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Heat Transfer Analysis of Hybrid Loop Heat Pipe System

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ABSTRACT

Hybrid Loop Heat Pipe System (HLHPS) is an advanced next generation Loop Heat Pipe system with multiple main evaporators and one small secondary evaporator adjacent to the only reservoir in the system. The HLHPS reservoir contains both liquid and vapor phases of the working fluid and, being the largest volume component, virtually dictates the level of the absolute pressure inside HLHPS.In this thesis, Hybrid Loop Heat Pipe System is and modelled in 3D modelling software Pro/Engineer. Heat transfer characteristics of HLHPS are determined by CFD and thermal analysis. CFD analysis is done to determine the heat transfer rate, pressure drop, mass flow rate, heat transfer coefficient different working fluids R134, R22, R34 and R124 . Thermal analysis is done to determine heat transfer rate and temperature distribution.CFD and thermal analysis is done in Ansys.

Keywords—*HLHPS, capillary wick,temperature control*

I. INTRODUCTION

Hybrid Loop Heat Pipe System (HLHPS) is an advanced next generation Loop Heat Pipe system with multiple main evaporators and one small secondary evaporator adjacent to the only reservoir in the system. HLHPS operational concepts are introduced by US Patents 6,889,754 and 7,004,240. It is a ―hybrid‖ in the sense of combining the best features typical for Loop Heat Pipes (LHPs) and Capillary Pumped Loops (CPLs). A HLHPS and a CPL with multiple evaporators have only one reservoir, which contains two-phase working fluid (liquid and saturated vapor). A HLHPS reservoir is hydraulically linked to the secondary evaporator with a secondary wick, similar to LHPs. This allows HLHPS to withstand severe fluid transients and tolerate vapor phase in the liquid flow returning to the reservoir from the condenser. The secondary evaporator with reservoir also provides for continuous sweepage liquid flow through the liquid cores of the main evaporators. The HLHPS operational concepts were described previously to a certain extent, for example by Bugby (2007) and by Wrenn and Wolf (2008), and are re-introduced below for the specific HLPS configuration modeled and presented in this paper.

II. HLHPS OPERATIONAL PRINCIPLES

A HLHPS schematic is provided by Fig. 1. Several main capillary evaporators acquire thermal energy from the heat dissipating components by vaporizing the working fluid at the interface between the porous primary wick and heated evaporator wall. Due to the capillary pressure developed by liquid-vapor meniscii in the primary wick, the vapor flows through the vapor transport line into the condenser bonded to the condenser plate, which is cooled by a heat sink. The liquid coming out of the condenser returns to the inner cores of the main evaporators and filters through the porous wick structure to its outer surface with the mass flow rate corresponding to that of the vapor flow. The liquid return is directed into the inner evaporator core through a small diameter tubing, which assists any possible vapor bubbles in the inner core to come out of the evaporator. The main evaporators can be plumbed either in parallel or in series via the liquid supply. Main evaporators in

Figure 1 are connected in series in the sense that the liquid coming out of the condenser flows to the first evaporator, where it is only partially consumed due to the vaporization on the outer surface of the primary wick, while the rest of the liquid comes out of the first evaporator core and flows into the second evaporator core and subsequently into the next evaporator core. Highpower HLHPS with multiple evaporators would benefit from configuring main evaporators in parallel via the liquid supply. The distinguishing feature of a HLHPS is that there is fluid flow coming out of the last main evaporator core, usually called sweepage flow‖ in the literature. Its important function is to sweep any possible vapor bubbles out of the liquid cores of the evaporators. This sweepage flow is created by heat loading a small secondary evaporator, constructed similarly to LHP evaporators, which generate some additional vapor flowinto the condenser which continues as additional liquid flow exiting the condenser. The sweepage flow passes through all the evaporator cores along with the evaporating liquid feed, and ultimately exits from the last main evaporator and enters the subcooler shown in Fig. 1 adjacent to the condenser plate. The liquid coming out of the subcooler flows into the inner core of the secondary evaporator, which is hydraulically coupled to the fluid inside the reservoir via a secondary wick, thereby completing the sweepage loop. The HLHPS major operational principles can be briefly summarized as follows:

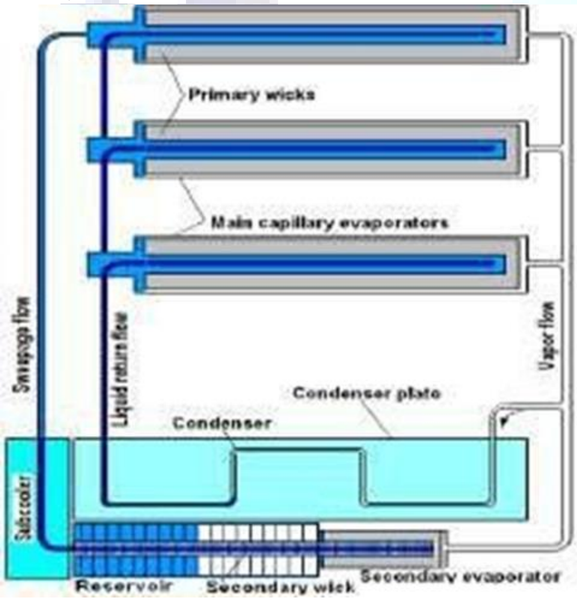


Fig 1: HLHPS schematic with three main in- series evaporators

The HLHPS reservoir contains both liquid and vapor phases of the working fluid and, being the largest volume component, virtually dictates the level of the absolute pressure inside HLHPS

The capillary pressure developed by the primary wicks provides the phase control and self-regulation in the system allowing the main evaporators to be subjected to different powers, locations (or distances), and elevations (in gravity), within the limits of the maximum capillary potential.

