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CFD Analysis and Effect of Filling Ratio on Thermal Performance of Pulsating Heat Pipe using Distilled Water

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Abstract:

The process of heat allocation in liquid and vapour phase change is influenced by the pulsating heat pipe. This paper presents an investigational work done on pulsating heat pipes using distilled water as working fluid. Pulsating heat pipe (PHP) made from a copper pipe having an internal diameter 3 mm and eight turns. The PHP is filled with distilled water fluid of varying filling ratios of 45%, 55%, 65%, 75%, and 85% of its volume. Differentiating heat inputs of 120W, 240W, 360W, 480W, and 600W are applied. The evaporator section is for charging dissimilar filling ratios with varying heat input. The results obtained showed an increase in the heat input with the thermal heater by using a flat plate mica heater and a condenser section cooled by a heat exchanger. Closed loop PHP was considered has more effective impact on heat transfer of filling ratio. The investigation required accepted resistance to decrease rapidly, and a lower rate of thermal resistance is perceived at a filling ratio of 55%. It was found that for distilled water measured for calculating heat transfer performance at 600W, evaporator temperature at 261.57°C, and condenser temperature at 57.062 °C, thermal resistance of 0.13°C, /W, heat transfer coefficient at 475.32W/m^2°C, and hence exhibits good enactment at a filling percentage of 55% and analysis of control volume fraction of pulsating heat pipe.

Keywords: Pulsating Heat Pipe, Heat Input, Filling Ratio, Distilled Water.

1. Introduction to the Closed Loop Pulsating Heat Pipe.

A Pulsating heat pipe is a thermo siphon-like multiphase heat transfer mechanism. Fig.1 shows It is filled with a working fluid that flows from the condenser section to the evaporator segment as observed B Y Tong et al., [1] by transforming from liquid to vapour. This type of working fluid chosen affects PHP's performance. Vipul M. Patel et al., [2] observed that the working fluid has high heat conductivity and has an impact on the starting mechanisms in PHP. Rudresha S et al., [3]: performed CFD analysis to contrast the performance of pulsating heat pipes using water and ethanol as working fluids. The results of this investigation demonstrate that when heat inputs are altered at a fixed filling ratio of 60%, water exhibited superior thermal performance over ethylene alcohol. Additionally, water exhibited greater thermal performance since its surface tension is higher than that of ethylene alcohol. Manzoni, M et al., [4] made a positive prediction for low, intermediate, and high temperatures in the later, which influenced the cooling method's outstanding heat transmission. Perpendicular heating mode experiments were carried out with a 50% filling ratio at heat input ranging from 10-110W. Resistance to distilled water was shown to be lower at increasing heat input irrespective of any fluids. In order to create nanofluids. Rudresha S et al., [5] five concentrations of titanium oxide nanoparticles in PHP: 0.18%, 0.36%, 0.54%, 0.72%, and 0.90%. The heat input was varied in steps of 100 W, ranging from 400 W to 800 W. Using a bath sonicator, the TiO_2 nanoparticles were combined with Dowtherm-A to create a stable solution. All the experiments were performed in vertical mode with varied mass concentrations and heat inputs. The evaporator and condenser temperatures, thermal resistance, heat transfer efficiency, and improvement of heat transfer performance were taken into account while evaluating heat transfer performance. We ix iu at al.,[6] used ethanol and Acetone working fluid with varying filling ratios of 50%, 75%, and 85% at heat input of 37 W, 68 W, 108 W, 141 W, and 175 W. A closed-loop pulsating heat pipe (PHP), also known as an oscillating heat pipe, (OHP) is typically made up of a sealed, meandering channel that is first evacuated and then partially filled with a working fluid. The vapour quality of such a two-phase system is frequently low, and a succession of internal vapour bubbles and liquid slugs is common. When the channel's diameter is small enough to be capillary, liquid slugs form. i.e., the liquid meniscus entirely fill the section and there is no fluid stratification. There may be one or more sections.

