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Effect of Filling Ratio on Thermal Characteristics and Performance of a Pulsating Heat Pipe

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***Abstract:** Pulsating heat pipes are complex devices for heat transfer and their optimal thermal performance depends mainly on different parameters. This work is about the thermal efficiency of a closed-loop oscillating heat pipe with a diameter of 2.0 mm and 3.0 copper tube inner and outer. For all experiments, the filling ratio (FR) was used 40%, 50 %,70%,80% and heat inputs of 20W, 40W, 60W, and 80W was provided to PHP. The position of the PHP was vertical bottom heat type. The length of evaporator, adiabatic and condenser section was maintained 52 mm,170mm,60mm. Water and benzene were selected as working fluids. From the available literature it is observed that working fluid and filling ratio are key factors in PHP's performance. The results show that the thermal resistance decreases rapidly with the increase in the heat input to 20 to 40 W., while it decreases gradually over 40 to 80W.Simulation is done in CFD and experimental data were equated to the results.*

***Keywords:** closed loop pulsating heat pipe (CLOHP); Thermal performance; Thermal resistance; Benzene*

I. INTRODUCTION

In 1990, Akachi [1] introduced and patented a modern wickless heat pipe component named pulsating heat pipes (PHPs). Its action is based on the oscillation theory of the working fluid and a phase shift effect in a capillary tube. The diameter of the tube must be sufficiently small to provide liquid and vapor plugs. Simple design, low cost, good thermal efficiency and fast response to high heat load, PHP is recognized as one of the innovative developments in a closed cabinet of digital cooling equipment, the spacecraft thermal control system, heat exchanger, designed for use in the thermal control of the nuclear reactor ,remove heat from leading edge of hypersonic aircraft,Electronic cooling has small high-performance components cause high heat fluxes and high heat dissipation demands, used to cool transistors and high-density semiconductors' Several mathematical models had also been developed in recent years to predict the oscillating behavior and heat transfer capacity of PHPs.

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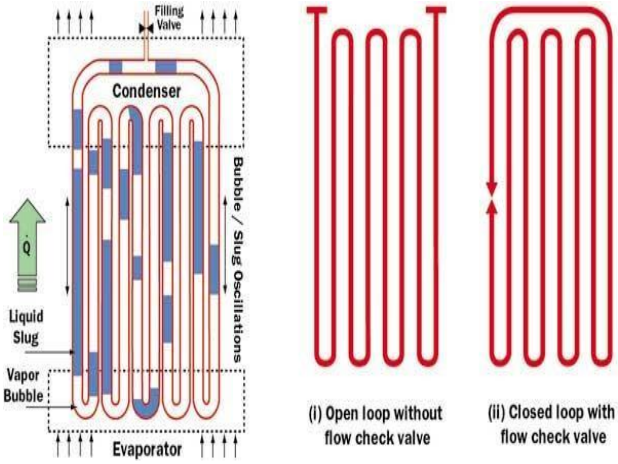

Fig 1: Schematic diagram of pulsating heat pipe

The most conspicuous condition of PHP in the form of liquid slug tube design Akachi et al. [2] specified that surface tension and channel buoyancy effect the presence and movement of bubbles. The dimensionless equation in Eq (1) may explain connection between surface tension and buoyancy.

$$E_{\ddot{o}} = \frac{g^{D^2}(\rho_{liq} - \rho_{vap})}{\sigma} \qquad (1)$$

If Eö≈ 4 seizes the bubble on two sides of wall. The terminal velocity is zero and in this case the fluid slug is developed. The dimensional equation is given in Eq ' (2) for the essential diameter ' D.

$$D < D_{cri} \leq 2\sqrt{\frac{\sigma}{(\rho_{liq} - \rho_{vap})g}}. \qquad (2)$$

Through Eq. (2), All pure and binary working PHP liquids are measured in critical diameters and located within 3-5 mm distance. But tube diameter is lower than critical diameter. According to the literature review and performance, most PHPs use 2.0 mm inner and equivalent 3.6 mm outer pipe diameter.