The secondary evaporator is always kept heat loaded, especially prior to heat loading of the main evaporators, so that the liquid flow through the liquid core of the main evaporators ensures the primary wicks are fully saturated with liquid and thus ready to be powered.

The liquid coming out of the condenser and flowing into the evaporator inner cores should be single-phase liquid preferably subcooled versus the saturation temperature in order to provide advantageous conditions for the main evaporators to be fully saturated with liquid and operating in the most reliable mode.

To guarantee reliable performance under severe transient conditions, it is desirable (although not always necessary) for HLHPS to be operated inside the design parameters envelope where the sweepage flow coming out of the last main evaporator and into the subcooler is a single- phase liquid.

Applying some minor heat load to the reservoir elevates the vapor temperature in HLHPs, which results in a shorter condensation length and colder liquid coming out of the condenser thus helping to avoid vapor phase in the liquid return line and the sweepage flow.

Powering small heaters on the liquid return lines during the time when the heat load on the main evaporators is zero but the secondary evaporator is heat loaded can be useful to prevent the main evaporator‘s temperatures from dropping to the sink temperature.

The HLHPS operational principles above are based on observations derived from the experimental HLHPS testing discussed below.

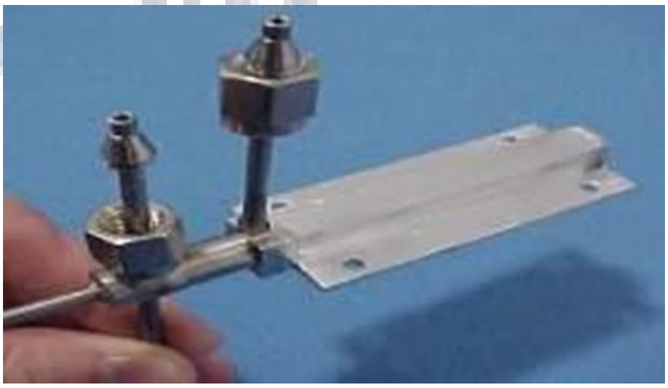


Fig 2 : Evaporator with porous Teflon wick

III. HLHPS MODELLING APPROACH

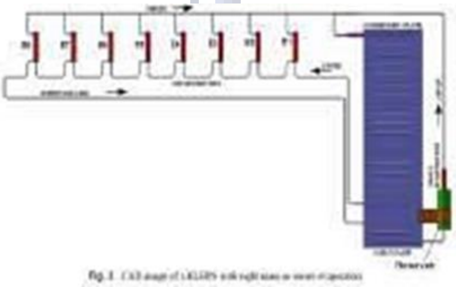
The configuration in the Thermal DesktopTM image in Figure 4 is consistent with that in Figure 3 while additionally communicating that the HLHPS components can be tightly packaged into a specific system volume where the components can have different orientations and the transport lines can be shaped in any way to comply with the application. Gravity effects on the pressure distribution are accounted for by the model. Following the modeling approach pioneered by Cullimore and Bauman (2000), which was also utilized by Khrustalev (2010a), the modeling philosophy used herein is to represent the HLHPS components with a minimal amount of details, which is still sufficient to capture transients in temperatures, pressures, mass flow rates, and flow qualities across the HLHPS. Consequent analysis of the model predictions leads to a better understanding of the thermal-fluid mechanisms that limit the HLHPS performance envelope. Some details of the modeling technique are discussed below.

Figure 5 illustrates a simple but adequate approach to modeling HLHPS components such as reservoir, capillary evaporator, and condenser. Fluid lumps and solid nodes essential for the model can be seen in the figure and are explained below. Fluid lump FF2701 is a two-phase tank‖ representing the reservoir with an internal volume of 115 cm3. Its quality varies in time depending on the mass of fluid contained in the condenser; however this ―tank‖ always has both saturated vapor and liquid inside it, due to the design intent. The tank‖ receives either single-phase liquid or two-phase fluid coming out of the subcooler through a path‖ from the liquid return fluid lump FF1446. This fluid lump is thermally tied to node LP1 representing cylindrical surface and metal wall

