al. [8] evaluates current heat pipe systems for heat recovery and renewable applications utility. Basic features and limitations are outlined and theoretical comparisons are drawn with respect to the operating temperature profiles for the reviewed industrial systems. Working fluids are compared on the basis of the figure of merit for the range of temperatures. The review established that standard tubular heat pipe systems present the largest operating temperature range in comparison to other systems and therefore offer viable potential for optimization and integration into renewable energy systems. The investigation revealed that heat pipes incorporated with sorption phenomenon display greater heat transfer capacity and tubular heat pipes have the highest working range on average with the maximum operating temperature from all reviewed systems being 453 K for the tubular heat pipe arrangement respectively.The study's conclusions are based on the research of various industrial products utilizing the heat pipe systems for their operations. Imperative factors including the figure of merit were calculated and compared for various suitable heat pipe working fluids. The findings revealed that water displayed the highest. Average Merit Number in comparison to ammonia and acetone for the operating temperature range of 293 – 393 K.

Table 2
Merit No. for various working fluid candidates at operating temperatures.

Medium	Melting point (K)	Boiling point (K)	Useful range (K)	Merit No. at operating temp. (293 K)	Merit No. at operating temp. (313 K)	Merit No. at operating temp. (333 K)	Merit No. at operating temp. (353 K)	Merit No. at operating temp. (373 K)	Merit No. at operating temp. (393 K)
Heptane	183.15	371.15	273–423	1.16E+10	1.25E+10	1.24E+10	1.24E+10	1.19E+10	1.10E+10
Water	273.15	373.15	303–473	1.78E+11	2.55E+11	3.27E+11	3.90E+11	4.55E+11	4.97E+11
Ammonia	195.15	240.15	213–373	7.02E+10	5.85E+10	4.50E+10	2.30E+10	1.45E+10	3.43E+09
Pentane	140.15	301.15	253–393	1.49E+10	1.48E+10	1.35E+10	1.22E+10	1.03E+10	7.76E+09
Acetone	178.15	330.15	273–393	3.20E+10	3.24E+10	3.17E+10	3.00E+10	2.57E+10	1.31E+10

Figure 2.11: Figure of merit[8]

Ahmadzadehtalatapeh [9] has focused on the energy saving and dehumidification enhancement aspects of horizontal heat pipe heat exchangers. Both configurations, i.e., vertical (thermosyphon) and horizontal configuration, can be implemented as an efficient energy recovery unit in air conditioning systems to remove heat or coolness and dehumidification purposes, but investigations regarding the application of horizontal configuration HPHXs for dehumidification enhancement and energy saving purposes in air conditioning systems are limited and most of the studies have been carried out in normal climatological conditions.

Yang et al. [10] summarized the recent developments of lightweight, high performance heat pipes. Various methods or approaches to achieve the requirements of lightweight and high performance are introduced. The applications of lightweight materials can help reduce by up to 80% the weight of conventional copper heat pipes, however the lightweight material often has problems of corrosion. Although improving the design of wick structures and changing the size of conventional heat pipe assemblies can help to reduce weight and achieve high heat flux, there are still some limitations to the applications of lightweight materials such as magnesium due to its incompatibility with some working fluids. An alternative direct method to reduce weight is to improve the wick structure of the heat pipes. Using light material (fiber) to make a mixed wick structure, such as mixed mesh and sintering, can help to increase the heat flux in boiling process and also reduce the pressure drop inside the heat pipe which leads to a much lower temperature change in the adiabatic section. However it doesn't give much of a visible weight reduction. Due to the low permeability and high effective thermal resistance of fiber wicks, they must be combined with other wicks to make prototypes. They developed heat pipe cooling application by introducing fins on outer side of cooling heat pipe to increase contact surface area with the air.

Jingtao and Zhao [11] presented an advanced CLHP operating at liquid-nitrogen temperature range. An improved condenser structure is introduced to the CLHP, which greatly reduces the flow resistance and increases the cooling capability of the condenser. Many experiments have been carried out on the CLHP prototype for performance test, and one set of the experimental results with a 3.2 MPa fill pressure at room temperature has presented. It was shown that the advanced CLHP prototype can be operated reliably with a high heat transfer capacity up to 41W and a limited temperature difference of 6 K and across a 0.48 m transport distance. The CLHP operates unsteadily with low heat loads because of the initial thermal capacitance of the components, it is able to operate smoothly when the heat load is large enough to drive the liquid to cool down the evaporation section sufficiently.

Shwin and Huan [13] presented visualization of the evaporation/boiling process and thermal

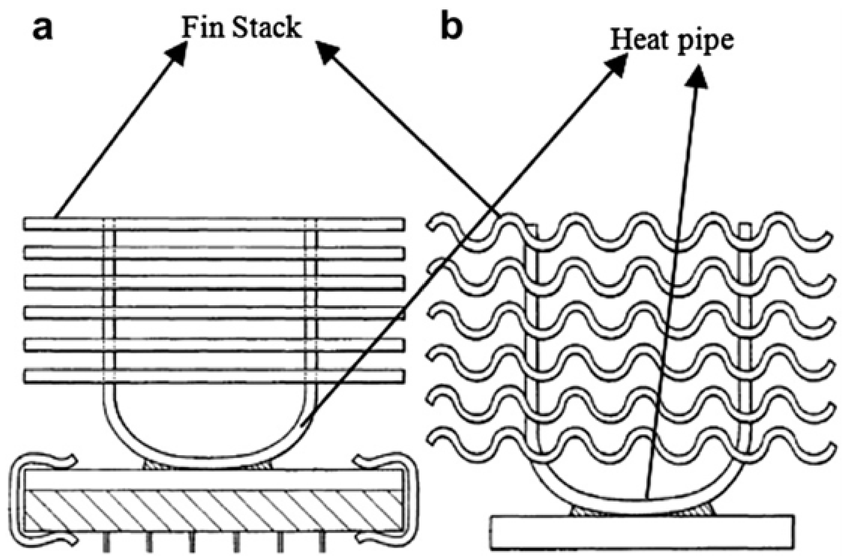


Figure 2.12: Fin heat pipe for cooling application[10]

measurements of operating horizontal transparent heat pipes. The heat pipes consisted of a two-layered copper mesh wick consisting of 100 and/or 200 mesh screens, a glass tube and water as the working fluid. Experimental results indicated that nucleate boiling was prompted for a wick having a fine 200-mesh bottom layer. When the fluid charge approximately equaled the pore volume in the wick, the water–vapor interface receded into more curved menisci with increasing heat load Q. Thus, larger capillary forces and evaporation areas were attained to meet the increasing need of liquid supply and evaporation rate at the water–vapor interface. The fine mesh wick prompted the onset of nucleate boiling by providing more nucleation sites and consequently reduced the evaporation temperature. With a smaller than saturation charge and a wick having a fine mesh bottom layer, partial dry-out was observed in the evaporator and the heating surface temperatures became excessively high. The heat transfer limit was the capillary limit.

Jung [14] described the design, modeling, and test of a heat sink with embedded L-shaped heat pipes and plate fins. This type of heat sink is particularly well suited for cooling electronic components such as microprocessors using forced convection. The mathematical model includes all major components from the thermal interface through the heat pipes and fins. It was augmented with measured values for the heat pipe thermal resistance. A Windows-based computer program uses an iterative superposition method to predict the thermal performance. Thermal performance testing shows that a representative heat sink with six heat pipes will carry 160W and has reached a minimum thermal resistance of 0.22 ^{0}C/W. The computer software predicted a thermal resistance of 0.21 ^{0}C/W, which was within 5% of the measured value. the superposition analytical method for obtaining the thermal performance of the heat sink with embedded L-type heat pipes is robustly established.

Fang and Chen [15] presented the energy balance of heat storage device the dynamic charging process model of heat storage device with heat pipe. Paraffin was taken as phase change material (PCM) and water was used as heat transfer fluid (HTF). R134a was taken as working medium of heat pipe. The temperature of PCM, heat storage capacity and heat storage rate are simulated. The effects of inlet temperature of heat pipe medium and initial temperature of the PCM on the thickness and temperature of the PCM, outlet temperature of heat pipe medium, total heat storage capacity and heat storage rate of heat storage device was discussed. The results indicated that total heat storage capacity and heat storage rate increase when the inlet temperature of heat pipe medium increases or the initial temperature of heat storage material decreases. The heat storage capacity is larger at the PCM initial temperature of 25 ^{0}C than those at the PCM initial temperatures of 30 ^{0}C and 35 ^{0}C. The heat storage rate decreases gradually in sensible heat storage phase of solid PCM, keeps constant during latent heat storage process of melting PCM and reduces quickly in sensible heat storage phase of liquid PCM. The heat storage rate is large when initial temperature of the PCM is low.

Tang et.al [16] presented one-dimensional steady-state model for determining the upper and lower operating boundaries of the initial filling ratio of the working fluid, as a function of the separate heat pipe geometry, vapor temperature of working fluid and power throughput, combined with two-phase heat exchange characteristics and distribution of the liquid film velocity along with the liquid film thickness direction. A parametric analysis is performed to investigate the effects of the length of the evaporator, vapor temperature, and power throughput on the critical values of the upper and lower boundaries. Simulation results show that the length of the evaporator makes almost no influence on the upper boundary, but great effect on the lower boundary. An increase of the vapor temperature leads to the easier arriving of the lower boundary. Moreover, operation ranges of the separate heat pipe vary with the working fluids. Water and methanol were used separately. An experiment was implemented to validate the simulated results.

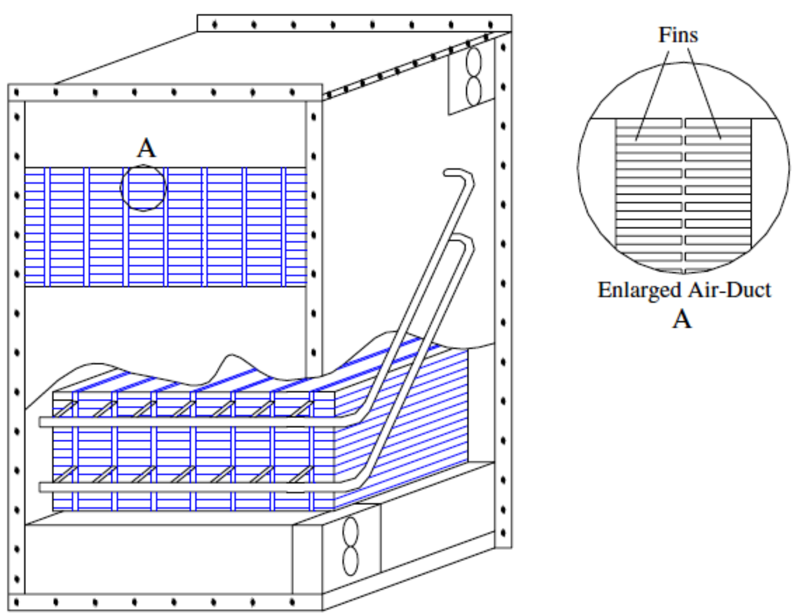


Fig. 2. Configuration of separate looped heat pipe facility.

Figure 2.13: Configuration of separate looped heat pipe facility[16]

The numerical predictions compare favorably with experimental results. Increasing the hydraulic diameter should significantly raise the upper critical values of initial filling ratio and slightly extend the lower boundary. The type of working fluid and the vapor temperature can influence the critical liquid film thickness. The liquid film thickness would became thinner along with increasing the vapor temperature and access to the lower critical value of initial filling ratio.

Kempers and Ching [18] have been performed experimental study to determine the effect of the number of mesh layers and amount of working fluid on the heat transfer performance of copper–water heat pipes with screen mesh wicks. It was found that the effective thermal resistance decreases with an increase in heat flux, and approaches an approximately constant value at higher heat fluxes. This non-linearity in the thermal resistance is larger for wicks with fewer mesh layers. There is a small increase in thermal resistance of the heat pipe when the thickness of the wick is increased, but this is significantly smaller than that predicted by models based on conduction heat transfer across the wick. For all orientations, the maximum heat transfer through the heat pipe increased as the number of mesh layers of the wick was increased, as expected. The heat pipes with amounts of working fluid close to that required to fully saturate the wick performed similarly. For the heat pipes with the smaller number of mesh layers, the effective thermal resistance was non-linear, especially at low heat fluxes. The non-linearity is more significant as the number of layers is decreased. The thermal resistance decreases significantly with the heat flux, and then approaches a constant value.

Jocelyn Bonjour et.al [19] presented an experimental study of a flat plate heat pipe (FPHP). Temperature fields in the FPHP was measured for different filling ratios, heat fluxes and vapour space thicknesses. The system was hermetically sealed with a transparent plate for meniscus curvature radius observations by confocal microscopy. Experimental results show that the liquid distribution in the FPHP and thus its thermal performance depends strongly on both the filling ratio and the vapour space thickness. A small vapour space thickness induces liquid retention and thus reduces the thermal resistance of the system. The vapour space thickness influences the level of the meniscus curvature radii in the grooves and hence reduces the maximum capillary pressure. As a result, it has to be carefully optimized to improve the performance of the FPHP. In all the cases, the optimum filling was in the range one to two times the total volume of the grooves. Experimental results show that the vapour space thickness and the heat flux have important consequences on the thermal performances of a FPHP. Two different liquid distributions in a FPHP can be observed, depending on the size of the vapour space.

Randeep and Aliakbar [20] presented experimentally investigate the effect of non-condensable gases (NCGs) on the thermal performance of the miniature loop heat pipe (mLHP). Copper mLHP with the flat disk shaped evaporator, 30 mm diameter and 10 mm thick, and fin and tube type condenser, 50 mm length and 10 mm height, located at a distance of 150 mm was used in the study. The device which was designed for the thermal control of computer microprocessor was capable of transferring maximum heat load of 70 W while maintaining Evaporator temperature below 100 ^{0}C limit for electronic equipments. Water was used as the heat transfer fluid inside the mLHP. All the tests were conducted with the evaporator and condenser at the same horizontal level. Simple methods were devised to detect and purge the generated NCG out of the loop heat pipe without disassembling the system. Experiments

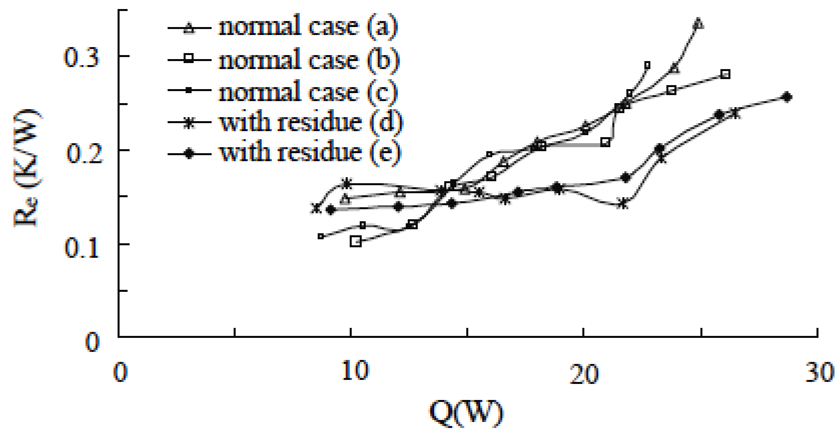


Fig. 11. Evaporator resistances versus heat load *Q* for some representative cases.