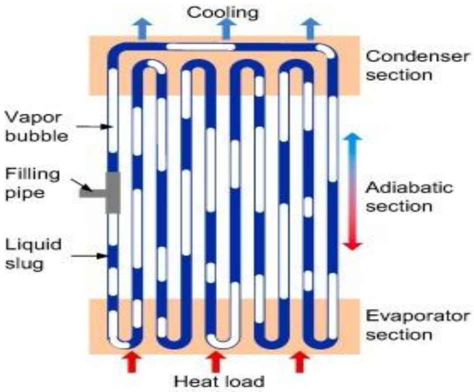

Fig. 1: Block Diagram of Pulsating Heat Pipe

Acetone illustrates a lower rate of thermal resistance as compared to ethanol. Anand Takawale et al., [7] used Capillary tube Pulsating heat pipes and flat plates pulsating heat pipe and carried out exploration to compare the heat transfer performance of heat input steps sizes of 20W to 180W through the filling percentage of 80%, 40%, and 60%. Ethanol was used as a working fluid. It was observed that the amplitude of oscillation was higher in CTPHP than in FPPHP. The thermal resistance reduced by 83% and 35% in FPPHP and CTPHP, respectively, in the presence of working fluid. Rudresha S et al., [9] in their research showed improved thermal performance of a heat pipe charged with nanofluids. In the heat pipe, the working nanofluids used were SiO_2/DI water and Al_2O_3/DI water, with concentrations of 10, 20, and 30g/lit, respectively. Analysis of CFD findings were compared with experimental finding like the thermal resistance, thermal heat transfer coefficient, thermal conductivity, and efficiency for the CLPHP SiO_2/DI Water and Al_2O_3/DI Water heat pipes respectively 69.37 %, 75.99 %, and 11.98% DI water nanofluids at heating powers of 10w, 14w, 18w, and 22w. Rudresha S et al.,[10] in their investigations indicated that a higher filling ratio and a vertical orientation improved heat transmission efficiency. The most efficient design factors took into account the heat input, the number of turns, the working fluids, the pre-installation conceptual model, the type of heat pipe being used, the appropriate materials selection, the filling ratio, the role of gravity, the orientation of the PHP, aspects ratio, and the diameter of these start-up characteristics. Vipul M. Patel et al., [11] showed that the heat input was controlled within a 5–200 W range. It has been found that FR of 50% and nine-turn CLPHPs performed better in all orientations. It has been found that a CLPHP's operating efficiency improved as heat input, FR, and the number of rotations increased. Thi ago Dutra et al., [12] in their experiment constructed and tested LHP using acetone as the working fluid to handle heat transfer rates of up to 70 W. The testing findings demonstrated that, despite the design constraints being imposed, the suggested LHP performed well in terms of heat management. Future space missions should consider using the suggested LHP as a dependable thermal control system, especially if a less dangerous working fluid is used. Kangli Bao et al., [13] used a charge ratio of 56.7% and a heat flow of 3503 W/m^2, and showed that the thermal resistance of the PHP with a 20 ppm surfactant solution fell by 27.8%. The findings also revealed that the surfactant-solution PHP had larger heat fluxes during dry-out and displayed rather steady temperature oscillation characteristics when compared to the DI water-PHP at a charge ratio of 67.9%. Bao, K., et al [14] showed that the dry-out heat flux of the PHP with the surfactant solutions rose by 17.80%. The experimental results indicated that PHP with surfactant solutions may begin with a lower heat flux than DI water-PHP. The PHP had the lowest thermal resistance under the testing circumstances when the solution concentration was 20 ppm. The PHP with 20 ppm surfactant solution had a lower thermal resistance than the DI water-PHP at a charge ratio of 56.7% and a heat flux of 3503 W/m2, dropping by 27.8%. Babu E.R et al., [15] carried out research using acetone with a filling percentage of 60% for varying mass concentrations of 1%, 2%, 3%, 4%, and 5% of Al_2O_3 nanoparticles with heat inputs varying from 20 W to 60 W. The greatest decreased value of thermal resistance, according to the results of varying percent mass concentration, was 0.28 K/W at 1% mass concentration. As a result, PHP performed better with an Al_2O_3 nanoparticle with mass concentration of 1%. V Srinivasan et al., [16]