of the reservoir, which can be radiatively exchanging thermal energy with a heat sink or environment. An arithmetic‖ (zero heat capacity) node LP55 linked to node LP1 is added for convenience and is programmed to read the instant heat flow rate on the reservoir. Node LP55 is also used to apply a minor heat load to the reservoir in order to keep it at a desired temperature level. Node LP1 is shown with a conductor leading to the condenser plate, which represents a thermal strap between the reservoir and condenser plate (visible in Figure 3), however the value of this conductor was conveniently set to be zero for the modeling results presented . Fluid lump FF2701 serves as the liquid supply for the Thermal Desktop TM macro CAPPMP representing the secondary capillary evaporator. Fluid lump FF2709 is the vapor-generating part of the CAPPMP evaporator and LP2 is the vapor saturation temperature node on the evaporator side. Lump FF2701 is linked to the evaporator saturation temperature‖ node LP2 via a thermal tie‖ FF2502. Thermal conductance of this tie is adjusted in Variables 1‖ at each time step of the transient solution to maintain the proper value of the heat leak flow rate, QBC, which is defined by Equation (1) and explained below. Heat loading of the evaporator outer surface generates vapor flow into the vapor transport line with the mass flow rate mv and simultaneously, some possible vaporization on the inner surface of the primary wick with the mass flow rate mBC and corresponding vapor flow into the reservoir. This possible vapor flow into the reservoir is due to the heat leak flow rate through the cylindrical wall of the primary wick, which is called back conduction‖ and is proportional to the temperature drop across the cylindrical wall of the primary wick. If the fluid in the inner evaporator core is two-phase, the temperature drop is calculated in the model based on the pressure drop between the saturation vapor pressure inside the axial grooves on the outer surface of the primary wick and the saturation vapor pressure inside the twophase reservoir. This pressure drop, ΔPV-R, is due to the viscous vapor and liquid flow through the transport lines and all other fluid passages as well as the liquid head in the field of gravity. Note that ΔPV-R does not include the viscous pressure drop for the liquid filtering through the porous walls of the primary wick from the inner surface to the outer surface were vapor generation takes place. Thus the main component of the back conduction‖ heat flow rate, QBC, between the evaporator and the reservoir is based on the pressure drop between these two components, ΔPV-R , and depends on the steepness of the saturation curve, dP/dTsat , as expressed by Equation (1):

$$QBC = [1/Cwick+1/(\pi DiLeh)f]\ -1\Delta PV-R(\ dP/dTsat)-1 + Cwall(Tev - Tres) \quad (1)$$

Equation (1) includes the primary wick thermal conductance, Cwick, as well as the term reflecting heat transfer from the inner (cylindrical) surface of the primary wick to the fluid flowing through the evaporator core, hf. The highest value of the back conduction corresponds to the situation with the vapor (two-phase fluid) filling most of the evaporator inner core where the heat transfer coefficient characterizing evaporation of the working fluid from the inner surface of the primary wick is relatively high. The lowest possible value of the back conduction is for the situation with the subcooled (or slightly superheated) single-phase liquid filling the inner evaporator core, where the heat transfer coefficient between the single-phase liquid and the inner surface of the primary wick is relatively low. The last term in Equation (1), which usually can be neglected in comparison with the first term, is related to the conductance of the metal connector, Cwall , welded to the reservoir on one end and to the evaporator on the other end. If the major term in Eq. (1) is based on the pressure drop rather than on the temperature drop, accounting for the back conduction in a Thermal DesktopTM model involves some additional programming.



The primary wick thermal conductance, Cwick , is not exactly a constant, since it can be affected by the liquid flowing through the porous structure from the inner surface of the primary wick to its outer surface. However, for the purpose of the presented HLHPS model, Cwick , is assumed to be a constant defined purely by conduction in the porous material.

If the fluid in the evaporator core is single-phase, the back conduction heat flow rate in Eq. 1 is based on the temperature difference between the saturated vapor in the evaporator and the reservoir. Such description of the back conduction is sufficient for the HLHPS model described in this paper due to the two reasons: (1) the reservoir is temperature controlled by applying a small portioned heat load to its wall, where the exact value of this heat load is not important for the qualitative predictions, and (2) back conduction in the main evaporators is very low due to low conductivity of the porous Teflon and large thermal resistance associated with the term reflecting convective heat transfer from the inner (cylindrical) surface of the primary wick to the singlephase liquid inside the evaporator core, hf. The evaporator saturation node LP2 is linked through a conductor to the evaporator wall node LP10, which represents the heat capacity and mass-averaged temperature of the evaporator cylindrical body and evaporator plate.

The conductor between these two nodes is assigned a conductance value equal to the experimental value of the evaporator thermal conductance. Fluid lump FF2709 stands for the vapor volume of the axial grooves in the evaporator. Fluid path FF1112 has a duplication factor‖ equal to the number of the axial grooves in the evaporator and leads the vapor generated by the evaporator to the vapor transport line. As soon as a heat load is applied to the evaporator wall (node LP10) and two-phase conditions exist, vapor is generated in the fluid lump

FF2709 with the mass flow rate equal to the consumption of liquid coming out of the reservoir (fluid lump FF2701) into the evaporator. This phase change process is modeled using CAPPMP macro available in Thermal DesktopTM, which does not have graphic visibility in Figure5.

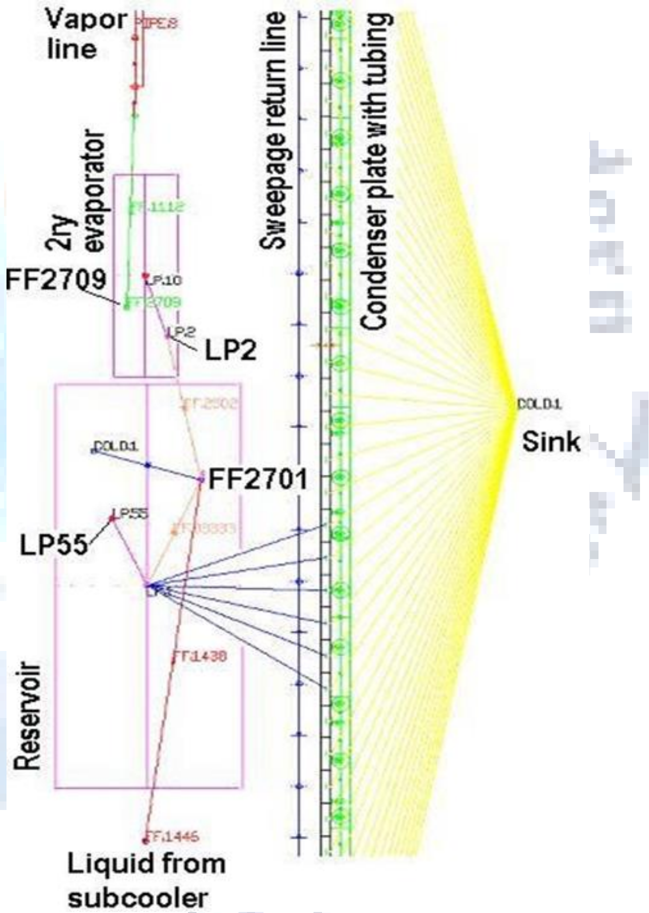


Fig. 5 : Fragment of the HLHPS model with Reservoir, Secondary evaporator and condenser