Figure 2.14: Evaporator resistance[22]

conducted to classify the trends in the NCG production and storage revealed that majority of the gas is generated in the first few thermal runs and is accumulated in the compensation chamber. an outcomes of the research work, it can be concluded that mLHPs are more tolerable to the NCGs than conventional heat pipes due to the presence of compensation chamber that can accumulate most of the released gas without major performance degradation. The net effect of the generated NCG was to produce an overall rise in the mLHP steady-state operating temperature due to the accumulation of gas inside the compensation chamber that reduces the liquid absorption area of the wick and produces an overall increase in the loop pressure. It was observed that the performance degrading effect of the NCG was more pronounced at low heat loads.

Shwin and Chung [22] described experimentally studied the evaporation characteristics in a groove-wicked flat-plate heat pipe charged with water. The parallel, U-shaped grooves had a width of 0.25 mm and a depth of 0.16 mm. Uniform heating was applied to the copper base plate near one end, and a cooling water jacket was connected at the other end. The evaporator resistance was calculated based on the difference of the plate temperature and the vapor temperature respectively under and above the center of the heated zone. With stepwise increase of heat load, the behavior of the working fluid in the grooves was visualized and the evaporator resistances were measured. Above a certain heat load, longitudinal liquid recession with a steep-sloped liquid front could be visualized. Behind the short liquid front is the accommodation region where the meniscus appeared to anchor on the top corners of the groove walls. Under a thermally stable situation, independent longitudinal oscillations of the liquid front existed in different grooves, forming a constantly varying zigzag front line. With increasing heat load, the liquid fronts gradually left the heated zone, accompanied by increasing plate temperatures. Under thermally stable operation, independent longitudinal oscillations existed in most grooves, exhibiting a constantly varying zigzag front line.

Hassam Nasarullah Chaudhry [8] has reviewed that El-Baky and mohamad investigated the overall effectiveness of utilizing heat pipe heat exchangers for heat recovery through external air conditioning systems in buildings in order to reduce the cooling load. A mathematical model was develop based on the experimental setup which included 25 copper tubes with evaporator and condenser section R-11 was used as a working fluid at a saturation temperature. The finding of the study indicated that effectiveness and heat transfer rates are increase in fresh air inlet temperature. Noie-Bagham and Majideian carried out work on the design and build of heat pipe arrangement to be installed in a heat pipe heat exchanger for the purpose of heat recovery in hospital and laboratory buildings where high air change is a primary equipment. Hussain carried out test work on the comparison of three cross sectional geometrics of wickless heat pipe with varying fill ratios in order to understand the impact of its performance on flat plate solar collector. This review established that standard tubular heat pipe system present the largest operating temperature range in comparison to other system and therefore offer viable potential for optimization and integration into renewable energy systems.

Hagens, Ganzevles and C.W.M. van der [23] presented measurement and predictions of a heat pipe equipped heat exchanger with two filling ratio of R-134a, 19% and 59% the length of the heat pipe is long 1.5m as compared to its diameter The airflow rate varied from 0.4 to 2 kg/s. The temperature at the evaporator side of the heat varied from 40 °C to 70 °C And the condenser part from 20 °C to 50 °C. The measured performance of the heat pipe been compared with predictions of two pool boiling models and two film-wise condensation model. This study demonstrate that a heat pipe equipped heat exchanger is good alternative for air-air exchangers in process conditions when air water cooling is impossible typically in warmer countries. A heat pipe equipped heat exchanger can replace a water cooled heat exchanger without loss of performance. This paper demonstrate that it is possible to apply heat pipe based cooling equipment in practical conditions of warmer countries.

Rosenfeld and North [24] indicated that a bidispread wick takes advantage of the highly effective heat transfer intrinsic to liquid film evaporation in heat pipe operation. North proposed that the loop heat pipe evaporators with bidispread wick are effective to prevent a vapour blanket of the wick and then work at very high heat flux densities. The statistical analysis indicated that the better parameter of the bi-porous wick tended to the lower level of partial size of pore former and the high level of pore former content. At the sink temperature of 10 °C and the allowable evaporator temperature of 85 °C the max heat transport capability is equal to 570W for the better bi-porous wick and 350W for the mono porous wick and the minimum value of the total thermal resistance is 0.10°C/W and 0.22 °C/W respectively. Feng Yang et al. presented the feasibility of using heat pipe heat exchangers for heating applying automotive exhaust gas is studied and the calculation method is developed.

Parameters of a single heat pipe are shown in below Table 2.5.

Table 2.5: Dimensions and results of experimented heat pipe

Material	steel
Working fluid	water
Outer diameter (mm)	20
Thickness (mm)	1.5
Length (mm)	310
Length of evaporator section (mm)	150
Length of condenser section (mm)	150
No. of tubes	50
Total contact surface area of heat pipe (m^2)	0.985
Heat input temp (^0C)	100
Air inlet temp (^0C)	8
Air outlet temp (^0C)	31
Heat Transfer (kW)	1.9
Heat Flux	1.92

The external ring steel fin is adopted at the condenser section. Thickness and height of fins are, respectively, 1.5 and 8 mm. The spacing interval between fins is 8 mm. Naked pipe is used in the heat pipe evaporator section in order to match the heat resistance.

Von der Fakultat et al. [30] has performed the experiments on loop heat pipe having copper as a wall material and water ass working fluid. Von der Fakultat et al. has observed that the minimum thermal resistance on heat pipe at 90^0oriented i.e. 3.7 k/w, further decreasing the degree of orientation, the thermal resistance is gradually increases which is shown in Figure2.15 . Hence with decreasing the degree of orientation from vertical to horizontal, the performance of heat pipe decreases simultaneously.

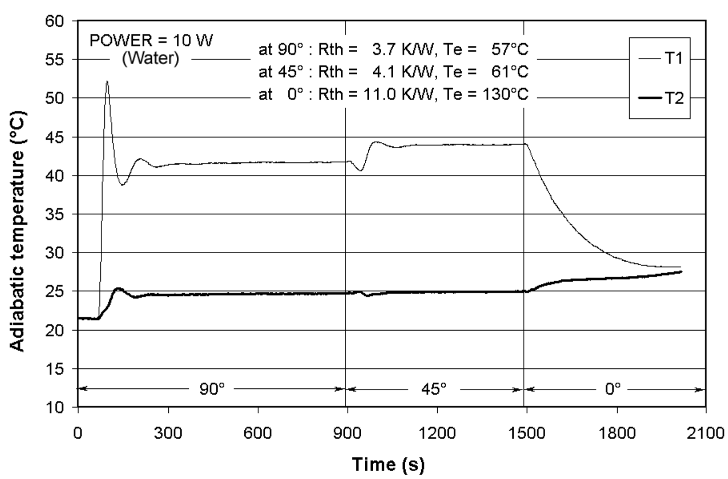


Figure 2.15: Thermal resistance Vs. degree of orientation[30]

2.6 Motivation of present study

In current scenario energy is the prime priority of each and every field. Thermal energy is most utilized form of energy in world. Re-covering and re-utilization of thermal energy is necessary in demand by using high efficient heat transfer systems. In conventional heat exchanger either shell and tube type or might be in finned tube type, we are normally getting heat flux is low compare to heat pipe. To fulfill current trend and requirement of thermal energy at low cost, needs to save it. This requirement motivated to develop high efficient heat pipe to enhance heat transfer to replace conventional heat exchangers.

2.7 Problem definition

Motive of this research work needed to develop heat pipe by approaching scientific method to evaluate heat flux, system efficiency and heat transfer coefficient by considering and varying different parameters to enhance heat transfer which can replace conventional heat exchangers. To design, develop and optimize heat pipe which can be use for the heat recovery system.

2.8 Objective of project

- To design heat pipe for waste heat recovery system as 120^0C evaporator temperature.
- To carry out experiment to find heat flux, system efficiency and thermal resistance for varying parameters.
- To do parametric study and optimization of different parameters i.e. wall material, wick material, wick layer, length of pipe and the degree of orientation.

Chapter 3

Heat Pipe Design Methodology

3.1 Problem definition and design criteria

The basic performance requirements of the specific application must be established before any design effort can be initiated. These parameters include operating temperature range, heat load requirements, allowable temperature drops, thermal control requirements, and size, weight and geometry limitations, design and operational constraints associated with testing, mechanical limitations, thermal interface requirements, filling, pressure containment specifications, toxicity requirements, and provisions for structural support must also be established. Also the type of application, aerospace or commercial, and ultimately cost effectiveness. This specification should be thorough and complete since it will be the document used for the design, development, and test efforts. A listing of the requirements which may be included in the heat pipe specification and their impact on the heat pipe design is given in Table3.1[3].

Table 3.1: Design Procedure [3]

Requirement	Impact on Heat Pipe Design
Operating Temperature Range	Choice of working fluid; pressure retention
Thermal Load	Heat Pipe diameter, number of heat pipes, wick design, and choice of working fluid
Transport Length	Wick design
Physical Requirements	Size, weight structural strength and geometry
Ground Testing	Degrees of freedom in orientation, limits on operating during testing
Thermal Environment	Atmospheric temperature consideration
Man Rating	Pressure Vessel Code; Fluid Toxicity
Mechanical Interfacing	Mounting and fabrication provisions
Transient Behavior	Choice of working fluid, wick design, variable conductance type
Reliability	Leak tightness requirements, material compatibility, processing care and control, Hazards

3.2 Design procedure

Once the specifications for a heat pipe application have been defined, the design selection and evaluation process can be classified in three basic considerations

(1) Selection of the working fluid.

(2) Selection of the wick design.

(3) Selection of the container design.

For a given application, several possible combinations of working fluid, wick structure and container design can be selected to satisfy the specifications.

3.2.1 Working fluid selection

A variety of physical, chemical, and thermodynamic properties of a particular working fluid must be evaluated to determine whether or not that fluid is suitable for the specific heat pipe application. The general considerations which apply to candidate fluids are given below Table 3.2.

Table 3.2: Properties of working fluid [4, 2]

Fluid	Group	Melting Point (K)	Normal Boiling Point (K)	Critical Temp (K)	Critical Pressure (10^5N m^{-2})	Temperature Range (K)
Helium	1	1	4	5	2.3	2-4
Hydrogen	1	14	20	33	12.9	14-33
Neon	1	24	27	44	26.5	27-44
Oxygen	1	54	90	154	50.9	55-154
Nitrogen	1	64	77	126	34.0	65-125
Argon	1	84	87	150	50.0	85-150
Propane	1	85	230	370	42.6	190-367
Freon	1	90	145	227	37.4	130-222
Ethane	1	90	185	305	49.1	100-305
Methane	1	91	112	190	46.4	91-190
Butane	1	94	272	42	38.0	260-350
Methanol	2	175	296	513	79.5	273-503
Toluene	2	178	337	593	41.6	275-473
Acetone	2	180	383	508	47.6	250-475
Ammonia	1,	195	239	405	112.9	200-405
Mercury	2,3	225	630	1763	15.10	280-1070
Dowtherm-E	2	-	453	690	40.3	283-610
Water	2	273	373	647	221.2	273-643
Benzene	2	278	353	562	49.2	280-560
Dowtherm-A	2	285	531	801	40.2	373-670
Cesium	3	301	943	2050	117.0	400-1500
Potassium	3	336	1032	2250	160.0	400-1800
Sodium	3	371	1152	2500	370.0	400-1500
Lithium	3	453	1615	3800	970.0	500-2100
Silver	3	1234	12450	7500	336.0	1600-2400

3.2.1.1 Operating temperature range

As per given task by industry's requirement operating temperature range is 120 ^{0}C to 130 ^{0}C which is in need to recover through heat pipe, heat pipe should be developed for this temperature range. For this temperature range from Table 3.2 water, methanol, ethanol, acetone and pantane are most suitable working fluids.

3.2.1.2 Liquid transport factor

In heat pipe liquid transport factor is basically depends on latent heat of working fluid. Liquid transport factor and merit no. are higher for working fluid having higher latent heat. For this application the latent heat of preferable working fluids are mentioned in Table 3.3.

Table 3.3: Latent heat of preferable working fluids[3]

Working Fluid	Latent Heat (kJ / kg)
Water	2257
Methanol	978
Ethanol	781
Acetone	468
Pantene	296

By, considering the above Table 3.3 water is having highest latent heat, and should be most preferable for this application.

3.2.1.3 Critical pressure properties

The heat pipe is a device, which could under high vacuum or high pressure also, if it is designed for high operating temperature range. For this purpose critical pressure should also be high. As per detailed given in Table 3.2 water is having highest critical pressure amongst all preferable fluids which is $221.2 \times 10^5 Nm^{-2}$.

3.2.1.4 Vapour specific heat for higher merit number

With latent heat, vapour specific heat also plays a great role to achieve high heat transfer and merit number. To select working fluid vapour specific heat should also be kept in consideration. The vapour specific heat of different preferred working fluids are covered in Table 3.4.

Table 3.4: vapour specific heat[3]

Working fluids	Vapour specific heat (kJ / kg^{0}C)
Water	2.34
Methanol	1.62
Ethanol	1.89
Acetone	2.01
Pantene	1.088

By, referring these values water having high vapour specific heat in comparison to other preferred working fluids.

3.2.1.5 Liquid thermal conductivity

To decide working fluid for particular heat pipe thermal conductivity should also be as high as possible to achieve high heat flux. The value of liquid thermal conductivity of different preferred working fluids are listed in Table 3.5.

Table 3.5: Liquid thermal conductivity[3]

Working fluids	Liquid thermal conductivity (W/m^{0}C)
Water	0.86
Methanol	0.189
Ethanol	0.154
Acetone	0.148
Pantene	0.128

With the reference of these values water is having highest liquid thermal conductivity amongst other preferred working fluids.

3.2.1.6 Fluid stability, toxicity, availability

Amongst all preferred working fluid, the water is most stable, non-toxic and non hazards and easily available with cheapest cost. While acetone, pantene, methanol and ethanol are partially toxic and creates irritation on skin and eyes. These working fluids are expensive compare to water. From operation point of view it is preferable to use water at priority level if it satisfy all other criterias.

3.2.1.7 Selected working fluid

By considering all parameters in deciding working fluid, water stand first among all preferred working fluids which satisfy all selection criteria. So it is decided to use water as a working fluid for this application.

3.2.2 Selection of wall material

The function of the container is to isolate the working fluid from the outside environment. It has to be leak-proof to maintain the differential pressure across the pipe wall to enable the transfer of heat from the working fluid. Selection of the container material depends on several factors.

3.2.2.1 Compatibility (both with working fluid and the external environment)

The two major results of incompatibility are corrosion and the generation of non-condensable gas. If the wall or wick material is soluble in the working fluid, mass transfer is likely to occur between the condenser and the evaporator, with solid material being deposited in the latter. This will result either in local hot spots or blocking of the pores of the wick. Non-condensable gas generation is probably the most common indication of heat pipe failure and, as the non-condensable tend to accumulate in the heat pipe condenser section, which gradually becomes blocked, it is easy to identify because of the sharp temperature drop that exists at the gas/vapour interface.