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observed that due to the creation of separate vortex pairs behind the meniscus as a result of the free-slip boundary condition, the flow inside the liquid plug exhibited distinctive non-Poiseuille flow characteristics close to the meniscus. With each oscillation stroke, the vortices also shifted their orientation, eventually resulting in an alternate recirculation pattern inside the plug. Arabnejad S et al.,[17] conducted investigations to study how the evaporator temperature affects the pulse amplitude and frequency, convection rate, and boiling heat transfer. The findings demonstrated that by raising the evaporator temperature, the rate of heat transfer via convection and boiling in the pipe also rose as a result of an increase in pulse amplitude and frequency. Additionally, it was deduced that an increase in evaporator temperature will result in a greater percentage of boiling heat transfer. M. Ebrahimi et al., [18] showed that filling ratio of 65% was found to be the point at which IC-FP-CLPHP performed at its most productive level. According to flow visualization interconnecting channels have an impact on the flow regime and improve flow circulation and heat transmission in CLPHPs, The importance of interconnecting channels in creating one-way flow has been demonstrated using a numerical approach on a single-phase liquid to further explore the idea's practicality. Rudresha S et al., [19] showed that the obtained CFD results are contrasted. The CFD study was completed, and the simulation outcomes are shown in graphs and outlines. Changes in the volume fractions of water-air and methyl alcohol-air in the evaporator, adiabatic zone, and condenser represent the flow pattern of the fluid inside the PHP. The analysis examines the thermal effects of fluid on the curvature of a single turn closed-loop pulsing heat pipe at various periods using a common Kepsilon model for the enhanced wall treatment. R.K.Sarangi, et al., [20] recorded the maximal heat loads and temperature variations of the adiabatic section at startup. According to experimental findings, maximum heat load was based on fill ratio, whereas the initial heat load was independent of fill ratio. The working fluid for a particular PHP and operating temperatures determined the best fill ratio for the highest heat load.

2. Experimental Set-up and Procedure.

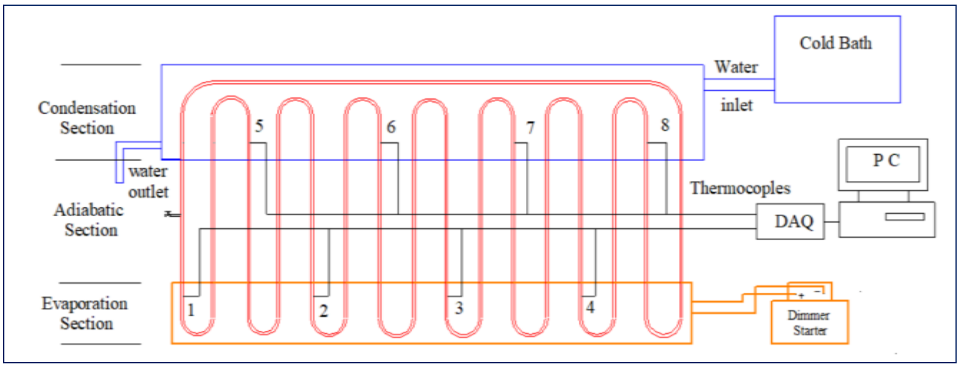

Fig. 2: Block Diagram of Experimental set up.