The secondary wick is not explicitly present in this model, since it is assumed that its design guarantees the supply of sufficient liquid to the primary wick under any transient conditions, while the secondary evaporator heat load is kept at a relatively low level. Thermal DesktopTM (Sinda/Fluint) LHP model shown in Figures 4 and 5 simulate transient circulation of the two-phase working fluid in the LHP components (condenser, evaporators, reservoir, and transport lines) and the temperature distribution across the LHP components including the condenser plate and subcooler. There are typically more than 500 fluid lumps (and also 500 corresponding nodes, paths, and ties) in a LHP model in order to present the condenser lines and other components in sufficient detail. The flow quality for each fluid lump is calculated by Sinda/Fluint based on the pressures, temperatures, and lump energy balance.

The condenser and subcooler tubing (―pipes with walls‖) is bonded to the condenser plate using contactors‖ accounting for the temperature drop corresponding to the bond design and material. The outlet end of the condenser tubing is typically filled with the liquid subcooled below the saturation temperature, while the beginning of the condenser tubing contains both vapor and liquid (condensate) phases. The total fluid mass contained in the condenser varies in time under transient regimes as well as the fluid mass in the reservoir.

The reservoir heat capacity is thus dependent on the current fluid distribution between the HLHPS components and is found as a result of solving the mass conservation equations by the model. The HLHPS model simultaneously solves the flow momentum, energy, and mass conservation equations for the two-phase and single-phase fluid flow for each fluid lump and path separately as well as for the entire fluid submodel. It includes all three heat transfer mechanisms: convective, conductive (conduction in the condenser plate and subcooler plate), and radiative (radiation to the heat sinks), and also accounts for the phase change heat transfer. Since Thermal DesktopTM package includes a thorough description of the conservation equations used to describe two-phase systems with all heat transfer mechanisms, these equations are not repeated in .The modeled HLHPS parameters are summarized in Tables 1 through 3. The modeled HLHPS is of the same scale that the ST-8 system described by Bugby (2007) and also uses porous Teflon as the primary wick material.

IV. LIMITATIONS

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 at which heat pipes work.

When heated above a certain temperature, all of the working fluid in the heat pipe vaporizes and the condensation process ceases; in such conditions, the heat pipe's thermal conductivity is effectively reduced to the heat conduction properties of its solid metal casing alone. As most heat pipes are constructed of copper (a metal with high heat conductivity), an overheated heatpipe will generally continue to conduct heat at around 1/80 of the original flux via conduction only rather than conduction and evaporation. In addition, below a certain temperature, the working fluid will not undergo phase change, and the thermal conductivityis reduced to that of the solid metal casing. One of the key criteria for selecting a working fluid is the desired operational temperature range of the application. The lower temperature limit typically occurs a few degrees above the freezing point of the working fluid.Most manufacturers cannot make a traditional heat pipe smaller than 3 mm in diameter due to material limitations (though 1.85 mm thin sheets with embedded, flattened heat pipes can be fabricated, as well 1.0 mm thin vapor chambers).

Experiments have been conducted with micro heat pipes, which use piping with sharp edges, such as triangular or rhombus-like tubing. In these cases, the Fig.7 : Final 3-D sketch of evaporato sharp edges transfer the fluid through capillary action, and no wick is necessary.

IV. MODELLING OF HLHPS WITH MULTIPLE EVAPORATORS IN CREO 2.0

For modeling of hybrid loop heat pipe systems with multiple evaporators the reference is taken from transient modeling of hybrid loop heat pip systems With multiple evaporators by Dmitry Khrustalev.