Table 3.6 provides compatibility information of all material and wick material for various working fluids. In earlier selection, it was decided to use water as a working fluid and from Table 3.6 copper, monol, stainless steel and carbon steel are compatible material. For deciding wick material to be used for contained certain further design and property analysis is required which is maintained in proceeding sections.

By this compatibility data for water as a working fluids, copper, 347-stainless steel and carbon steel satisfies the requirement with ease of market availability.

Table 3.6: Compatibility Recommendation [3, 2, 26]

Working Fluid	Recommended	Not-recommended
Ammonia	Aluminum	Copper
	Carbon steel	
	Nickel	
	Stainless steel	
Acetone	Copper	
	Silica	
	Aluminum	
	Stainless steel	
Methanol	Copper	
	Stainless steel	
	Silica	
Water	Copper	Stainless steel
	Monel	Aluminum
	347 Stainless steel	Silica
	Carbon steel	Inconel
		Nickel
Dowtherm A	Copper	
	Silica	
	Stainless steel	
Potassium	Stainless steel	Titanium
	Inconel	
Sodium	Stainless steel	Titanium
	Inconel	

3.2.2.2 Thermal conductivity

After analysis, compatibility of different material with selected working fluid, also the consideration of thermal conductivity for different materials, which plays important role in heat transfer property should be considered. It should be as high as possible to achieve high heat flux. The thermal conductivity of different short listed material are given in Table 3.7.

Table 3.7: Thermal conductivity of different heat pipe wall material [3]

Material	Thermal conductivity(W/m ^{0}C)
Aluminum	205
Brass	113
Copper	**398**
Glass	0.75
Nickel	88
Mild steel	45
Stainless Steel	**17.3**
Carbon Steel	**54**
Teflon	0.17

Amongst the compatible working fluids selected in 3.2.2.1, copper is having highest thermal conductivity. Thermal conductivity of copper is approximately 22 times higher then stainless steel and 8 times higher then carbon steel. Thus from heat transfer point of view, copper is best material amongst all compatible wall materials.

3.2.2.3 Ease of fabrication, including weld-ability, machine-ability and ductility

From fabrication consideration copper, stainless steel and carbon steel, all stands almost equal as well established fabrication technologies are available.

3.2.2.4 Cost effectiveness

In current scenario for heat transfer and heat recovery application cost stands is prior requirement to replace the conventional heat exchangers by heat pipe heat exchangers. With having high thermal conductivity and good compatibility it must to be cheap in cost to make it acceptable in current market. Some preferred wall material are listed below with it's approx. and estimated current market cost in Table 3.8.

Table 3.8: Cost analysis of wall material

Material	Approx. Cost (Rs./kg)
Copper	620
Stainless Steel	410
Carbon Steel	115

By referring above Table 3.8 carbon steel is preferred material as it is cheap also easily and locally available material amongst all three.

3.2.2.5 Selected wall material

Based on section 3.2.2.1 to 3.2.2.4, it can be concluded that from heat transfer point of view carbon steel and copper is bast suitable material compared to stainless steels decided to take carbon steel and copper for parametric study to optimize the difference of heat flux at low cost which could be alternative of conventional heat exchangers.

3.2.3 Selection of screen mesh wick

The selection of the wick for a heat pipe depends on many factors, several of which are closely linked to the properties of the working fluid. The prime purpose of the wick is to generate capillary pressure to transport the working fluid from the condenser to the evaporator. It must also be able to distribute the liquid around the evaporator section to any areas where heat is likely to be received by the heat pipe. Often these two functions require wicks of different form particularly where the condensate has to be returned over a distance of, say, 1 meter [2].

The heat transport capability of the heat pipe is raised by increasing the wick layers. The overall thermal resistance at the evaporator also depends on the conductivity of the working fluid in the wick. Other required properties of the wick are compatible with the working fluid. It should easily form to be molded into the wall shape of the heat pipe. It should be economical.

Two types of wick structure are proposed for this heat pipe, homogeneous and arterial types.

3.2.3.1 Selected wick

As shown in the result Table 3.9 and as per industrial guidance, selected 200 mesh stainless-steel and carbon bronze wick for heat pipe application should be carried out to decide single layer is sufficient or 2 layers to improve the result and to carried out effect of wick layer on heat transfer.

Table 3.9: Wick results [7, 3, 2, 26]

Working Fluid	Wick	Vapour Temp. (^{0}C)	Heat Flux(W/cm^2)
Water	100 s.s	140	2.5
Water	150 s.s	155	3.4
Water	200 s.s	170	6.2
Water	300 s.s	185	4.6
Water	Nickel felt	90	3.5
Water	100 copper	160	5.8
Water	200 copper	160	7.1

3.2.4 Selection of working fluid filling ratio

As described in Table3.1 due to filling of working fluid in heat pipe, maximum heat duty has been observed between 30-40% of total volume of heat pipe for water as a working fluid. hence, the filling ration of working fluid has been taken as 35% to gain higher output

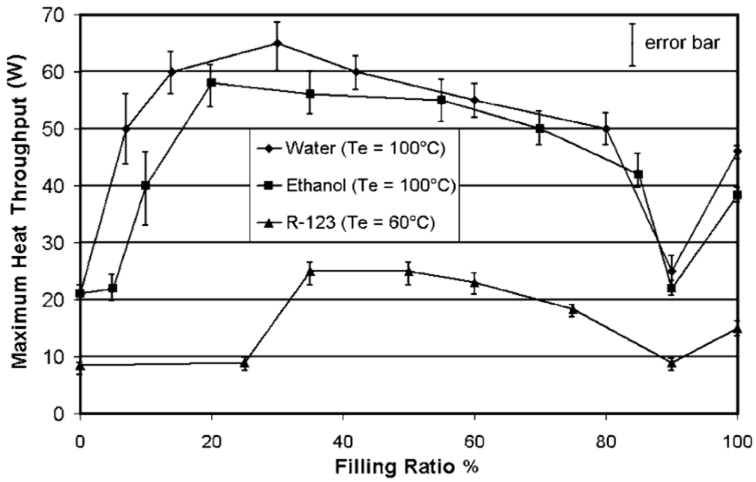
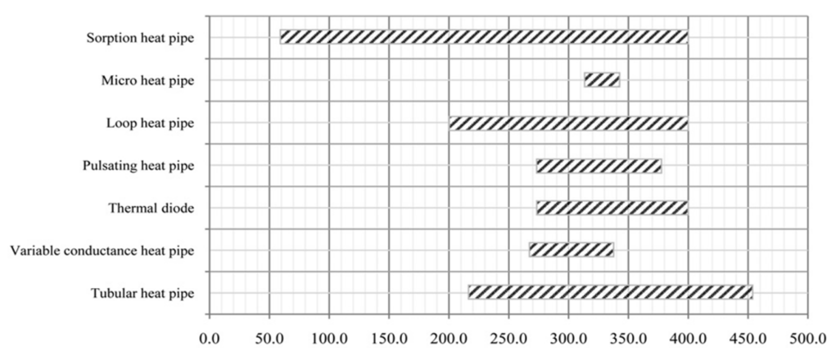


Figure 3.1: Maximum performance of heat pipe obtained with respect to the filling ratio[27]

3.2.5 Selection of shape

To decide shape of the heat pipe for required temperate range the preferred shape of heat pipe is concluded as tubular by referring literature Figure 3.2.



	Tubular heat pipe	Variable conductance heat pipe	Thermal diode	Pulsating heat pipe	Loop heat pipe	Micro heat pipe	Sorption heat pipe
Average Operating Temp. (K)	335.5	303.0	335.5	325.5	300.0	273.0	230.0

Average Operating Temp. (K)

Figure 3.2: Shape of heat pipes [8]

Chapter 4

Fabrication of Heat Pipe

4.1 Cleaning of container

All the materials used in a heat pipe must be clean. It ensures that the working fluid will wet the materials and that no foreign matter is present which could creates decrement in capillary action or creates incompatibilities. The cleaning procedure depends upon the material used, the process undergone in manufacturing location of wick, and the requirements of the working fluid.

Working fluid is, water, no extreme precautions are necessary to ensure good wetting, and an acid pickle appears to be satisfactory. Process of cleaning naked pipe is mentioned below:

- Initially naked pipes are taken for water wash for preliminary cleaning.
- The pickling process for carbon steel involved immersing the components in a solution of 50 per cent sulfuric acid and 5 per cent hydrofluoric acid. This is followed by in demineralized water.
- After cleaning and pickling process satisfactorily, naked pipe is kept at approx. 55 ^{0}C for 1 hour for drying process.
- After completion of 1 hour it is cleaned with brush and compressed air washing is carried out at inside and outside of pipe.

4.2 Wick and cap installation

- Mandrel is used for inserting wick inside tubular container. Tube internal diameter is 22.2mm so mandrel size selected is smaller then tube diameter.
- Number of layers of wick are wrapped on mandrel as shown in Figure 4.6.
- This wick wrapped on mandrel, which is inserted in naked pipe with continuously rotating both in opposite direction while inserting it which is shown in Figure 4.1.

- After inserting wick successfully and carried out check of proper positioning of wick towards pipe wall, upper and lower caps were fitted by argon arc welding as shown in Figure 4.3 and Figure 4.4 .
- The advantage of welding over brazing or soldering is that no flux is required, therefore the inside of cleaned pipes do not suffer from possible contamination.
- The schematic diagram of heat pipe with wick is shown in Figure 4.6.
- After completing the welding process it is necessary to remove all flux which is settled down on pipe peripheral edges by grinding process as shown in Figure 4.5.

Figure 4.1: Preparation of wick to insert in pipe

Figure 4.2: Inserting Wick in pipe

Figure 4.3: Upper and lower sealing Cap of pipe

Figure 4.4: Welding of caps with pipe

Figure 4.5: Grinding caps to insert pipe in test rig

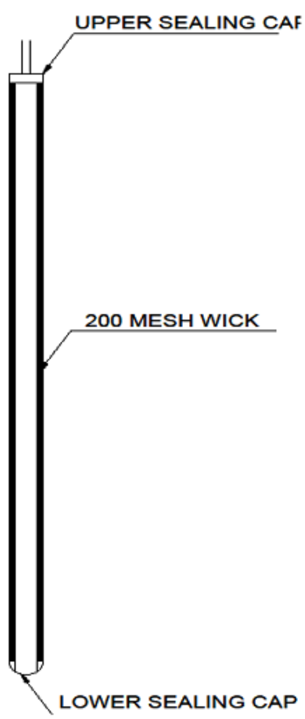


Figure 4.6: Schematic diagram [5]

4.3 Leak detection

All welds on heat pipes should be checked for leaks.A rigorous leak check procedure is necessary because a small leak that may affect heat pipe performance as shown in Figure 4.7. The procedure of leak detection is as follows:

- After inserting wick and sealing upper and lower caps the pressure pump is used to develop pressure in the pipe system.
- After creating the pressure inside the sealed pipe leak test was carried out in which heat pipe has been deepen in water tank. Based on bubbling it is possible to detect the leakage and its place. If leak is detected then re-weld and re-check it for leakage. When leak test is OK, heat pipe is ready for filling operation.

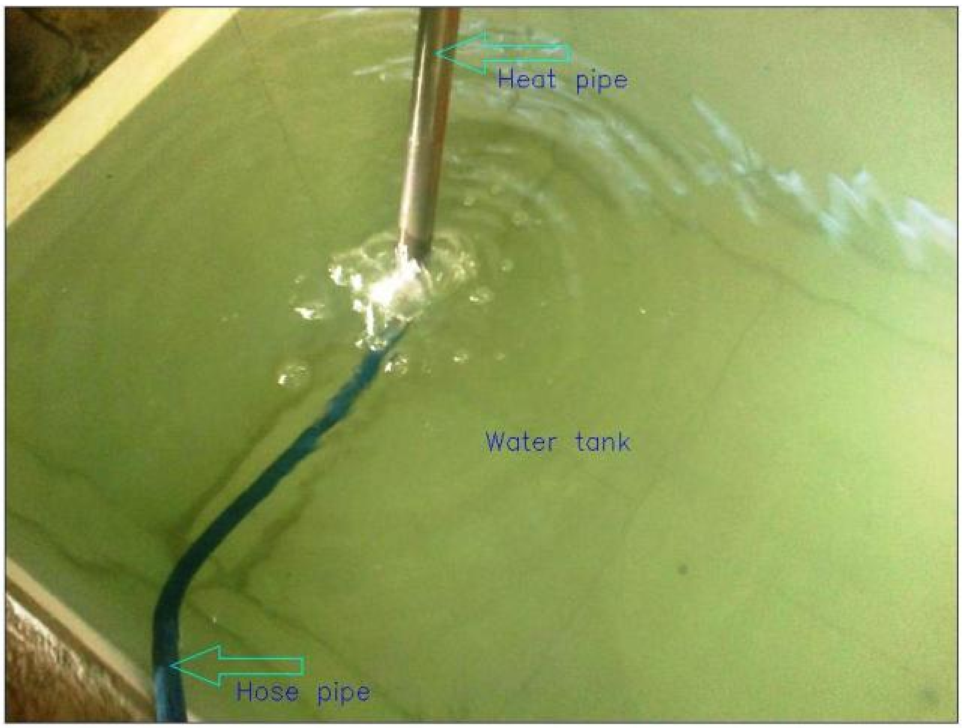


Figure 4.7: Leak detection test

4.4 Preparation of the working fluid

It is necessary to treat the working fluid used in a heat pipe with the same care as that given to the wick and container. The Demenerlized water is used as working fluid, still additional purification process is required for removal of dissolved gases which is mentioned as follows:

- Initially freeze-degassing process has been carried out on deminerlized water by freezing it at below -5^{0}C. This process removes all dissolved gases from the working fluid, and if the gases are not removed they could be released during heat pipe operation and can collected in the condenser section which would affect the performance of heat pipe.

- After freeze-degassing, Predefined mass of working fluid (D.M water) is filled in heat pipe and then heat pipe is connected to filling setup.