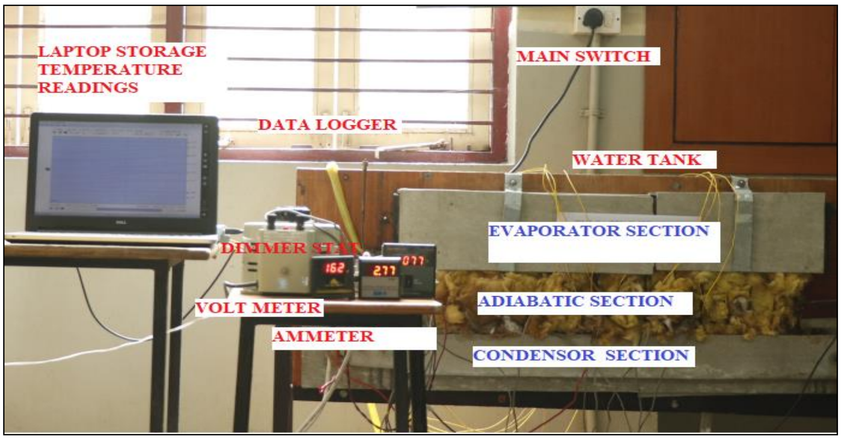

Fig. 3: Experimental setup of PHP

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Figures 2 and 3 show a schematic block diagram and an experimental arrangement for PHP loops with eight loops. A Copper pipe with an interior diameter of 3 mm and an outside thickness of 4 mm with a thickness of 1 mm was used in this experiment. Copper was chosen because of its excellent thermal conductivity. The PHP section is divided into three zones: the major source of heat input is in the evaporator section; the adiabatic section where no heat is absorbed or rejected due to the use of glass wool as an insulator, and the condenser section where heat is discarded. The evaporator part includes a 60x120mm flat plate heater with a maximum capacity of 2000W. Heater was powered by a variable voltage supply ranging from 0 to 230V. The temperature fluctuations inside the pulsating heat pipe were recorded using a laptop and a DAQ system. T4, T3, T2, T1 (Evaporator Section), T8, T7, T6, and T5 (Condensers Section) are the thermocouples used in this system. As shown in figure 3, the analogue data obtained during testing was converted into digital form using the DAQ equipment that comes with the system. The complete system was cooled by continuously supplying continuous water with a flow rate of 0.0032 kg/sec. as a working fluid filled inside the pipe with percentages ranging from 45%, 55%, 65%, 75%, and 85% accordingly. Heat generation in the evaporator portion was used at 120W, 240W, 360W, 480W, and 600W by a dimmer stat modulating on a flat plate heater in steps of 120W. Experiments were carried out for specific parameters over time, and it was discovered that as the temperature in the evaporator section rose, pulsations inside the PHP appeared, and once the steady state was attained, –there was continuous pulsations from the evaporator to the condenser section appeared.

2.1 Working Fluid Selection.

Selection of the base fluid is a significant factor that controls PHP heat transfer performance. The working fluid was selected based on fluid characteristics and properties such as viscosity, specific heat, latent heat, and thermal conductivity. It is denoted by M.

$$M= \frac{\rho\sigma\measuredangle}{\mu} \text{ ----------------- Eq (1)}$$

2.2 Uncertainties and Data Accuracy.

The thermal resistance of PHP and heat transfer coefficient were calculated by using the formula as shown in Eq 2 and Eq 5.

$$Rth = \frac{Te-Tc}{Q} \text{ ----------- Eq (2)}$$

Evaporator temperature and Condenser temperatures of PHP were calculated by using the formulas shown in Eq.3 and Eq.4.

$$T_e = (T_4+T_3+T_2+T_1)/4 \text{ -----------Eq (3)}$$

$$T_c = (T_8+T_7+T_6+T_5)/4 \text{ -----------Eq (4)}$$

The heat input Q and Heat Transfer Co-efficient of PHP were alculated by using the formulas as shown in Eq 5 and 6.

$$Q = V*I \text{ ------------ Eq (5)}$$

$$h= \frac{Q}{As(Ts-TC)} \text{ ---------- Eq (6)}$$

3. Geometry of the Pulsating Heat Pipe.

Fig 4: Geometry of Pulsating Heat Pipe

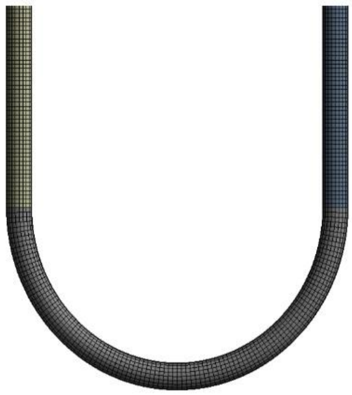

Fig 5: Mesh Boundary

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As shown in figure 4, CFDs meshing allows us to define the pairs of point controls, edge controls, surface controls, and body controls, giving us more control over the automatic settings. They each have their own set of options and can be used to change the mesh in a variety of ways and in this instance an automated mesh method was chosen. On the other hand, the mesh is manually put together. PHP is a closed loop with a 2 mm diameter Copper Tube and a tube with a single turn. Only the fluid domain is taken into account in the geometry modelling. Copper tube with a total channel length of 361.68 mm and copper pipe are used in the PHP design. Because the copper domain is simply the subject of the liquid domain of interest, it is not monitored for geometry simulation. With lengths of 90.84 mm, 180 mm, and 90.84 mm, respectively, the PHP is divided into three sections: evaporator, adiabatic, and condenser.

3.1 Meshing of Pulsating Heat Pipe.

Figure 5 shows that CFDs meshing allows us to define the pairs of point controls, edge controls, surface controls, and body controls giving us more control over the automatic settings. They each have their own set of options and can be used to change the mesh in a variety of ways in this instance. An Automated mesh method was chosen. On the other hand, the mesh is manually sized. The minimum and maximum mesh sizes were chosen to be 0.0002 mm, as shown in figure. 5. When this control option was employed, 42959 nodes and 37246 elements were used to mesh. The portions of the geometry are given names during the meshing step of CD Fluent, making it easy to define the domain and establish boundary constraints in the setup stage. Three domains were defined for PHP's overall geometry, namely, condenser region, the adiabatic region, and the evaporator region. Further the adiabatic zone is divided into two domains to account for the evaporator at the bottom which causes an increase in distilled-water levels in the PHP. For a water filling ratio of 65% by volume. Variable heat inputs of 120W, 240W, 360W and 480W, 600W were delivered to the heater section or Evaporator zone. The adiabatically cool area has no heat flux. In the condenser region, a negative heat flux was recorded; this heat flux was calculated using an experimental reference article. VOF model was employed in the CFD study, coupled with the enhanced wall treatment and k-epsilon. All the domains' walls were made of non-slip material.