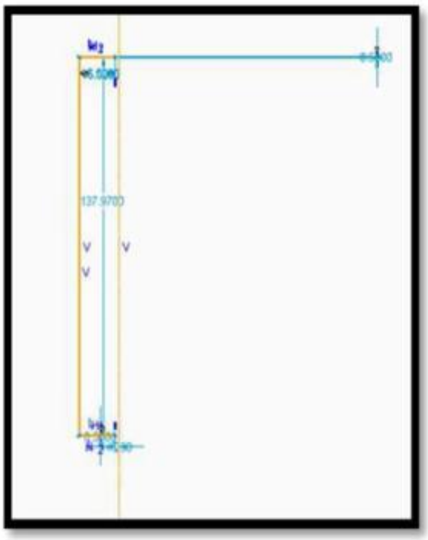


Fig.6 : 2-D sketch of single evaporator

Fig.7 : Final 3-D sketch of evaporator

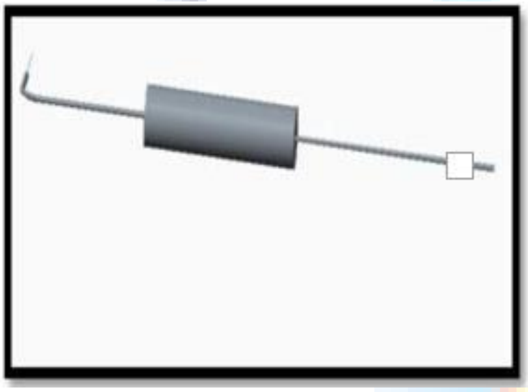


Fig.8 : Single evaporator with a tube

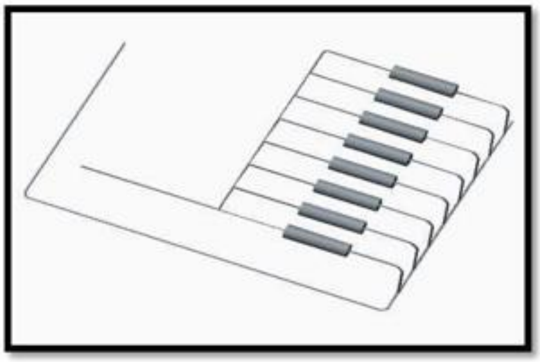


Fig. 9 : Multi-evaporators with tubes

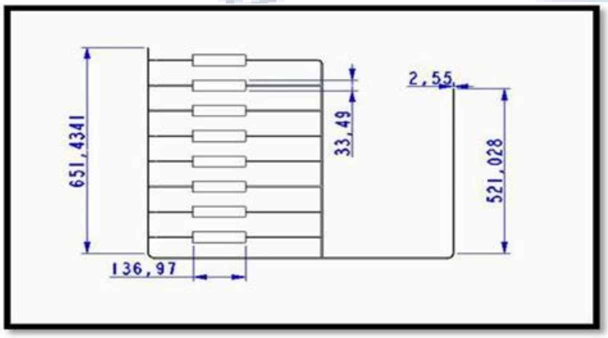


Fig. 10 : 2-D dtafting of multi-evaporator

VIII. ANALYSIS OF HLHPS WITH MULTIPLE EVAPORATORS BOUNDARY CONDITIONS

The hybrid loop heat pipe systems with multiple evaporators is analyzed for a mass flow rate 1 kg/s The value is taken from the Standard book (book name is plate fin tube condenser and evaporator by Monifa fela wright)

The material properties are specified in the below table which are taken from website www.matweb.com

MATERIAL	Thermal conductivity (W/m-k)
Aluminum	210
copper	385

Material Properties

CFD ANALYSIS WITH FLUID R-132

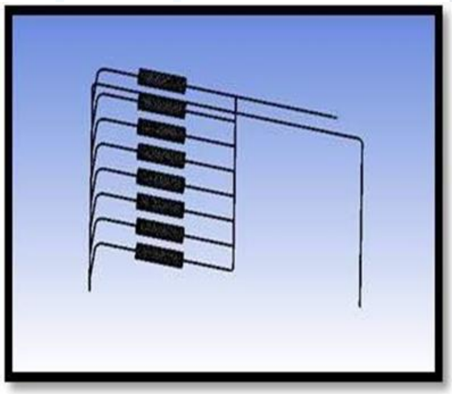
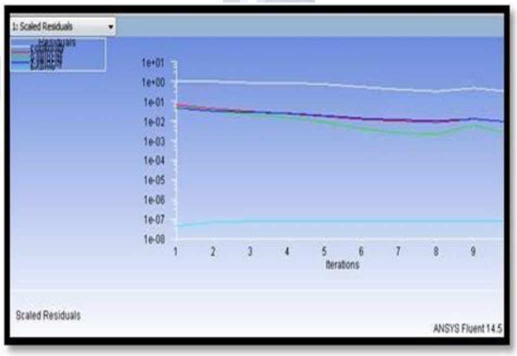


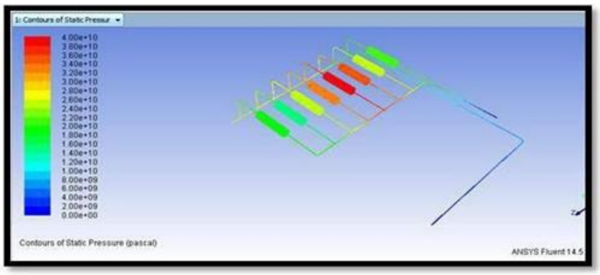
Fig.11 : Meshed model of HLHPS imported from CREO 2.0