4.5 Filling and sealing procedure of heat pipe

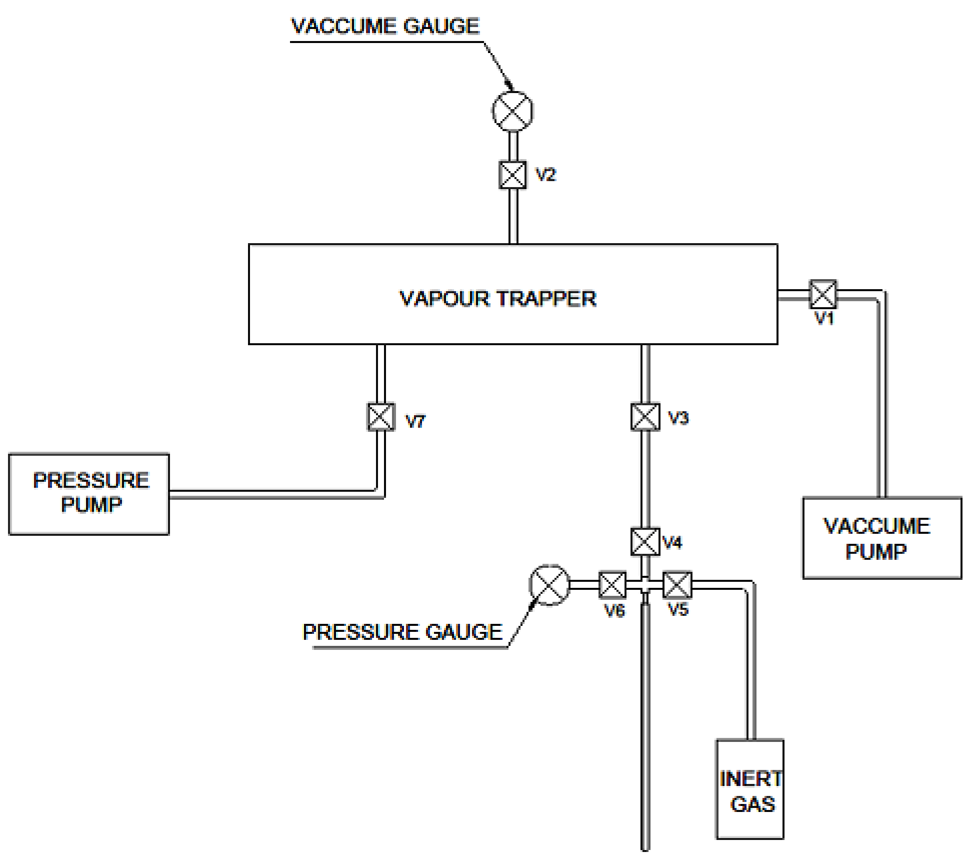


Figure 4.8: Schematic diagram of experimental setup for heat pipe filling

An experimental setup for filling heat pipe is shown in Figure 4.8.

Procedure carried out while filling and preparing heat pipe are as follows:

- After completing filling, cleaning, sealing processes connect pipe with flare set at 4-way between V5 and V6 as shown in Figure 4.8.
- Close V1, V2, V5 and V6 and open V7, V3 and V4.
- Switch on pressure pump and insert pipe in water tank to check leakages in system.

- After checking for leakages, close V7 and open V1 and V2.
- Switch on vacuum pump and make required vacuum in the system, which is checked by vacuum gauge placed above vapour trapper.
- Close V1, V2, V3 and V4 and open V5 and V6 and insert inert gas as per required pressure which would be observed from pressure gauge connected at V6.
- Close V4, V5, V6. Make pinch at upper filler copper pipe.
- Braze the pinched copper pipe.

Vacuum pump, pressure pump, cylinder, heat pipe, control valves, pressure gauge and liquid trapper is used for filling of heat pipe as shown in Figure 4.9.

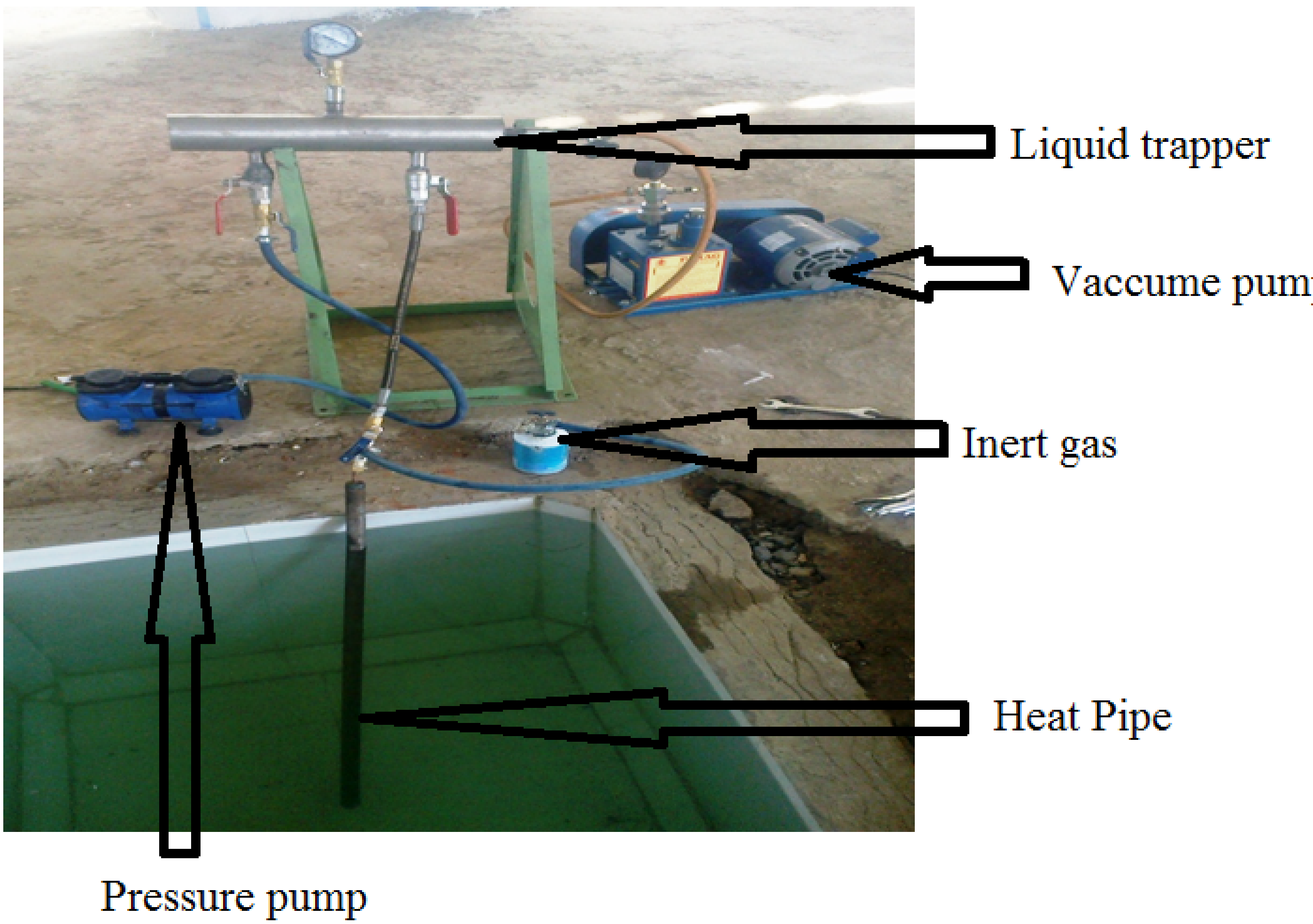


Figure 4.9: Heat pipe filling

Actual sealing process has been carried out by using crimp tool is shown in Figure 4.10 and crimped heat pipe is shown in Figure 4.11.

Figure 4.10: Crimping of processed heat pipe

Figure 4.11: Crimped heat pipe

Chapter 5

Results and Discussions

5.1 Heat pipe performance measurement set-up

The measurement of the heat pipes performance is comparatively easy and requires general equipment available in any laboratory engaged in heat transfer work. During testing parametric study was carried out for different:

- Wall material
- Wick material
- Wick layer
- Length of the pipe
- Degree of orientation

A schematic diagram of test rig is shown in Figure 5.1.

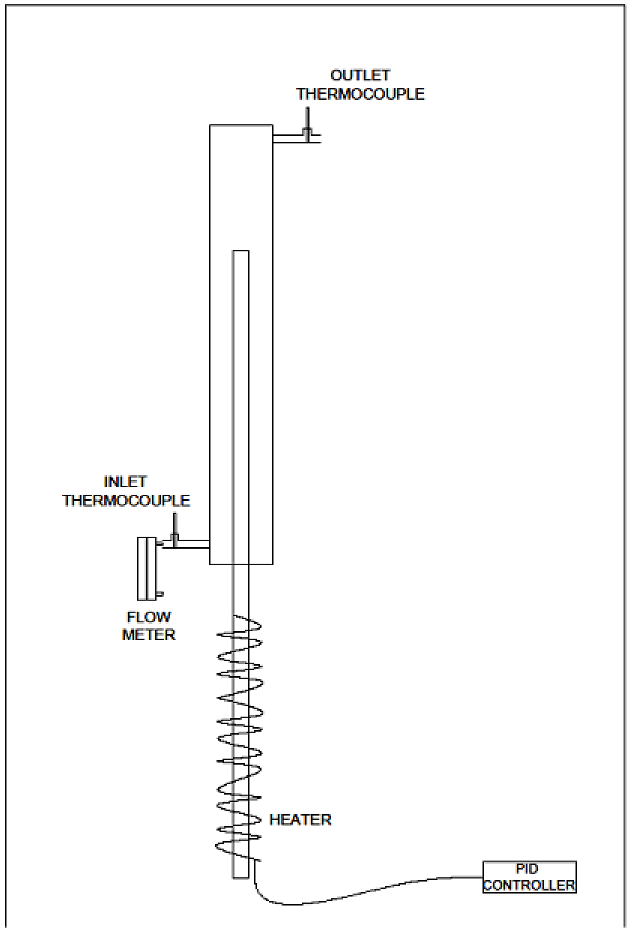


Figure 5.1: Proposed test rig

5.1.1 Heater for evaporator section

At the evaporator section standard heaters were tightly wrapped on the surface of the circular heat pipe as shown in Figure 5.2, which provides uniform heat flux to the evaporator section.

Specifications

Heater length: 250 mm

No. of heater: 2 nos.

Heat Input : 1kW

Heater ohms: 54 Ω

Evaporator length: 500 mm and 1 meter.

diameter: 25.4 mm OD & 22.2 mm ID

Figure 5.2: Heater winded pipe

5.1.2 Condenser section

Condenser section is used of heat pipe to transfer the heat from vapour of the working fluid to the cold side media. Condenser section of the heat pipe consisting specifications are:

Condenser length: 500 mm and 1 meter.

diameter: 25.4 mm OD & 22.2 mm ID

5.1.3 Watt meter for power input measurement

Watt meter has been used to measure the power input by the electric heater by which we can measure energy input data.

5.1.4 PID controller for power control

Temperatures was observed and recorded by a data acquisition system of standard SELEC and PID controller, as shown in Figure 5.3. Figure 5.2 shows various locations where sensors are attached and temperatures are measured. Calibration certificates of different temperature sensors and controllers are shown in Appendix - IV.

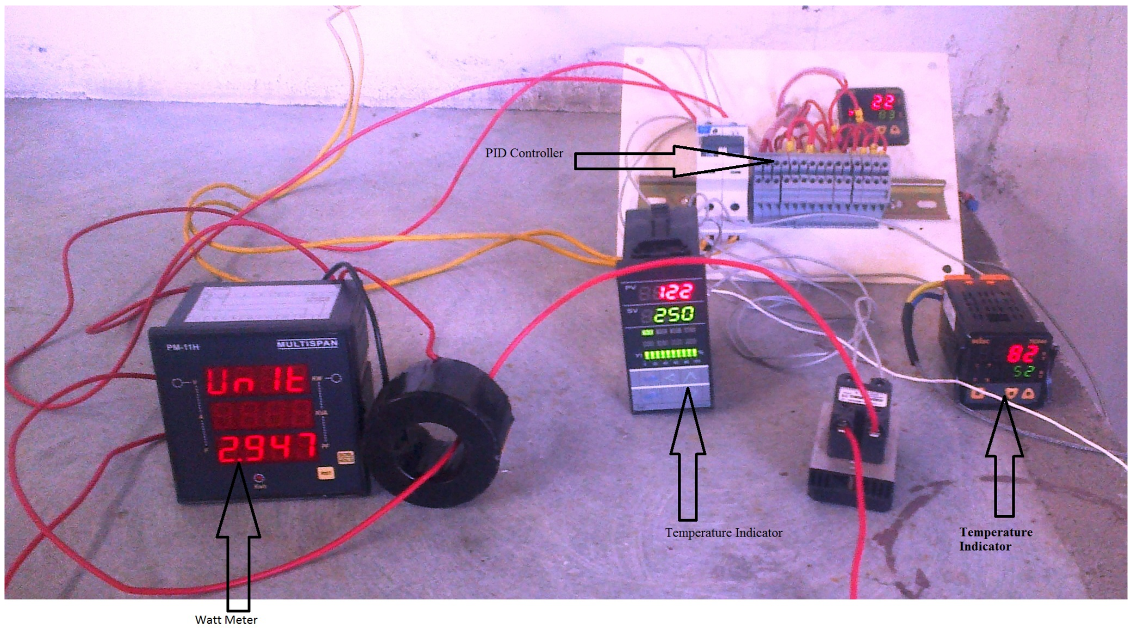


Figure 5.3: Data acquisition system

5.1.5 Condenser for Heat Transfer

Heat transferred by condenser is measured by circulating water surrounding to condenser section into inner shell. Water is coming from the source at lower temperature and leaving the shell at higher temperature. Th M.O.C of shell is stainless steel and is shown in Figure 5.4. In Figure 5.4the condenser shell for 1 meter and 2 meters length of pipes are shown having technical specifications are as follows:

Internal diameter : 122mm

Outer diameter : 130mm

Height : 700mm

Fluid storage capacity : 15 kg

Figure 5.4: Condenser shell

5.1.6 Flow measuring device (Rota-meter)

To measure inlet and outlet temperature of water and temperature of input energy thermocouples are used with range and calibration are described in calibration certificate shown in Figure **??**. Also rota-meter as shown in Figure 5.5 has been used for measurement of water flow going to condenser section to carry out analysis of total heat gain by the heat pipe. The calibration certificate of Rota-meter which is used in experiment, is shown in Appendix - V.

Figure 5.5: Flow measuring device with control valve

5.1.7 Insulating Materials

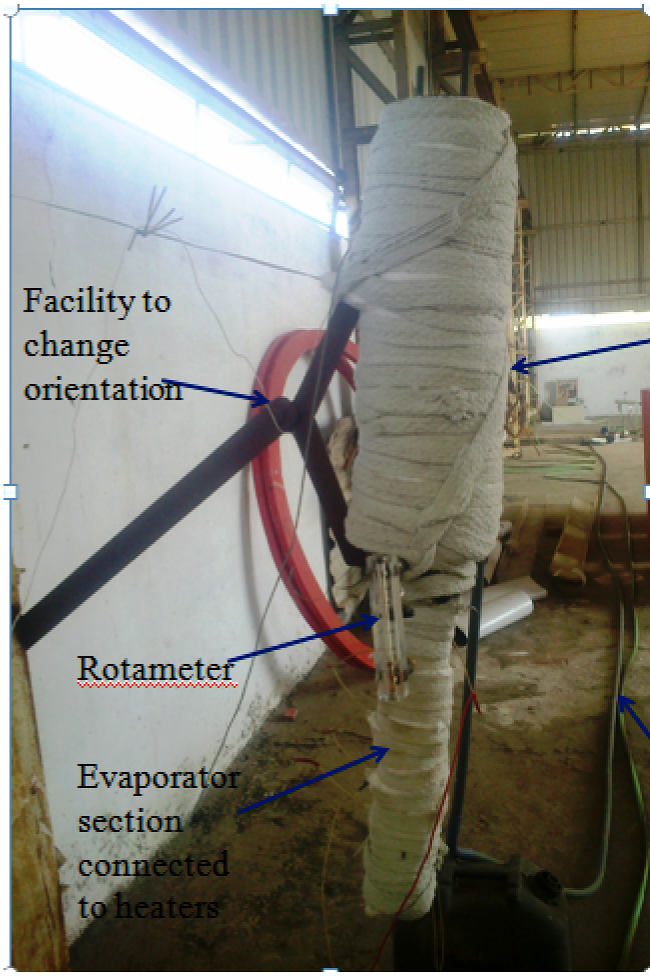
The heat pipe, evaporator section and condenser section were insulated by glass wool and teflon tape. The thermal conductivity of glass wool is 0.04W/m-K at 25 ^{0}C and Teflon tape is 0.264 W/m-K. First Teflon tape was used in sealing accessories and three layers of glass wool was wrapped in order to reduce the heat losses by convection and radiation to the environment from the experimental apparatus.