3.2 Turbulence Model of Pulsating Heat Pipe

The k-epsilon model, one of the most well-known turbulence models, was utilised in this study, as it is in most general-purpose CFD systems, and is widely regarded as the industry standard. It has ~~a~~ well-known predictive power and has been shown to be stable and numerically robust. For general-purpose simulations, the k-epsilon strikes a good balance between accuracy and resilience. Traditional two-equation models are used in the k-epsilon turbulence model in CFX. These two equations are just transport equations (or partial differential equations) for turbulent kinetic energy (k) and dissipation, and they use the scaled wall-function technique to increase resiliency and accuracy when the mesh of the near wall is one of the finest. As a result, it provides precise fluid flow projections for the domain. The condenser zone's velocity components (u, v, and w) were set to 0 m/s, and the relative pressure was set to 40 Pa. There would be less turbulent flow in the condenser because the beginning condition was just low pressure air at **a** low temperature. As a result, the volume fraction value for air was 1, while DI water vapour has both zero volume fraction values. This was due to the lack of ddistilled water vapour in the condenser in the first place, and the establishment of a thermal boundary condition with a fixed temperature of 298 K when the evaporator was heated. Similarly, the conditions for water and water vapour in the presence of air were maintained, with the volume fraction for water and steam set to 0 and for air set to 1. The temperature of the condenser portion was maintained at 298 degrees Fahrenheit. All of the velocity components in the evaporator were set to 0 m/s, and the relative pressure was set at 40 Pa. Water vapour, and air were assigned volume fractions of 0, 0 and 1, correspondingly. All of the domains' walls were made of non-slip material. It's assumed that all of the walls are smooth. In adiabatic zones, the boundary condition is also adiabatic, defining the heat flux rate as 0 Wm-^{2}K-^{1}. Similarly, the conditions for water and water vapour in the presence of air were maintained, with the volume fraction for ~~d~~istilled water and steam set to 0 and for air set to 1. The temperature in the condenser area was maintained at 373K.