Fig. 12 : Representation of boundary conditions of fluids

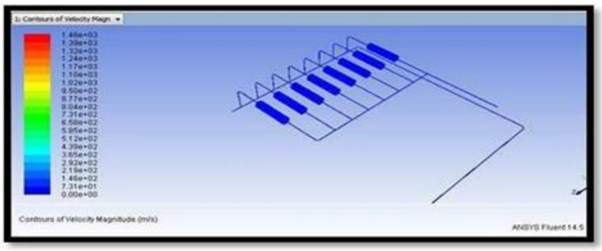
Iterations



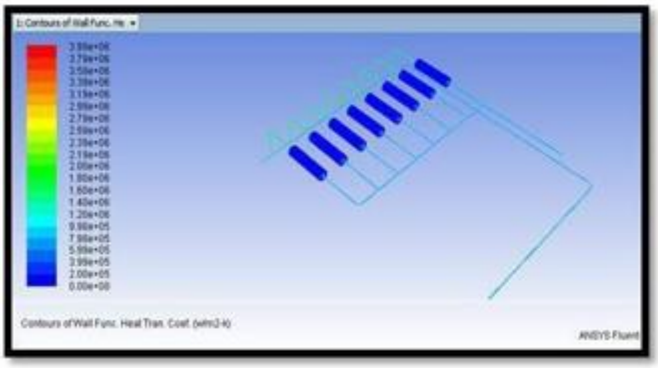
Pressure



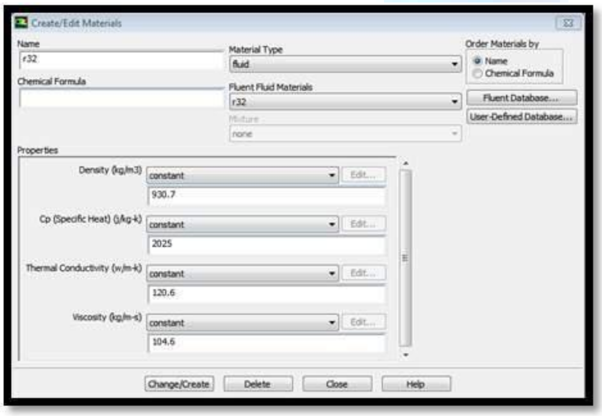
Velocity



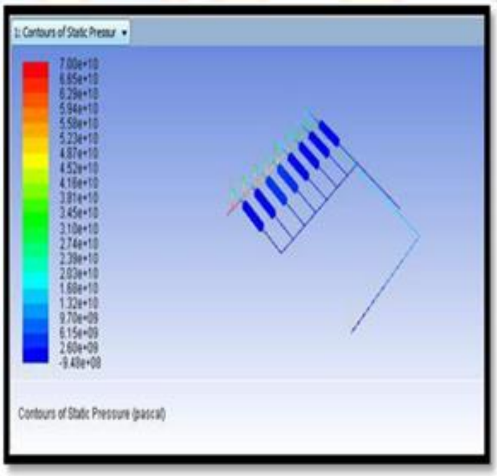
Heat transfer coefficient



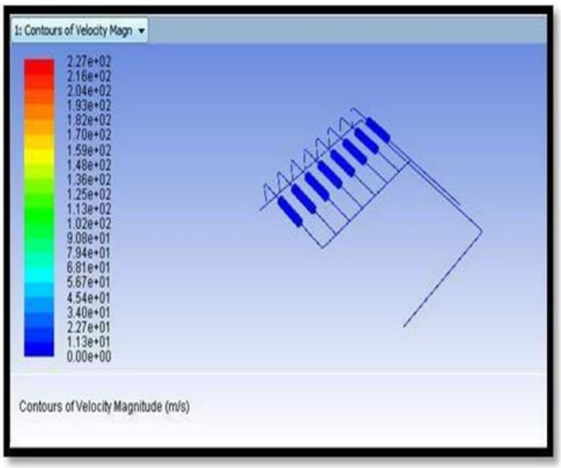
CFD ANALYSIS WITH FLUID R-32



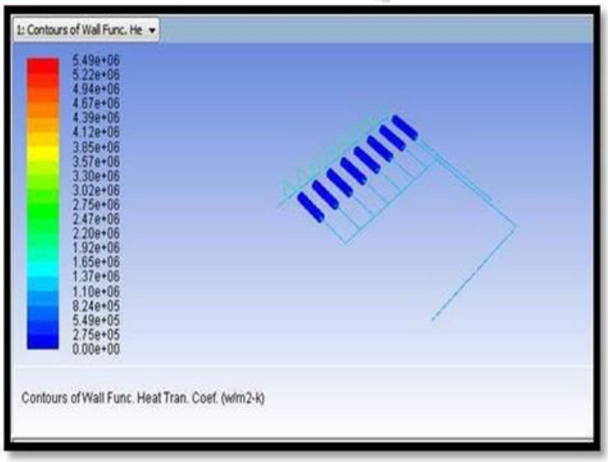