5.1.8 Heat Pipe

Heat pipe was prepared having wall materials are carbon steel and copper with the combination of wick having 1 and 2 layer, length of 1 meter and 2 meters having diameter of 25.4 mm.

5.1.9 Heat sink

To transfer heat from condenser section of the heat pipe, water was used. The water jacket with 30 LPM flow rate water can be considered as constant temperature heat sink. Figure 5.6 shows the setup detail.



Evaporator section

Water supply line

Figure 5.6: Water supply

5.2 Experimental investigation

An experimental apparatus for the heat pipe performance test is composed of the evaporator section, At the evaporator section, two heaters were wrapped on the heat pipe and between that thermocouple was mounted on the outer surface of the heat pipe. All the heaters were connected in parallel to supply uniform heat flux in the evaporator section. To account for this heat transferred to condenser section, temperature at the inlet and outlet and the flow rate of cooling water was measured. Heat transferred to cooling water was measured using flow rate from rota-meter and density of the water.

$Q = mC_p(T_{wo} - T_{wi})$ 5.1

Where

T_{wo} and T_{wi} is the outlet and inlet temperatures of the cooling water flow respectively

Cp is the specific heat of water m is the mass flow rate of water, which was calculated by collecting water in a shell

of known volume.

5.3 Experimental Procedure

- A constant water with measured flow was circulated through condenser shell.
- The constant water flow rate was measured by rota-meter, which was attached in line to measure the mass of water.
- Heat input of 2KW was given to the evaporator section by electrical heaters.
- It took 90 minutes for the system to be in steady state condition, the temperature readings were noted and watt meter were re-set to zero. It was observed, there was no temperature change in time period of 60 minutes, again final temperature and power input were noted. Once the steady state condition has been reached after 60 minutes, reading of water flow rate, temperatures water inlet and outlet, power input was measured for various parametric parameters.
- Similar experiments were repeated for the varying parameters like wall material, length, wick material, wick layers and orientation of heat pipe.

5.4 Results and discussions

Experimentation was performed on heat pipe to determine heat transfer rate, heat flux, system heat transfer efficiency, resistance. This parameter are calculated base on input energy from heater to evaporator section, energy transfer to cooling water from condenser section, heat transfer area etc. Primary measurements are temperature of water inlet, water outlet, mass flow rate of water, wattage, dimensions of heat pipe. Parametric study was carried out to know effect of various parameters. For parametric study following variations are included as follows:

- Effect of change of wall material
- Effect of change in length of heat pipe
- Effect of use of single and double layer of wick
- Effect of change of orientation

5.4.1 Experimental result

5.4.1.1 Response time of Heat Pipe

Table 5.1 gives detail of variation in water inlet and water outlet temperature with time and same is plotted in Figure 5.7. It was observed that within approximately 90 minutes steady state condition had been achieved.

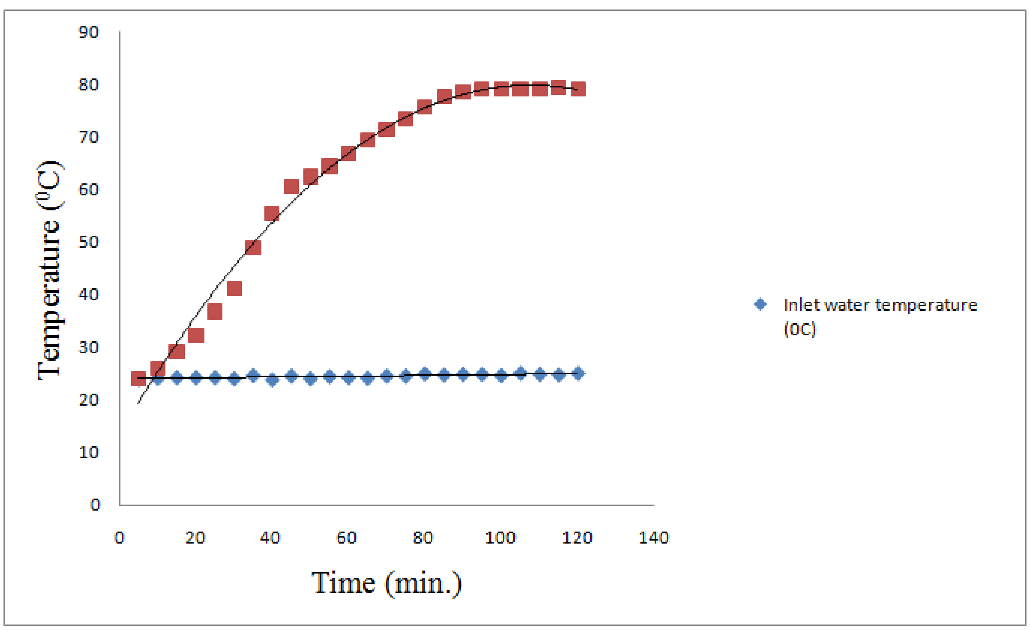


Figure 5.7: Time Vs. Temp(IN-OUT)

Table 5.1: Result of Response Time

Time (min.)	Inlet Water Temp. (^0C)	Outlet Water Temp(^0C)
5	24	24
10	24.2	26
15	24.3	29.1
20	24.3	32.3
25	24.3	36.7
30	24.1	41.1
35	24.7	48.8
40	23.9	55.3
45	24.6	60.5
50	24.1	62.4
55	24.5	62.4
60	24.3	64.4
65	24.2	66.8
70	24.6	69.4
75	24.6	71.4
80	25	73.3
85	24.8	75.6
90	24.9	77.6
95	24.9	78.5
100	24.7	79.1
105	25.1	79
110	24.9	79.1
115	24.8	79.3
120	25.1	79.1

5.4.1.2 Heat Flux:

Table 5.2 and 5.3 mentioned below compares average of 3 experimental datas, detailed given in appendix I,

M.O.C:

length	:	1 and 2 meter
wall material	:	Carbon steel and copper
wick	:	1 and 2 layer
wick material	:	stainless steel and copper

Table 5.2: Results of different combination of (1 meter) heat pipes at 90^0orientation

	Case-1	Case-2	Case-3	Case-4	Case-5
Wall material	C.S	C.S	C.S	C.S	Copper
Heat Pipe Length (m)	1	1	1	1	1
Wick Material	nil	S.S	S.S	Copper	S.S
Wick Layer	nil	1	2	1	1
Input Heater Temp.(^0C)	120.4	123	121.3	118.1	122.26
Input Water Temp.(^0C)	22.03	19.76	21.96	21.26	22.06
Output Water Temp.(^0C)	57.93	67.26	76.26	71.36	73.1
Mass Flow Rate (LPH)	30	30	30	30	30
Heat exchange surface area(m^2)	0.0398	0.0398	0.0398	0.0398	0.0398
Energy Input by Heater (kJ / hr)	7339.96	6839.63	7501.91	6994.46	7037.96
Energy Gain by Water (kJ / hr)	4501.86	5956.5	6809.22	6282.54	6399.58
Total Heat Flux (kJ $/m^2$)	112890.8	149368.1	170751.3	157544	160479
Total Heat Flux (kW $/m^2$)	31.4039	41.55	47.49	43.82	44.64
Efficiency (%)	61.35	87.093	90.76	89.82	90.93

Analysis of Table 5.2 and 5.3 shows that energy input are almost same in all heat pipes used for parametric studies. While comparing energy transfer using heat pipe in different case,

a) It is found that heat pipe made of carbon steel wall material with stainless steel 1 layer wick is able to transfer approximately 87% of total heat input.

b) With 2 layers of stainless steel wick or 1 layer of copper wick, carbon steel heat pipe increases heat transfer rate by 4% against 1 layer wick of stainless steel.

c) If 1 layer of copper wick used with carbon steel container then heat transfer efficiency is similar to double layer wick of stainless steel with carbon steel as a wall material.

d) Copper heat pipe with 1 layer of stainless steel wick gives best efficiency i.e. 91%, however its result is similar to carbon steel heat pipe with 2 layer of stainless steel wick and single layer of copper wick.

It seems that if copper heat pipe with double layer stainless steel wick is used then results improves, but if copper wick with 2 layers is used then it may give better result. However initial cost of this heat pipe is quite high compared to carbon steel heat pipe with 2 layer of stainless steel wick. One should do economic analysis based on application for deciding optimum results.

Table 5.3: Results of different combination of (2 meter) heat pipes at 90^0orientation

	Case-6	Case-7
Wall material	C.S	C.S
Heat Pipe Length (m)	2	2
Wick Material	S.S	S.S
Wick Layer	1	2
Input Heater Temp.(^0C)	156.76	164
Input Water Temp.(^0C)	23.16	20.66
Output Water Temp.(^0C)	71.5	73.46
Mass Flow Rate (LPH)	60	60
Heat exchange surface area(m^2)	0.0799	0.0799
Energy Input by Heater (kJ / hr)	13917.7	14564.97
Energy Gain by Water (kJ / hr)	12122	13242.24
Total Heat Flux(kJ $/m^2$)	151988.6	166034.4
Total Heat Flux (kW$/m^2$)	42.28	46.18
Efficiency (%)	87.11	90.91

Experiments where carried out on 2 meters length of carbon steel heat pipe with 1 and 2 layer of stainless steel wick. It is observed from Table 5.2 and5.3 that heat transfer efficiency result is not affected due to change in length of heat pipe.

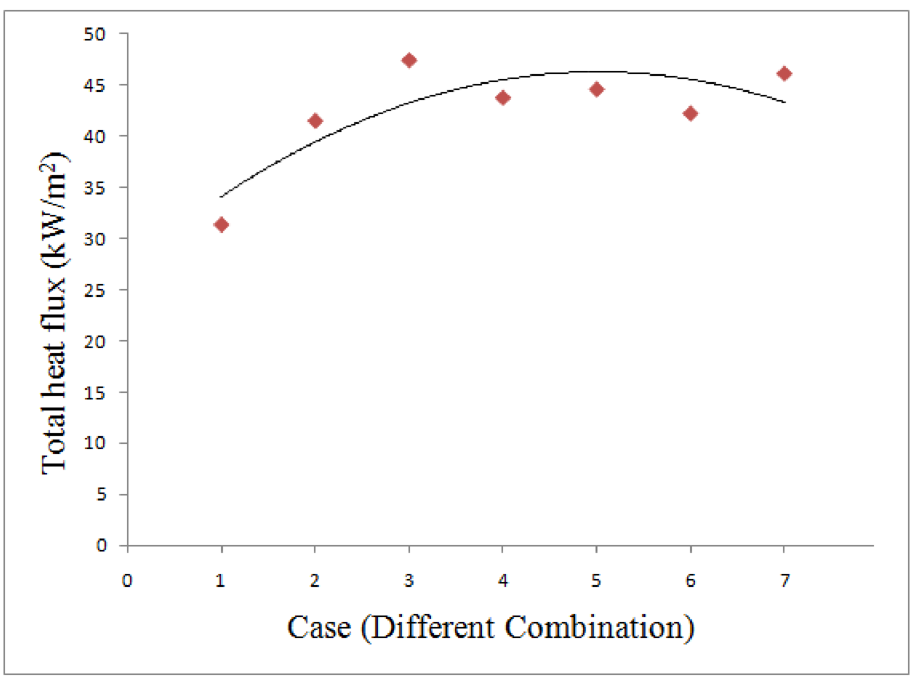


Figure 5.8: Heat Flux Vs. Different Combination

Figure 5.8 shows information of heat flux of various types of heat pips (mentioned in Table 5.2 and5.3 as case 1 to 7 respectively). It can be seen from Figure 5.8 that heat flux improves drastically by use of wick as result for case 2 to 7 is much higher then case 1. Use of different wall material, wick material and wick layers results are varying in range of 41 to 47 kW/m^2. Result variations are about 15% from case-2 (single layer stainless steel wick with carbon steel heat pipe to case-3 double layer of stainless steel wick with carbon steel heat pipe). By use of copper wick, copper wall material result is not improving to great extent inspite of cost of heat pipe definitely improves as copper is vary costly then carbon steel.

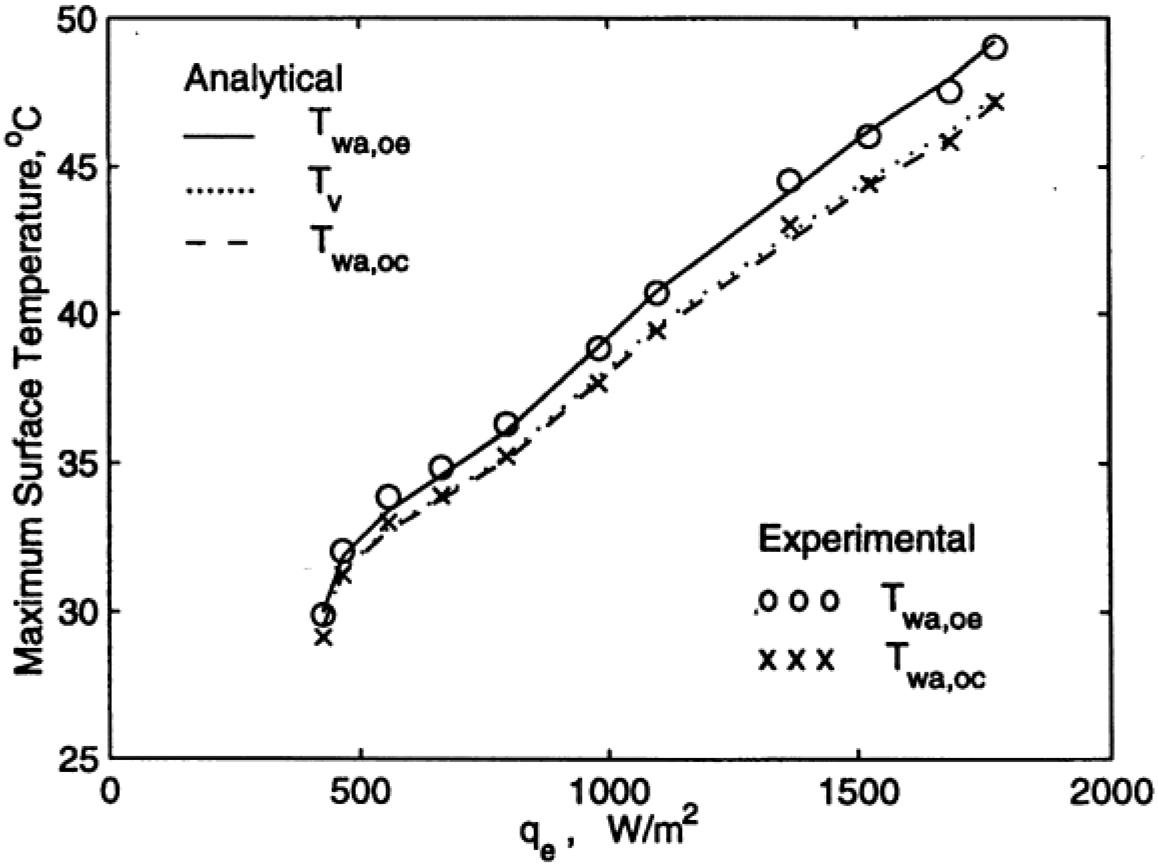


Figure 5.9: Effect of the input heat flux on the maximum surface temperature[28]

Results obtain by present study is quite higher then the results of heat flux given in Figure 5.9.