4. **Analysis using Contours of Volume Fraction.**

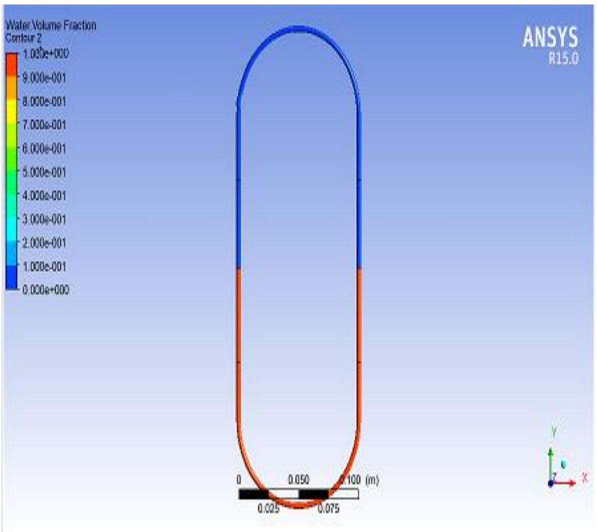

Fig. 6: Water Volume fraction contours at t=10mts

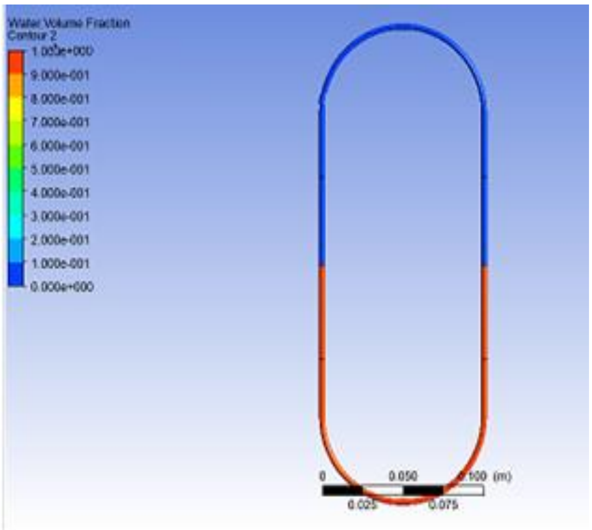

Fig. 7: Water Volume fraction contours at t=20min

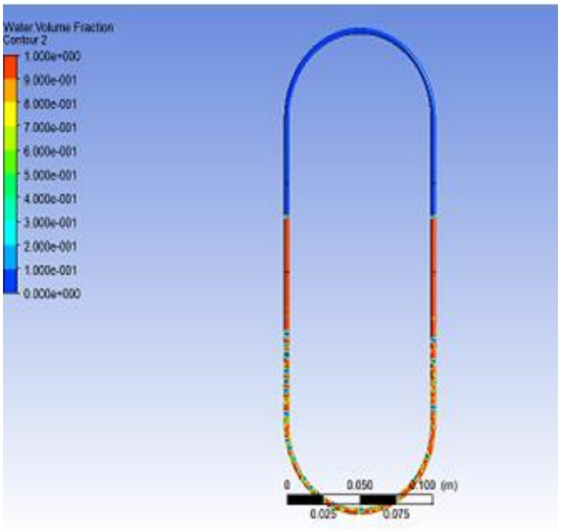

Fig. 8: Water Volume fraction contours at t=30min

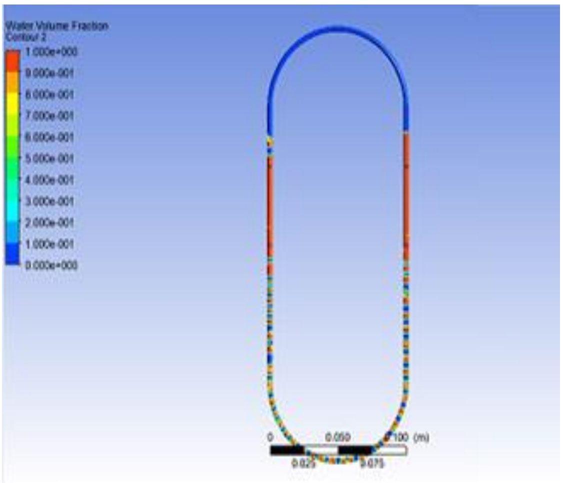

Fig. 9: Water Volume fraction contours at t=40min

The simulation results of the change in distilled water volume of a fraction of the pulsating heat pipe at different time intervals are shown from figure 6 to 8. Figure 6 shows the generation of vapour at the beginning and the figure 7 shows the growth of generation of vapour at the beginning and the figure 8 shows the growth of generation of vapour bubbles and its movement towards condenser section. When heat input increases more and more vapour bubble formations take place and are moved with a high velocity of about 0.678m/s as shown in figure 9. The vapour phase is condensed in the condenser section, and the condensed liquid slug comes back to the evaporator, and then the heat transfer cycle repeats. From the water volume fraction vectors shown in figures 8 to 9, the working fluid circulates in the adjacent channels as it reaches stable oscillation. Also, it was observed that the vapour slug and liquid slug ~~is~~ are moving to and frost different time intervals, which carries the heat flux from evaporators to condenser by an oscillating behaviour. During this process, there was a change in phase from liquid to vapour and vice versa. From the submission results, it is also proved that heat transfer in pulsating heat pipe pipes depends on the fluid oscillations.

After conducting iterations using CFD Fluent software for a count of roughly 47,000+ iterations for both water and air, the output of the simulation of the single turn pulsing heat pipe is observed to be utilising the contours of the volume fraction of water and air. The volume percentage of water

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contours was obtained by maintaining all relevant operating circumstances and design parameters, such as pressure at 40 Pascal, number of mass transfer mechanisms at 1, and number of Euler an phases at 3. The time step size was kept at 0.001, and the maximum number of time steps done at one time was 500, with each time step having 20 iterations. The tube's radius was retained at 2 mm, while the heater and condenser sections' temperatures were kept at 373K and 298K, respectively, and the modelling was done in the ZX plane with gravity equal to g=9.8m/sec2 in the y direction. The number of tubes for PHP was fixed at one, and the tube material was copper. The number of time steps shown in Fig. (4) Represents the number of times the flow equations are solved at each time step. The liquid slug was created near the evaporator portion, as evidenced by the contours of distilled water in fig 5, 6, 7, and the liquid and vapour slug does-not reach the condenser section after 47,000 iterations. Because there is less circulating flow of distilled water inside the PHP after extracting heat from the heater section and releasing it to the condenser section, the thermal performance of distilled water is not up to par. The number of time steps shown in Fig (8) represents the number of times the flow equations are solved at each time step. The liquid slug was created near the evaporator section, as demonstrated by the contours of volume percentage of water in fig 9, 10, 11, and the liquid and vapour slugs reached the condenser portion after 45000 iterations. This demonstrates that water has better thermal performance than ethylene alcohol because the circulating flow of water inside the PHP after extracting heat from the Heater Section and releasing it to the Condenser Sections is much better in water when operated at the same temperature.