Pressure



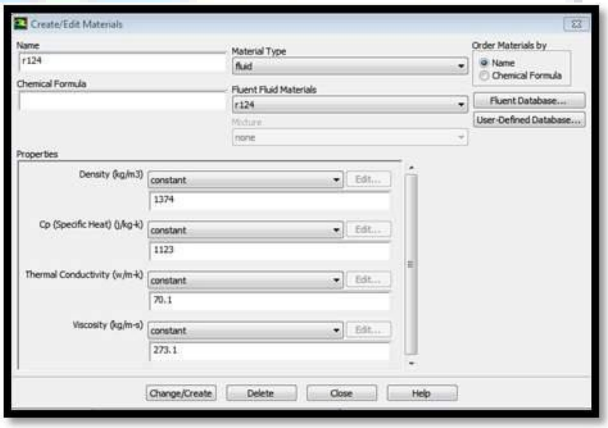
Velocity



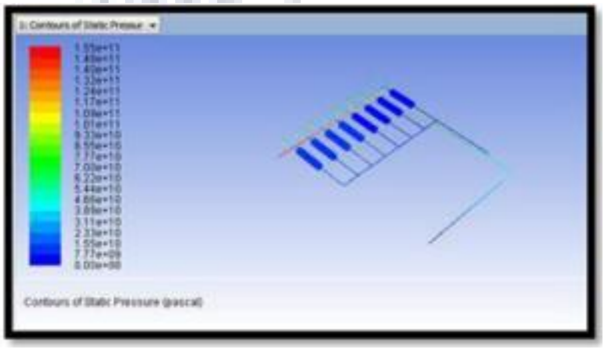
Heat Transfer Coefficient



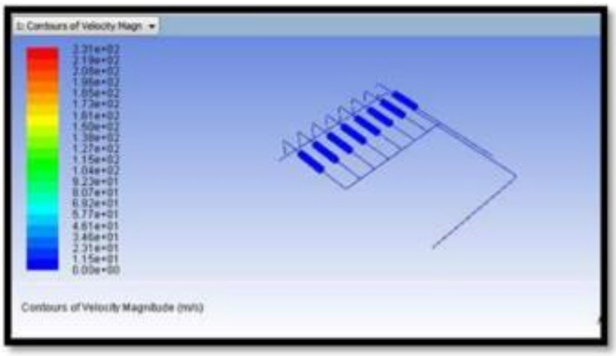
CFD ANALYSIS OF FLUID R-124



Pressure



Velocity



Heat transfer coefficient

V. THERMAL ANALYSIS OF HLHPS

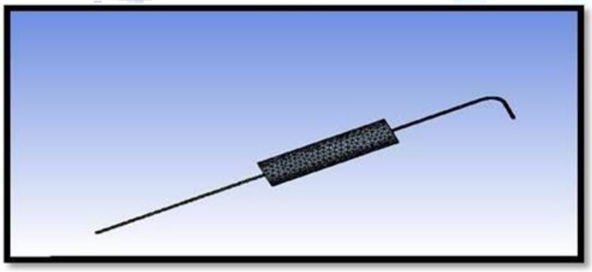
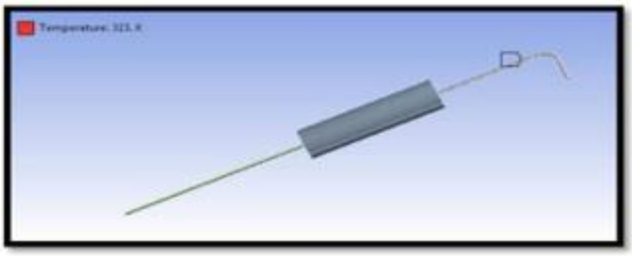
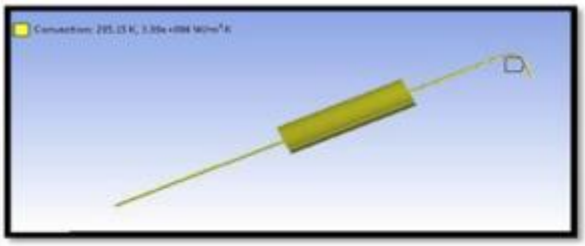