5.4.1.3 Efficiency:

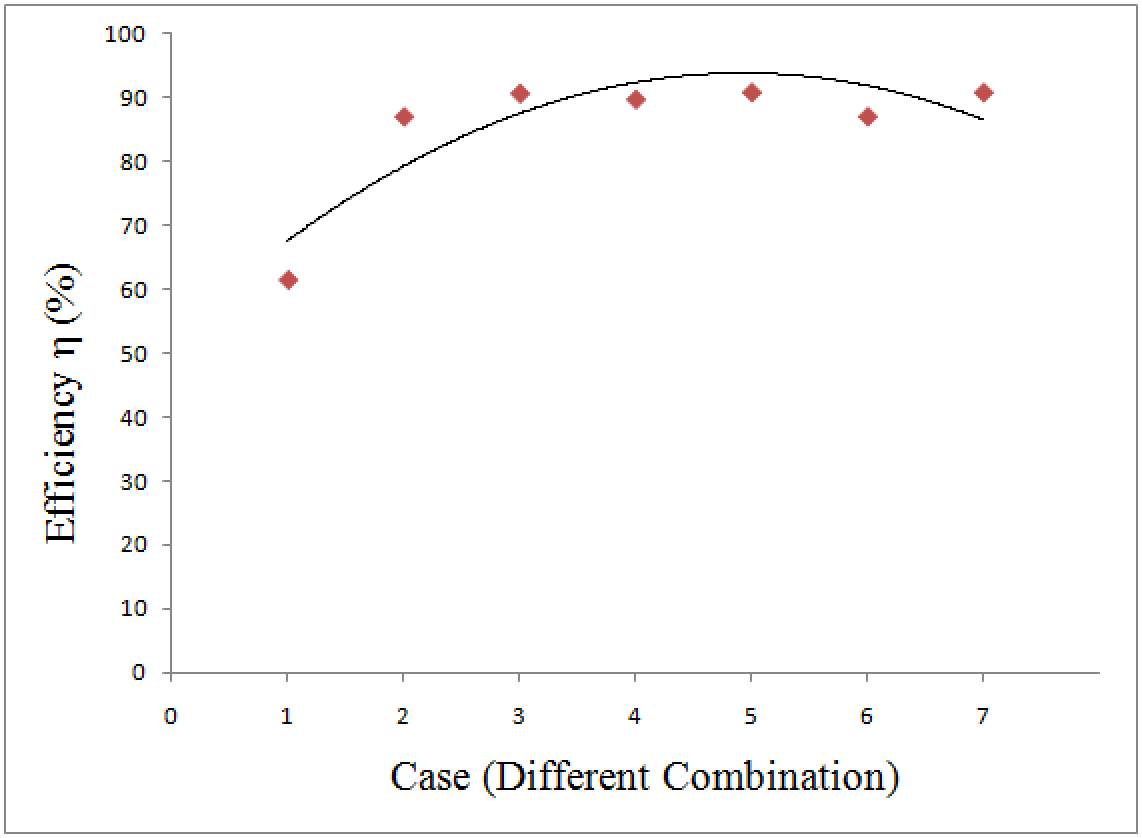


Figure 5.10: Efficiency Vs. Different Combination

Figure 5.10 shows plot of efficiency vs. cases mentioned in Table 5.2 and5.3. Detailed discussion is mentioned above.

B sivraman et al. carried out work on copper material heat pipe of 1 meter length and 22 mm diameter with 50 mesh single layer wick of stainless steel. Details of B sivaraman et al. work is mentioned in figure . Dimensions of heat pipe is to similar to present work however evaporator and condenser length in their case is 0.85 meter and 0.15 meter respectively and in present work evaporator and condenser sections is 0.5 meter each. Wick size in present work is 200 mesh and B sivaraman et al. used 50 mesh wick. Thus both cases are not 100% similar still it gives platform for comparison of present result with literature. Maximum efficiency achieved by B sivaraman et al. is 67% which is 87% in present work of single layer of stainless steel wick.

Table —1 Specifications of heat pipe

Description	Pipe – 1	Pipe - 2
Outer diam of heat pipe, m	0.019	0.022
Inner diam of heat pipe, m	0.017	0.019
Total length, m	1.0	1.0
Evaporator length, m	0.85	0.85
Condenser length, m	0.15	0.15
Wick mesh size number	50	50

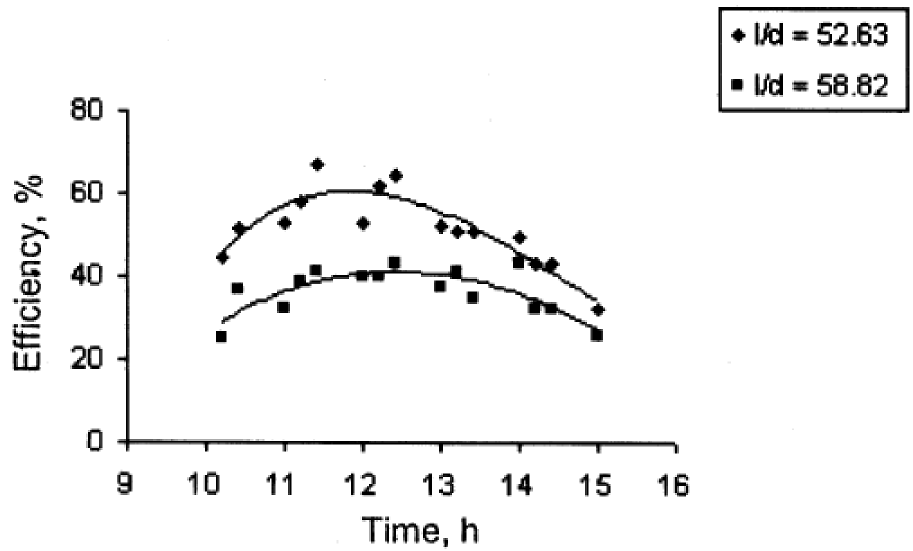


Fig. 7—Time vs efficiency at different L/d ratio
(flow rate 30 kg/h)

Figure 5.11: Experimental results of Different L/d_i ratio[29]

5.4.1.4 Effect of Orientation:

Table and figure provides information of heat flux and efficiency variation with heat pipe orientation for carbon steel heat pipe with 2 layers of stainless steel wick.

Table 5.4: Effect of Degree of Orientation

Degree of Orientation	Total Heat Flux (kW/m^2)	Efficiency (%)
90^0	47.49952471	90.76660579
60^0	44.26290885	86.74404298
45^0	41.46367351	82.62729889
30^0	39.71415142	76.60849088
0^0	35.25287009	67.66670237

It can be observed from Table 5.4 and Figure 5.12 and 5.13 that as heat pipe orientation changes from vertical position to horizontal heat pipe heat flux and efficiency decreases continuously. Fluid flow from condenser section to evaporator section is carried out due to two effects: capillary action and gravity assistance. As orientation decreases, gravity assistance also decreases which results in reduction in condensates traveling from condenser to evaporator, which affects on heat flux and efficiency. Similar trend of variation is found by Von der Fakultat et al. [30] which is mentioned in literature review section as shown in figure

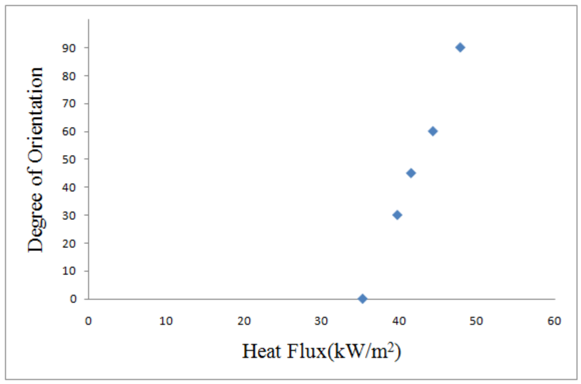


Figure 5.12: Degree of Orientation Vs. Heat Flux(kW/m^2)

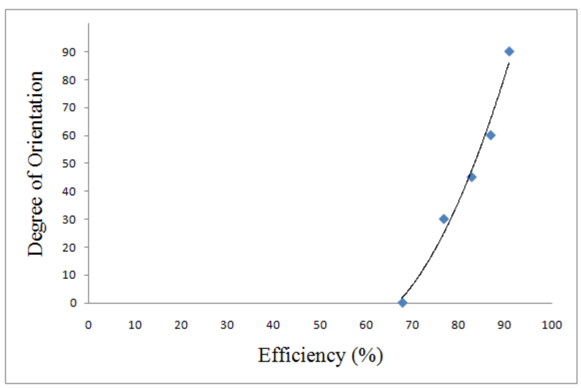


Figure 5.13: Degree of Orientation Vs. Efficiency(%)

5.4.1.5 Heat Transfer Coefficient:

It is seen from earlier section, carbon steel heat pipe with 2 layers of wick gives best result with economical viability. For this heat pipe overall heat transfer coefficient analysis calculations are as below:

$R = R_{ct} + R_{et} + R_{cw} + R_{ew}$

$R_{et} = \ln(d_o/d_i) / 2\pi L_e k_t$

$\quad = \ln(0.0254/0.0222) / 2\pi \times 0.5 \times 43$

$=9.96\times10^{-4}$w / mK

R_{ct}= ln (d_o/d_i) / $2\pi L_c k_t$

=ln (0.0254/0.0222) / $2\pi\times0.5\times43$

$=9.96\times10^{-4}$w / mK

k_{eq}=K $[(k_l+k_w) - (1\text{-}\varepsilon)(k_l\text{-}k_w)] / [(k_l+k_w) + (1\text{-}\varepsilon)(k_l\text{-}k_w)]$

$=1.44\times[(0.68+16) - (1\text{-}0.91)\times(0.68\text{-}16)] / [(0.68+16) - (1\text{-}0.91)\times(0.68\text{-}16)]$

=1.69

R_{ew}= ln (d_i/d_v) / $2\pi L_e k_{eq}$

=ln (0.0222/0.0198) / $2\pi\times0.5\times1.69$

=0.02154 w / mK

R_{ev}= ln (d_i/d_v) / $2\pi L_c k_{eq}$

=ln (0.0222/0.0198) / $2\pi\times0.5\times1.69$

=0.02154 w / mK

R= $9.96\times10^{-4}+ 9.96\times10^{-4}+ 0.02154 + 0.02154$

R= 0.045

U= $1/(R\times L) = 1/ (0.045\times0.5) =$ **44.44 W/m^{2}K**

Chapter 6

Conclusions and Future Scope

6.1 Conclusions

The following conclusions are derived from the present study

- Heat pipe developed and tested for 120^0C evaporator temperature is working satisfactorily & suitable for such waste heat recovery application.
- Time required for steady-state achievement is 90 minutes.
- Heat flux and heat transfer efficiency obtained for carbon steel as wall material 1 meter heat pipe

  a) with no wick is 31.4039 kW/ m^2 and 61.35 % respectively.

  b) with single layer stainless steel wick is 41.55 kW/ m^2 87.093 % respectively.

  c) double layer stainless steel wick is 47.49 kW/ m^2 90.76 % respectively.

  d) single layer copper wick is 43.82 kW/ m^2 89.82% respectively.

- Heat flux and heat transfer efficiency obtained for copper as wall material 1 meter heat pipe

  a) with single layer stainless steel wick is 44.64 kW/ m^2 90.93 % respectively.

- Experimental results of 2 meter heat pipe for same parameters i.e. working fluid, wall material, wick material same as 1 meter heat pipe, heat flux and heat transfer efficiency obtained

  a) with single layer stainless steel wick is 42.28 kW/ m^2 87.11 % respectively.

  b) double layer stainless steel wick is 46.18 kW/ m^2 90.91 % respectively.

- With changes in the orientation of the heat pipe, the performance also changes. Results shows that vertically oriented heat pipe gives highest heat transfer as gravitational force also supports the condensates to flow from condenser section to evaporator section. While the orientation of heat pipe changes from vertical to horizontal the performance decreases as reduction in gravitational force support. Hence it is advisable to orient the heat pipe vertically in absence of any practical constraints.

- Copper material either wick or wall gives better results, however the cost of copper is quite high as stainless steel and carbon steel. Based of functional & economic analysis, carbon steel wall material with stainless steel wick gives better results.

- By considering literature, obtain results are far high, acceptable and appreciable, which has been compared with given literature result.

Maximum % Error associated with different parameters, as calculated by statistical analysis, are as under:

Quantity	Statistical analysis % error (at 95% confidence level)
Heat flux	0.254%

6.2 Future Scope

The following work may be considered as further extension of the present project

- The performance of heat pipe may be evaluated for further operating temperature ranges.

- The heat pipe may be integrated with other systems and then its performance evaluated for practical applications.

- To know the effect of various parameters and applications, simulation can be carried out for optimization.

- The performance of a heat pipe designed for different working fluids (for high temperature application) can be studied.