5. Results and Discussions.

5.1: Evaporator Temperature (Te) with heat Input and Condenser Temperature (Tc) with heat input.

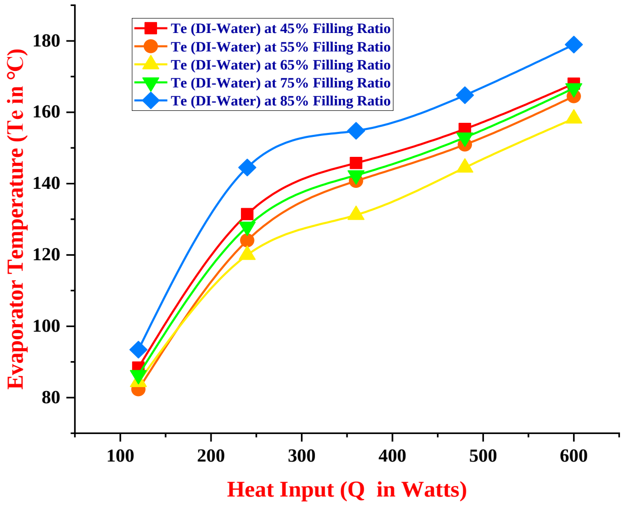

Fig. 10: Evaporator Temperature v/s Heat Input

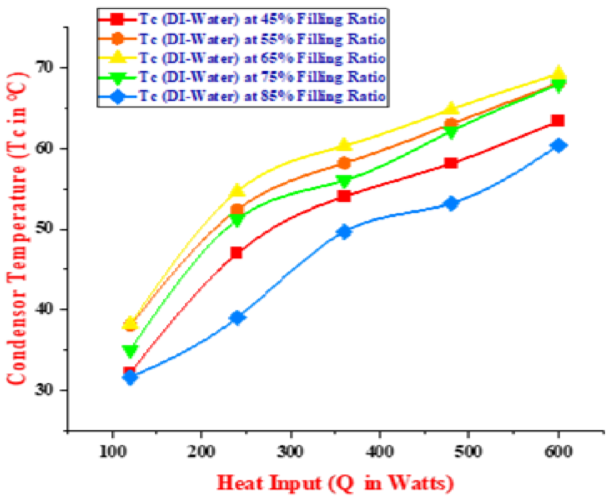

Fig. 11: Condenser Temperature v/s Heat Input

Figure 10 and 11 displays the variation of evaporator and condenser temperatures of PHP, where the heat is transferred from the evaporate section to the condenser section. It is seen that the evaporator temperature is very low at lower heat input and increases with an increase in heat input for all the filling ratios considered. As the evaporator temperature increases, the vapour bubbles increase and the liquid plugs in the pulsating heat pipe accelerate, which enhances the heat transfer. As in the condenser section, there is more heat isinput, more fluid fluctuations occur inside the PHP and hence heat transfer rate is higher. The movement of the fluid is slow initially due to the inertia of the system. As the condenser temperature increases, the liquid plugs and vapour bubbles in the pulsating heat pipe accelerate, which enhances the convective heat transfer and causes more heat transfer through the sensible heat in distilled water.

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5.2: **Thermodynamic resistance of distilled water for different filling ratios and heat input and heat transfer Co-efficient of distilled water for different filling ratio ratios and heat input**.