Fig. 115 : Meshed model of single evaporator

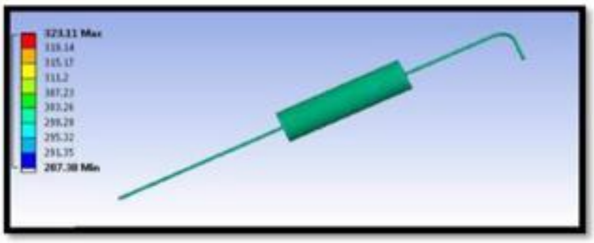
Applied Temperature



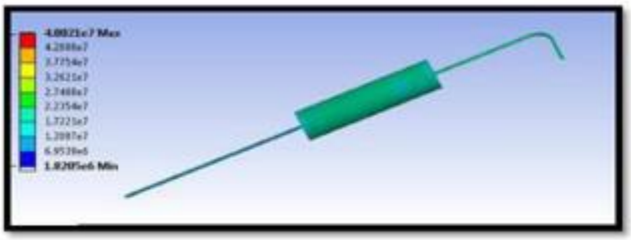
CONVECTION



Temperature



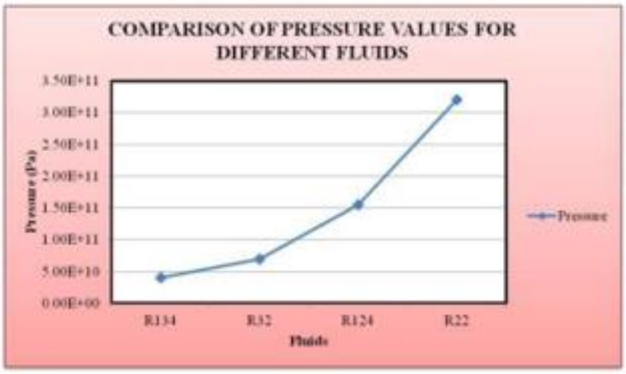
Heat flux

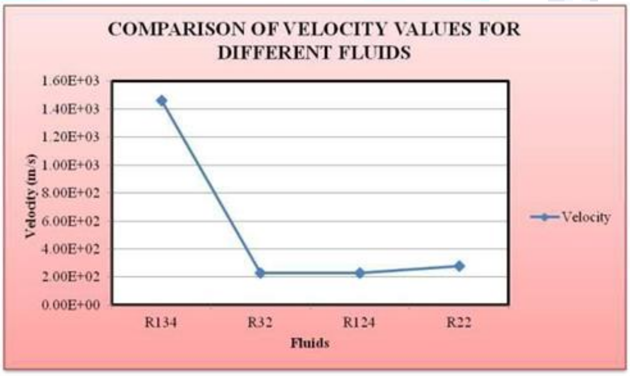


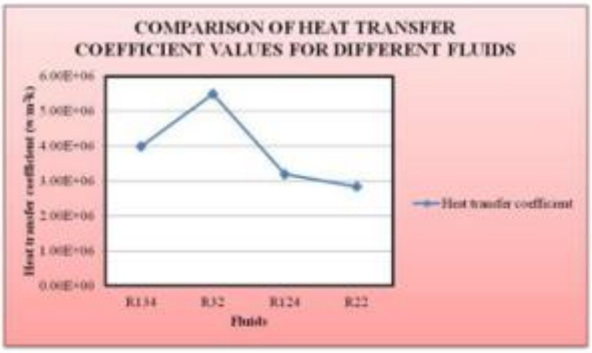
RESULT TABLES AND GRAPHS

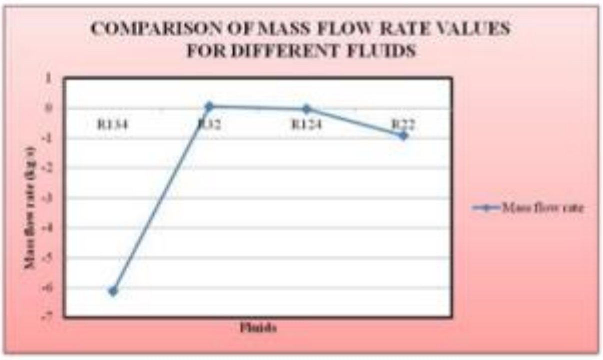
CFD ANALYSIS

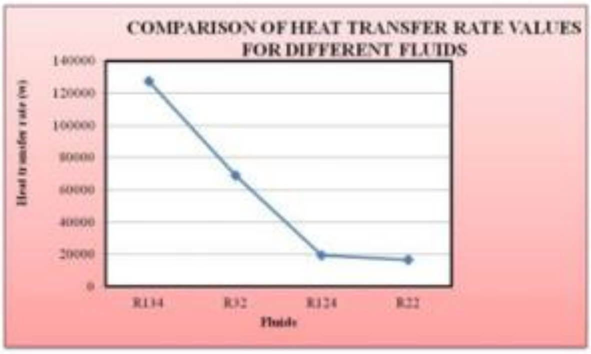
Fluids	Pressure	Velocity	Heat transfe	Mass flow	Heat transfer
R1	4.00E	1.46E	3.99E	-	127236
R3	7.00E	2.27E	5.49E	0.04578	69215.
R1	1.55E	2.31E	3.19E	-	19602.
R2	3.20E	2.78E	2.84E	-	16739.



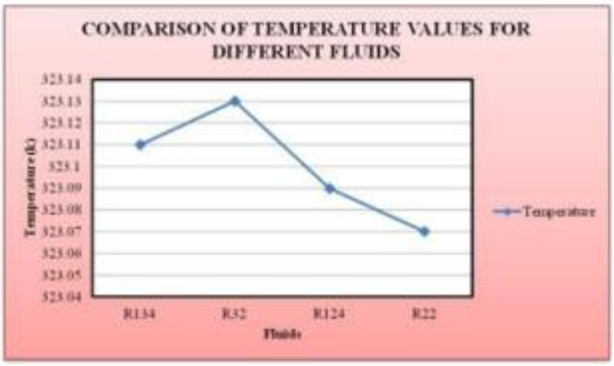


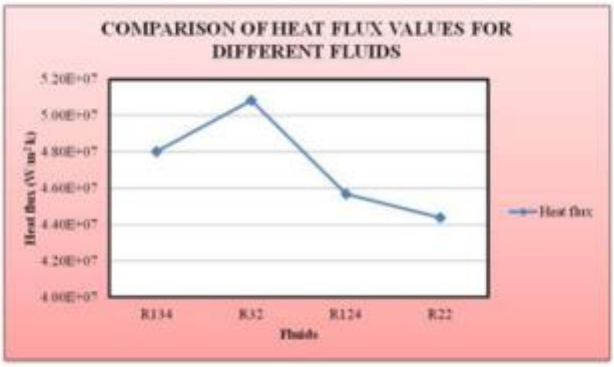






THERMAL ANALYSIS





VII CONCLUSION

In this thesis, Hybrid Loop Heat Pipe System is and modeled in 3D modeling software Pro/Engineer.

Heat transfer characteristics of HLHPS are determined by CFD and thermal analysis. CFD analysis is done to determine the heat transfer rate, pressure drop, mass flow rate, heat transfer coefficient different working fluids R134, R22, R34 and R124 . Thermal analysis is done to determine heat transfer rate and temperature distribution.

By observing CFD analysis results, the pressure in the evaporator is more when fluid R22 is used. The value is more by 87.5% when compared with that of R134, by 78% when compared with that of R32 and by 31% when compared with that of R124. The heat transfer coefficient is more when fluid R32 is used. The value is more by 27% when compared with that of R134, by 41.8% when compared with that of R124and by 48.26% when compared with that of R22. The heat transfer rate is more when R134 is used. The value is more by 45.6% when compared with that of R32, by 84.5% when compared with that of R124 and by 86.84% when compared with that of R22.

By observing thermal analysis results, the heat flux is more (i.e) heat transfer rate is more when R32 is used since the heat transfer coefficient is more.

Fluids	**Temperature (k)**	**Heat flux (w/m2k)**
R134	323.11	4.8021e7
R32	323.13	5.0847e7
R124	323.09	4.57e7
R22	323.07	4.4387e7

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