Appendices

Appendix: I

Observations Data: case-1

	Reading-1	Reading-2	Rading-3	Avg.
Orientation	90^0	90^0	90^0	90^0
Material	C.S	C.S	C.S	C.S
Length (meter)	1	1	1	1
Wick	Nill	Nill	Nill	Nill
Wick Material	Nill	Nill	Nill	Nill
Input heater temp (°C)	120.9	119.8	120.5	120.4
Input Water Temp (°C)	21.4	22.1	22.6	22.03
Output water temp (°C)	57.7	57.9	58.2	57.93
Mass flow rate (LPH)	30	30	30	30
Total heat exchange surface area (m^2)	0.039878	0.039878	0.039878	0.039878
Energy gain by water (kJ / hr)	4552.02	4489.32	4464.24	4501.86
Energy input by heater (kJ / hr)	7546.7	7174.9	7298.3	7339.96
Total heat flux (kJ / m^2)	114148.65	112576.35	111947.43	112890.81
Total heat flux (kW / m^2)	31.75	31.31	31.14	31.40
Efficiency (%)	60.31	62.56	61.16	61.35

Observations Data: case-2

	Reading-1	Reading-2	Rading-3	Avg.
Orientation	90^0	90^0	90^0	90^0
Material	C.S	C.S	C.S	C.S
Length (meter)	1	1	1	1
Wick	1 layer	1 layer	1 layer	1 layer
Wick Material	S.S	S.S	S.S	S.S
Input heater temp (°C)	122.8	123.9	122.3	123
Input Water Temp (°C)	19.5	20.7	19.1	19.76
Output water temp (°C)	67.4	67.5	66.9	67.26
Mass flow rate (LPH)	30	30	30	30
Total heat exchange surface area (m^2)	0.039878	0.039878	0.039878	0.039878
Energy gain by water (kJ / hr)	6006.66	5868.72	5994.12	5956.51
Energy input by heater (kJ / hr)	6912.3	6832.6	6773.9	6839.6
Total heat flux (kJ / m^2)	150625.90	147166.85	150311.44	149368.07
Total heat flux (kW / m^2)	41.90	40.93	41.81	41.55
Efficiency (%)	86.89	85.89	88.48	87.09

Observations Data: case-3

	Reading-1	Reading-2	Rading-3	Avg.
Orientation	90^0	90^0	90^0	90^0
Material	C.S	C.S	C.S	C.S
Length (meter)	1	1	1	1
Wick	2 layer	2 layer	2 layer	2 layer
Wick Material	S.S	S.S	S.S	S.S
Input heater temp (°C)	121.7	120.3	121.9	121.3
Input Water Temp (°C)	21.9	22.3	21.7	21.96
Output water temp (°C)	76.7	76.8	75.3	76.26
Mass flow rate (LPH)	30	30	30	30
Total heat exchange surface area (m^2)	0.039878	0.039878	0.039878	0.039878
Energy gain by water (kJ / hr)	6871.92	6834.3	6721.44	6809.22
Energy input by heater (kJ / hr)	7568.1	7543.3	7394.3	7501.9
Total heat flux (kJ / m^2)	172323.58	171380.20	168550.07	170751.29
Total heat flux (kW / m^2)	47.93	47.67	46.88	47.49
Efficiency (%)	90.80	90.60	90.90	90.76

Observations Data: case-4

	Reading-1	Reading-2	Rading-3	Avg.
Orientation	90^0	90^0	90^0	90^0
Material	C.S	C.S	C.S	C.S
Length (meter)	1	1	1	1
Wick	1 layer	1 layer	1 layer	1 layer
Wick Material	Copper	Copper	Copper	Copper
Input heater temp (°C)	117.4	118.7	118.2	118.1
Input Water Temp (°C)	20.7	21.5	21.6	21.26
Output water temp (°C)	70.9	71.8	71.4	71.36
Mass flow rate (LPH)	30	30	30	30
Total heat exchange surface area (m^2)	0.039878	0.039878	0.039878	0.039878
Energy gain by water (kJ / hr)	6295.08	6307.62	6244.92	6282.54
Energy input by heater (kJ / hr)	7067.2	7009.3	6906.9	6994.4
Total heat flux (kJ / m^2)	157858.46	158172.92	156600.63	157544.00
Total heat flux (kW / m^2)	43.91	44.00	43.56	43.82
Efficiency (%)	89.07	89.98	90.41	89.82

Observations Data: case-5

	Reading-1	Reading-2	Rading-3	Avg.
Orientation	90^0	90^0	90^0	90^0
Material	C.S	C.S	C.S	C.S
Length (meter)	2	2	2	2
Wick	1 layer	1 layer	1 layer	1 layer
Wick Material	S.S	S.S	S.S	S.S
Input heater temp (°C)	156.7	156.3	157.3	156.76
Input Water Temp (°C)	22.6	23.2	23.7	23.16
Output water temp (°C)	70.3	71.7	72.5	71.5
Mass flow rate (LPH)	60	60	60	60
Total heat exchange surface area (m^2)	0.079756	0.079756	0.079756	0.079756
Energy gain by water (kJ / hr)	11963.16	12163.8	12239.04	12122
Energy input by heater (kJ / hr)	13753.23	13917.81	14082.11	13917.71
Total heat flux (kJ / m^2)	149996.99	152512.66	153456.04	151988.56
Total heat flux (kW / m^2)	41.72	42.42	42.68	42.28
Efficiency (%)	84.95	88.44	87.93	87.11

Observations Data: case-6

	Reading-1	Reading-2	Rading-3	Avg.
Orientation	90^0	90^0	90^0	90^0
Material	C.S	C.S	C.S	C.S
Length (meter)	2	2	2	2
Wick	2 layer	2 layer	2 layer	2 layer
Wick Material	S.S	S.S	S.S	S.S
Input heater temp (°C)	163.1	164.7	164.2	164
Input Water Temp (°C)	20.1	21.6	20.3	20.66
Output water temp (°C)	73.2	73.9	73.3	73.46
Mass flow rate (LPH)	60	60	60	60
Total heat exchange surface area (m^2)	0.079756	0.079756	0.079756	0.079756
Energy gain by water (kJ / hr)	13317.48	13116.84	13292.4	13242.24
Energy input by heater (kJ / hr)	14434.3	14693.6	14567	14564.96
Total heat flux (kJ / m^2)	166977.78	164462.10	166663.32	166034.49
Total heat flux (kW / m^2)	46.44	45.75	46.36	46.18
Efficiency (%)	90.63	90.87	91.25	90.91

Observations Data: case-7

	Reading-1	Reading-2	Rading-3	Avg.
Orientation	90^0	90^0	90^0	90^0
Wall Material	Copper	Copper	Copper	Copper
Length (meter)	1	1	1	1
Wick	1 layer	1 layer	1 layer	1 layer
Wick Material	S.S	S.S	S.S	S.S
Input heater temp (°C)	121.4	123.1	122.3	122.26
Input Water Temp (°C)	22.3	21.8	22.1	22.06
Output water temp (°C)	73.8	72.4	73.1	73.1
Mass flow rate (LPH)	30	30	30	30
Total heat exchange surface area (m^2)	0.039878	0.039878	0.039878	0.039878
Energy gain by water (kJ/hr)	6458.1	6345.24	6395.4	6399.58
Energy input by heater (kJ/hr)	6992.64	6973.93	7147.31	7037.96
Total heat flux (kJ/m^2)	161946.43	159116.30	160374.14	160478.96
Total heat flux (kW/m^2)	45.05	44.26	44.61	44.64
Efficiency (%)	90.35	90.74	91.70	90.93

Appendix: II

Statistical analysis

The following table consists statistical analysis of case 2: 1 meter length, 2 layer of stainless steel wick.

Xi(Heat Flux) (kw/m^2)	x_m (Avg)	x_i - x_m	$\sigma = [\frac{1}{(n-1)}] \times \Sigma(x_i - x_m)^2]^{1/2}$	$\triangle = \frac{z \times \sigma}{n}$	$(error = \frac{\Delta}{x_m}) \times 100$
47.93		0.4966	0.248333333	0.1622	0.342047318
46.88	47.43	-0.5533	0.276666667	0.1807	0.381072851
47.49		0.0566	0.028333333	0.0185	0.039025533
			Average error (%)	:	0.2540

Appendix: III

Uncertainty Analysis

Uncertainty analysis for heat flux has been presented by Kline and McClinLock [31]. The method is based on a careful specification of the uncertainties in the various primary experimental measurements. The uncertainty in the measurement by various instruments and detailed analysis is given below. For an example for one reading procedure to and uncertainty value in the heat flux is given below. Heat flux can be given as,

$$\frac{Q}{A} = \frac{\rho \times V_f \times C_p \times (T_o - T_i)}{\pi \times D \times L}$$

Differentiate above equation w.r.t T_o, T_i, V_f, D and L we get,

$$\frac{\partial(Q/A)}{\partial T_0} = \frac{Vf \times C_p \times \rho}{\pi \times D \times L} \times \omega T_0 = \frac{0.0000083 \times 4.18 \times 1000}{3.14 \times 0.025 \times 0.5} \times 0.1$$

$$\frac{\partial(Q/A)}{\partial T_i} = -\frac{Vf \times C_p \times \rho}{\pi \times D \times L} \times \omega T_i = -\frac{0.0000083 \times 4.18 \times 1000}{3.14 \times 0.025 \times 0.5} \times 0.1$$

$$\frac{\partial(Q/A)}{\partial Vf} = \frac{C_p \times \rho \times (T_o - T_i)}{\pi \times D \times L} \times \omega V_f = \frac{4.18 \times 1000 \times (76.26 - 21.96)}{3.14 \times 0.025 \times 0.5} \times 0.000000277$$

$$\frac{\partial(Q/A)}{\partial D} = -\frac{Vf \times C_p \times \rho \times (T_o - T_i)}{\pi \times D^2 \times L} \times \omega D = -\frac{0.0000083 \times 4.18 \times 1000 \times (76.26 - 21.96)}{3.14 \times 0.0254 \times 0.0254 \times 1} \times 0.00002$$

$$\frac{\partial(Q/A)}{\partial L} = -\frac{Vf \times C_p \times \rho \times (T_o - T_i)}{\pi \times D \times L^2} \times \omega L = \frac{0.0000083 \times 4.18 \times 1000 \times (76.26 - 21.96)}{3.14 * 0.0254} \times 0.0001$$

Uncertainty in derived quantities is due to uncertainty in primary measurements. Uncertainty equation for heat flux in percentage can be given as,

$$\omega\frac{Q}{A} = \frac{\sqrt{(\frac{\partial(Q/A)}{\partial T_0}\times\omega T_0)^2+(\frac{\partial(Q/A)}{\partial T_i}\times\omega T_i)^2+(\frac{\partial(Q/A)}{\partial Vf}\times\omega f)^2+(\frac{\partial(Q/A)}{\partial D}\times\omega D)^2+(\frac{\partial(Q/A)}{\partial L}\times\omega L)^2}}{\frac{Q}{A}}\times 100$$

where,

$\frac{Q}{A}$ = average heat flux = 47.49 kW/m^2

T_0 =water outlet temperature = 76.26 ^0C

T_i = water inlet temperature = 21.96 ^0C

V_f =volume of water = 0.0000083 m^3/s

D = heat pipe outer diameter = 25.4 mm

L = heat pipe condenser length= 0.5 meter

C_p = specific heat of water= 4.18 kJ / kg m^0C

ρ =density of water= 1000 kg/m^3

ωT_0 = uncertainty in outlet water temperature= 0.1 ^0C

ωT_i = uncertainty in inlet water temperature= 0.1 ^0C

ωV_f = uncertainty in water flow volume= 0.000000277 m^3/s

ωD = uncertainty in heat pipe diameter= 0.00002 m

ωL = uncertainty in heat pipe length= 0.0001 m

$\omega\frac{Q}{A}$ = uncertainty in heat flux

$$\omega\frac{Q}{A} = \frac{\sqrt{(0.088392357)^2+(-0.088392357)^2+(1.576603591)^2+(-0.018598894)^2+(0.00236206)^2}}{47.49952471}\times 100$$

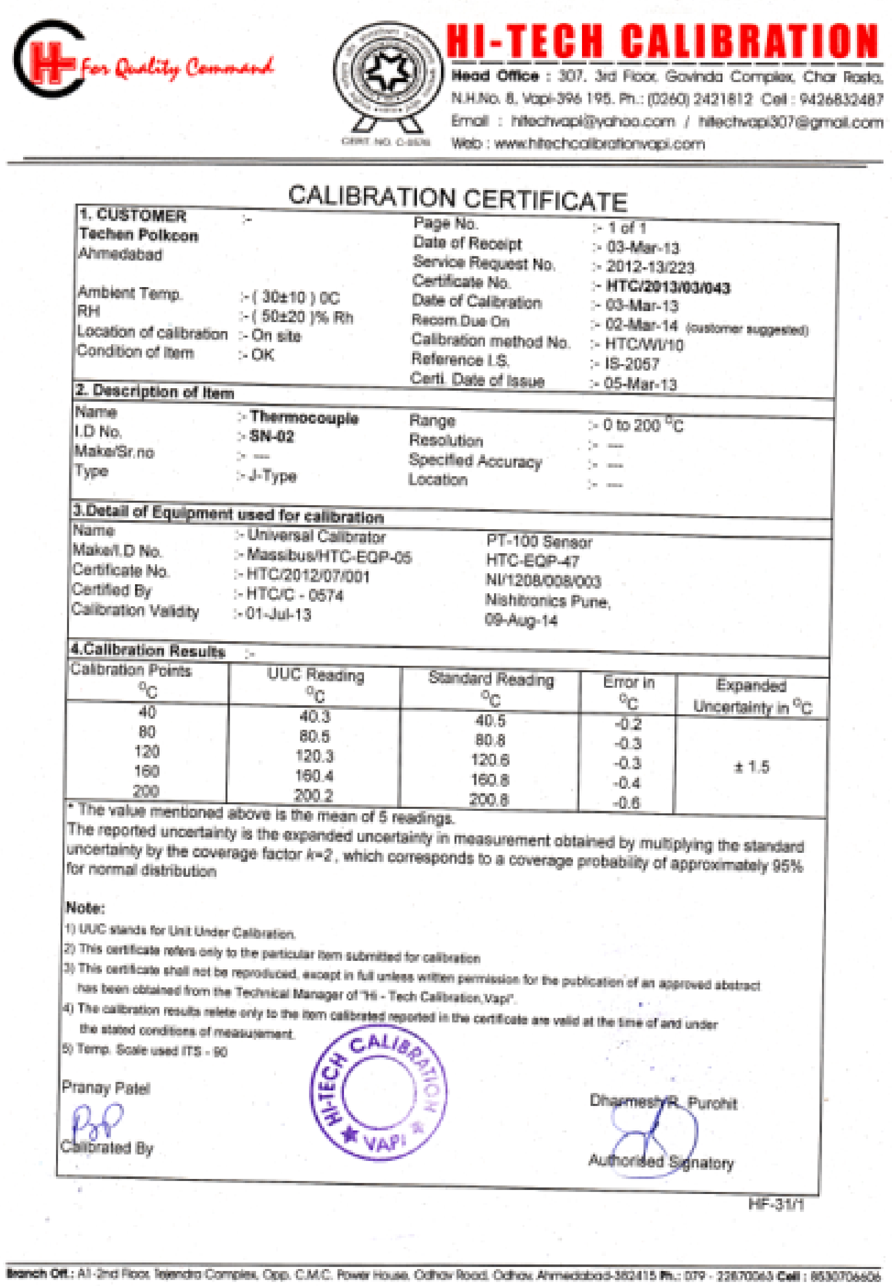
=3.329849464

uncertainty in heat flux = 47.49 $\pm$ 3.32% (kW / m^2)