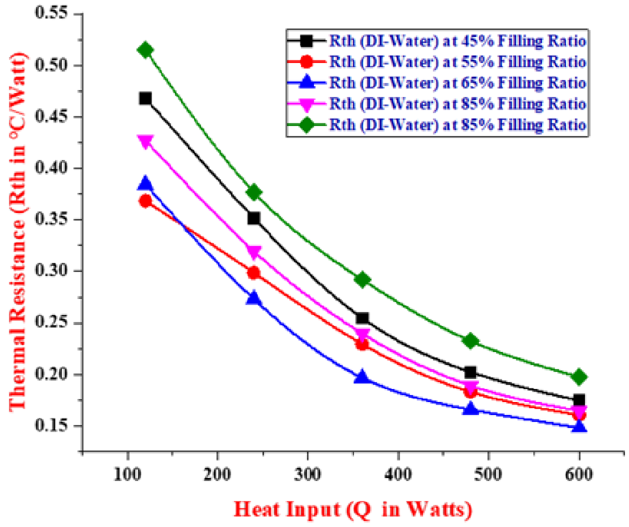

Fig. 12: Thermal Resistance versus Heat Input

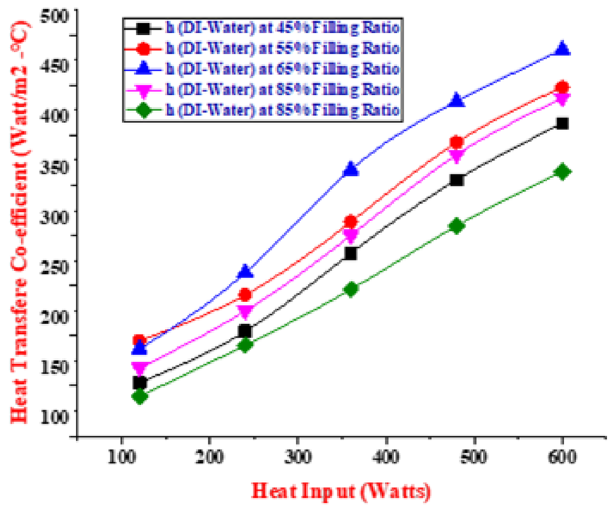

Fig. 13: Heat Transfer Co-efficient versus Heat input

Figure 12 shows variations of thermal resistance versus heat input intended for different filling ratios of 45%, 55%, 65%, 75%, and 85%. From figure7, it is evident that thermal resistance declines very rapidly during initial heat input due to low pulsations and gradually thermal resistance values reduce with an increase in heat input because of high pulsating movement. Distilled water starts the pulsations in the intermediate temperature range 150^0C to 200^0C. It is found from the results that at a 55% substantial ratio, a lower assessment of thermal resistance of 0.153^0C/W is found at 600 watts. Therefore, the PHP functions well at at substantial ratio of 55% compared to other substantial ratios.

Figure 13 demonstrates the influence of heat input and heat transfer co-efficient on distilled water. The heat at the evaporator section moderately increases with respect to time, and the fluid gets converted from a liquid to a vapor state by reducing its density. At the evaporator section, bubbles are created and collapse into each other at the condenser section and pressure increases which in turn starts the pulsating movement. This causes distilled water to start impulse movement from the evaporator to the condenser section. At a substantial ratio of 55%, the maximum value of heat transfer coefficient of 475.32W/m^2 ^{0}C is obtained as compared with other substantial ratios.

6. Conclusion.

Provisional research on heat transfer enactment of PHP through DI water as a working fluid charged with different substantial ratios and different heat inputs was carried out and the following conclusions are drawn

- In general, with an increase in heat input; there is a decrement ~~of~~ in thermal resistance.
- Thermal performance with distilled water is enhanced in heat transfer at 55% substantial ratios.
- Lowest rate of thermal resistance of ^{0}C/W with the best combination of heat input of 600W and optimum substantial ratios of 55% is obtained.
- Enhancement of heat input leads to an enhanced heat transfer rate of PHP. Maximum value of heat transfer rate of 475.32W/m^2 ^{0}C was obtained at 55% substantial ratios at 600W.
- CFD analysis of DI water volume fraction in variations of liquid to vapour.

7. Nomenclature

h = Heat transfer co-efficient w/m^2- ℃
As = surface area heat transfer in m^2
L =length of PHP in m
A_{CS} = area of cross area of pipe in m
PHP =Pulsating heat pipe
VOF = Volume of Fluid
T4, T3, T2, T1=Evaporator Temperatures in ℃
T8, T7, T6, T5=Condenser Temperature in ℃
Dcr = diameter of pipe in mm
g = acceleration due to gravity in m^2/sec
DAQ = Data Acquisition System
FR = substantial ratio
M = merit number

ρ= density in Kg/m^3
σ = surface Tension N-m
λ = Latent heat J/kg
μ =Liquid Viscosity Ns/m^2

8. Acknowledgments

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