Appendix: IV

Following Figure shows the calibration certificate of temperature sensors and controller which are used in the experiments:

Sensors:



HI-TECH CALIBRATION

For Quality Command

Head Office : 307, 3rd Floor, Govinda Complex, Char Rasta, N.H.No. 8, Vapi-396 195. Ph.: (0260) 2421812 Cell : 9426832487
Email : hitechvapi@yahoo.com / hitechvapi307@gmail.com
Web : www.hitechcalibrationvapi.com

CALIBRATION CERTIFICATE

1. CUSTOMER :-
Techen Polkcon
Ahmedabad

Ambient Temp.	:- (30±10) 0C	Page No.	:- 1 of 1
RH	:- (50±20)% Rh	Date of Receipt	:- 03-Mar-13
Location of calibration	:- On site	Service Request No.	:- 2012-13/223
Condition of Item	:- OK	Certificate No.	:- **HTC/2013/03/043**
		Date of Calibration	:- 03-Mar-13
		Recom.Due On	:- 02-Mar-14 (customer suggested)
		Calibration method No.	:- HTC/WI/10
		Reference I.S.	:- IS-2057
		Certi. Date of Issue	:- 05-Mar-13

2. Description of Item

Name	:- **Thermocouple**	Range	:- 0 to 200 °C
I.D No.	:- **SN-02**	Resolution	:- ---
Make/Sr.no	:- ---	Specified Accuracy	:- ---
Type	:- J-Type	Location	:- ---

3.Detail of Equipment used for calibration

Name	:- Universal Calibrator	PT-100 Sensor
Make/I.D No.	:- Massibus/HTC-EQP-05	HTC-EQP-47
Certificate No.	:- HTC/2012/07/001	NI/1208/008/003
Certified By	:- HTC/C - 0574	Nishitronics Pune,
Calibration Validity	:- 01-Jul-13	09-Aug-14

4.Calibration Results :-

Calibration Points °C	UUC Reading °C	Standard Reading °C	Error in °C	Expanded Uncertainty in °C
40	40.3	40.5	-0.2	± 1.5
80	80.5	80.8	-0.3	
120	120.3	120.6	-0.3	
160	160.4	160.8	-0.4	
200	200.2	200.8	-0.6	

* The value mentioned above is the mean of 5 readings.
The reported uncertainty is the expanded uncertainty in measurement obtained by multiplying the standard uncertainty by the coverage factor $k=2$, which corresponds to a coverage probability of approximately 95% for normal distribution

Note:
1) UUC stands for Unit Under Calibration.
2) This certificate refers only to the particular item submitted for calibration
3) This certificate shall not be reproduced, except in full unless written permission for the publication of an approved abstract has been obtained from the Technical Manager of "Hi - Tech Calibration,Vapi".
4) The calibration results relete only to the item calibrated reported in the certificate are valid at the time of and under the stated conditions of measurement.
5) Temp. Scale used ITS - 90

Pranay Patel
Calibrated By

HI-TECH CALIBRATION VAPI

Dharmesh R. Purohit
Authorised Signatory

HF-31/1

Branch Off.: A1-2nd Floor, Tejendra Complex, Opp. C.M.C. Power House, Odhav Road, Odhav, Ahmedabad-382415 **Ph.:** 079 - 22870063 **Cell** : 8530706606

For Quality Command

HI-TECH CALIBRATION

CERT. NO. C-0574

Head Office : 307, 3rd Floor, Govinda Complex, Char Rasta, N.H.No. 8, Vapi-396 195. Ph.: (0260) 2421812 Cell : 9426832487
Email : hitechvapi@yahoo.com / hitechvapi307@gmail.com
Web : www.hitechcalibrationvapi.com

CALIBRATION CERTIFICATE

1. CUSTOMER :-
Techen Polkcon
Ahmedabad

Ambient Temp.	:- (30±10) 0C	Page No.	:- 1 of 1
RH	:- (50±20)% Rh	Date of Receipt	:- 03-Mar-13
Location of calibration	:- On site	Service Request No.	:- 2012-13/223
Condition of Item	:- OK	Certificate No.	:- **HTC/2013/03/043**
		Date of Calibration	:- 03-Mar-13
		Recom.Due On	:- 02-Mar-14 (customer suggested)
		Calibration method No.	:- HTC/WI/10
		Reference I.S.	:- IS-2057
		Certi. Date of Issue	:- 05-Mar-13

2. Description of Item

Name	:- **Thermocouple**	Range	:- 0 to 200 °C
I.D No.	:- **SN-02**	Resolution	:- ---
Make/Sr.no	:- ---	Specified Accuracy	:- ---
Type	:- J-Type	Location	:- ---

3.Detail of Equipment used for calibration

Name	:- Universal Calibrator	PT-100 Sensor
Make/I.D No.	:- Massibus/HTC-EQP-05	HTC-EQP-47
Certificate No.	:- HTC/2012/07/001	NI/1208/008/003
Certified By	:- HTC/C - 0574	Nishitronics Pune,
Calibration Validity	:- 01-Jul-13	09-Aug-14

4.Calibration Results :-

Calibration Points °C	UUC Reading °C	Standard Reading °C	Error in °C	Expanded Uncertainty in °C
40	40.3	40.5	-0.2	
80	80.5	80.8	-0.3	
120	120.3	120.6	-0.3	± 1.5
160	160.4	160.8	-0.4	
200	200.2	200.8	-0.6	

* The value mentioned above is the mean of 5 readings.
The reported uncertainty is the expanded uncertainty in measurement obtained by multiplying the standard uncertainty by the coverage factor $k=2$, which corresponds to a coverage probability of approximately 95% for normal distribution

Note:
1) UUC stands for Unit Under Calibration.
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4) The calibration results relate only to the item calibrated reported in the certificate are valid at the time of and under the stated conditions of measurement.
5) Temp. Scale used ITS - 90

HI-TECH CALIBRATION VAPI

Pranay Patel
Calibrated By

Dharmesh R. Purohit
Authorised Signatory

HF-31/1

Branch Off.: A1-2nd Floor, Tejendra Complex, Opp. C.M.C. Power House, Odhav Road, Odhav, Ahmedabad-382415 **Ph.:** 079 - 22870063 **Cell :** 8530706606

Controller:

digicon

Automation Pvt. Ltd.

Telefax: 22902955, 22902965 Phone: 9725002238
Email: info@digicon.in / sales@digicon.in

PLOT NO - 179, ROAD NO 4 , G. I. D. C. , KATHWADA, AHMEDABAD, PIN – 382430.

NOVEMBER 24, 2012
REF NO : DAPL/CERT/1090/12-13

TECHEN POLKCON.
Plot no:-86,Mahalaxmi Estate,
Post:- Lyava, Tal:- Sanand,
Ahmedabad – 382176.

KIND ATTN: MR. KISHOR VYAS

Re: J & D PID Controller Test Certificate & Confirmation of Test / calibration

Dear Sir,

This is to inform you in above said subject matter. As per your request Digicon Engineer has Tested all the below J & D after its fitment in a respective machine.

No.	Product Description	Order Code	Product Sr. No.	Qty
1	I/P PT-100, O/P SSR	J & D DB 5040	SP12080980162	1

For your kind & immediate perusal.

Yours truly,
For Digicon Automation Pvt. Ltd.

Authorized Signatory

digicon

Automation Pvt. Ltd.

Telefax: 22902955, 22902965 Phone: 9725002238
Email: info@digicon.in / sales@digicon.in

PLOT NO - 179, ROAD NO 4 , G. I. D. C. , KATHWADA, AHMEDABAD, PIN – 382430.

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Plot no:-86,Mahalaxmi Estate,
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Dear Sir,

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No.	Product Description	Order Code	Product Sr. No.	Qty
1	I/P PT-100, O/P SSR	J & D DB 5040	SP12080980162	1

For your kind & immediate perusal.

Yours truly,
For Digicon Automation Pvt. Ltd.

DIGICON AUTOMATION PVT. LTD.

Authorized Signatory

Appendix: V

The calibration certificate of rota-meter which is used in experiment is shown in following Figure:

MAXON ENGINEERS

[illegible] Road, Ahmedabad-[illegible] (M) Phone [illegible]
MFGRS.THERMOWELL,MANOMETER,SYPHONE,SS &BRASS FITTING&ALL KIND OF PRECISION JOB WORK
E-mail : maxon_engineers@yahoomail.com

TEST CERTIFICATE

NAME OF PARTY	:	
MASTER FLOWMETER MAKE	:	ABBKENT TAILOR
CERTIFICATE NO.	:	FCRI/HPATF/01-02K/00112
DUE DATE OF CALIBRATION	:	6-10-2013
INSTRUMENTS UNDER TEST	:	ACRYLIC BODY ROTAMETER
ORDER NO.	:	BY-MAIL
SERVICE	:	Water
MODEL NO	:	SF/ABR/206
MAKE	:	
SP. GRAVITY/DENSITY.	:	1.25 KG/NM3
CONN	:	3/8"BSP
FLOW RANGE	:	0-60 LPH
INSTRUMENTS NO.	:	SF/ABR//11-12
CERTIFICTE NO	:	SF/ABR/1-445
FLOAT	:	SS304
WETTED PARTS MOC.	:	SS304
DATE OF CALIBRATION	:	06 -10-2012
RANGE ABILITY	:	10 TO100%

SR.NO.	MASTER FLOWMETER LPM	ACTUAL RANGE LPM	ACRYLIC BODY ROTAMETER DIFF. ERROR IN LPH
1.	25	25.375	+0.375
2.	20	19.700	- 0.300
3	15	15.225	+0.225
4.	10	9.850	-0.150
5	5	5.075	+0.075
6.	2.5	2.4625	-0.0375

Remarks the instruments founds of the range of Rotamameter +- 1.5 % full scale reading

Checked by
Tapan Patel

For, Maxon Engineers

Appendix: IV

Expenditure details

All expenditures of this project are tried to conclude as below table. All amounts are estimated and approximate.

Sr. No	Item name	cost(approx.)
1	Pipe material	7000/-
2	Pickling and cleaning	4700/-
3	Wick material	8000/-
4	Vaccum pump	19000/-
5	Pressure pump	8000/-
6	wick insertion arrangement	4000/-
7	Pipe manufacturing facility	12000/-
8	Chemicals and inert gas	4000/-
9	Electrical Heaters	12000/-
10	Measuring instruments	13000/-
11	Insulation material	900/-
12	PID controller	9000/-
13	Temperature indicators	7000/-
14	Experiment structure	9000/-
15	Labor and miscellaneous	8000/-
	Total	1,24,600/-

Bibliography

[1] Yunus a. cengel, "Heat transfer a practical approch: 2^{nd}edition", Tata McGrawhill.

[2] "Heat pipe design handbook by national aeronautics and space administration goddard space flight center vol 1", 1979.

[3] "Heat pipes theory, design and application" by david reay and peter kew, p 1-397, 2001.

[4] A.Faghri, "Heat pipe science and technology", 1995.

[5] G.Q.Lu K.C.Leong and C.Y.Liu, "Characterization of sintered copper wicks use in heat pipes", 1997.

[6] A.P.Bhatt, "Design and analysis of an axially grooved heat pipe", M tech thesis, 2010.

[7] Waste heat recovery system, "Fuel Economy in furnaces and Waste heat recovery - PCRA" .2007.

[8] H.N. Chaudhry et al., "A review of heat pipe systems for heat recovery and renewable energy applications", Renewable and Sustainable Energy Reviews 16, 2012; p2249– 2259.

[9] Y.H. Yau, M. Ahmadzadehtalatapeh, "A review on the application of horizontal heat pipe heat exchangers in air-conditioning systems in the tropics", Applied Thermal Engineering 30, 2010; p77–84.

[10] X. Yang et al, "Recent developments of lightweight, high performance heat pipes", Applied Thermal Engineering 33-34, 2011; p1-14.

[11] Y. Zhao et al, "Experimental study on a cryogenic loop heat pipe with high heat capacity", International Journal of Heat and Mass Transfer 54, 2010; p3304–3308.

[12] P. Nemec et al, "Mathematical model for heat transfer limitations of heat pipe", Mathematical and Computer Modeling, 2009.

[13] S.C. Wong, Y.H. Kao, "Visualization and performance measurement of operating mesh-wicked heat pipes", International Journal of Heat and Mass Transfer 51, 2008; p4249–4259.

[14] J.C. Wang, "L-type heat pipes application in electronic cooling system", International Journal of Thermal Sciences 50, 2011; p97-105.

[15] X. Liu et al, "Dynamic charging characteristics modeling of heat storage device with heat pipe", Applied Thermal Engineering 31, 2011; p2902-2908.

[16] D. Liu et al., "Modeling and experimental investigation of looped separate heat pipe as waste heat recovery facility", Applied Thermal Engineering 26, 2006; p2433–2441.

[17] C. Ren, "Parametric effects on heat transfer in loop heat pipe's wick", International Journal of Heat and Mass Transfer 54, 2011; p3987–3999.

[18] R. Kempers et al, "Effect of number of mesh layers and fluid loading on the performance of screen mesh wicked heat pipes", Applied Thermal Engineering 26, 2005; p589–595.

[19] S. Lips et al, "Combined effects of the filling ratio and the vapour space thickness on the performance of a flat plate heat pipe", International Journal of Heat and Mass Transfer 53, 2010; p 694–702.

[20] R. Singh et al, "Operational characteristics of the miniature loop heat pipe with non-condensable Gases", International Journal of Heat and Mass Transfer 53, 2001; p3471–3482.

[21] E. Azad, "Theoretical and experimental investigation of heat pipe solar collector", Experimental Thermal and Fluid Science 32, 2007; p1666–1672.

[22] S.C. Wong, C.W. Chen, "Visualization and evaporator resistance measurement for a groove-wicked flat-plate heat pipe", International Journal of Heat and Mass Transfer 55, 2012; p2229–2234.

[23] H. Hagens et al, "Air heat exchangers with long heat pipes: Experiments and predictions", Applied Thermal Engineering 27, 2007; p2426–2434.

[24] C.C. Yeh et al, "Heat transfer analysis of a loop heat pipe with biporous wicks", International Journal of Heat and Mass Transfer 52, 2009; p4426–4434.

[25] F. Yang et al., "Waste heat recovery using heat pipe heat exchanger for heating automobile using exhaust gas", Applied Thermal Engineering 23, 2003; p367–372.

[26] C. Silvrstain, "Design and technology of heat pipe for cooling and heat exchange", Research paper, 2002.

[27] S. Khandekar et al., "Understanding operational regimes of closed loop pulsating heat pipes: an experimental study", Applied thermal engineering 23, 2003; p707-719.

[28] Y. Wang, K. Vafai, "An experimental investigation of the thermal performance of an asymmetrical at plate heat pipe", 2000.

[29] B Sivaraman, N Krishna Mohan, "Experimental analysis of heat pipe with different L/di ratio of heat pipe", 2005.

[30] V. Fakultät et al., "Thermo-hydrodynamics of Closed Loop Pulsuating Heat Pipe", 2004.

[31] Richard , Donald, "Theory and design of mechanical measurements", 3^{rd} edition, 2004.