II. EXPERIMENTAL SET-UP

CLPHP's laboratory set-up made up of evaporator, condenser, and opaque adiabatic, as shown in Fig. 2.1. In the laboratory, CLPHP is built and tested, consisting of five capillary turns, partly made of copper (ID: 2.0 mm, OD: 3.6 mm) and partly glass (ID: 2.0 mm, OD: 4.0 mm).A mixture of M-seal, araldite and welded is wrapped to the gap. To visualize liquid slugs and liquid plugs, typical sealing technique on copper and glass tube is arranged. As 42 mm, 50 mm and 170 mm respectively, the size of evaporator, condenser and adiabatic are chosen.

The attempt to test and contrast the affects of 0° (vertical), 30°, 45° and 90° in PHP. Tube distance is 20 mm and CLPHP's total size is 60 mm x 262 mm. The evaporator is well protected by simultaneous use of cloth, glass wool and asbestos plate. To attain vacuum in pipe, the filling valve has been linked to a reciprocating vacuum-pump, producing a vacuum rate at least 70cm Hg.

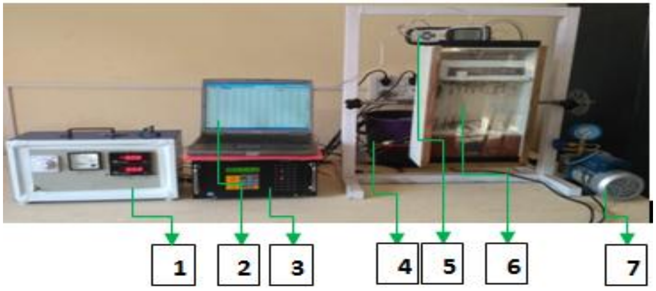

Fig. 2.1 Experimental Setup of PHP
1. Heat variac 2. Data Recording laptop 3. Data logger 32 channel 4. water motor 5. Manometer 6. PHP 7. Vacuum pump

Ten thermocouples (T1~T10) are attached symmetrically arranged in condenser and ten thermocouples (T13~T22) in evaporator. Alternatively, in condenser at inlet thermocouple (T11) and outlet (T12) to monitor temperature difference. Four thermocouples (T23~T26) are evenly attached to walls of evaporator; and four thermocouples (T27~T30) are evenly attached to walls of condenser. A flow meter is employed in condenser to record the mass flow-rate of cooling water. The digital logger in setup has 32 channels, the type of inputs was RTD Pt100 (3 wire), DCVolt, DC milliampere, and Thermocouples J Type of input is settable for channel from keypad.

III. RESULTS AND DISCUSSION

Thermal resistance is inversely proportional to CLPHP's thermal performance attained by the procedure below. Until a quasi-thermal equilibrium was created, heat output was gradually increased. To obtain thermal resistance, temperatures and heat output are reported [2], which is defined by Eq. (3), as

$$R = \frac{Te-Tc}{Q} \quad {}^{0}\text{c/w} \;\ldots\ldots\; (3)$$

Where:
R = Thermal Resistance ^{0}c/w
Q = heat flux in (w)
T_e= Evaporator Temperature in (k)
T_c= Condenser Temperature in (k)

3.1 Effect of different working fluids on thermal performance

3.1.1 Water as working fluid

The affect of thermal resistance is plotted in Fig.2 for different heat inputs.

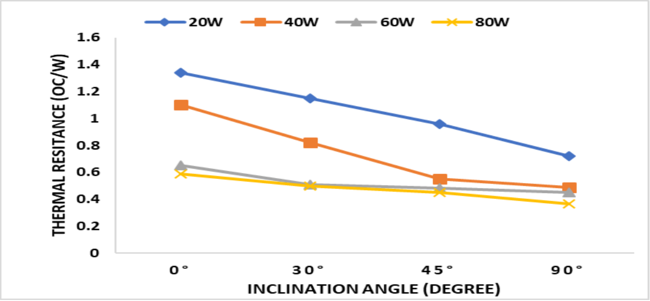

Fig. 2(a) Variation of thermal resistance in DI water with different Inclination angles at 50% Fill ratio

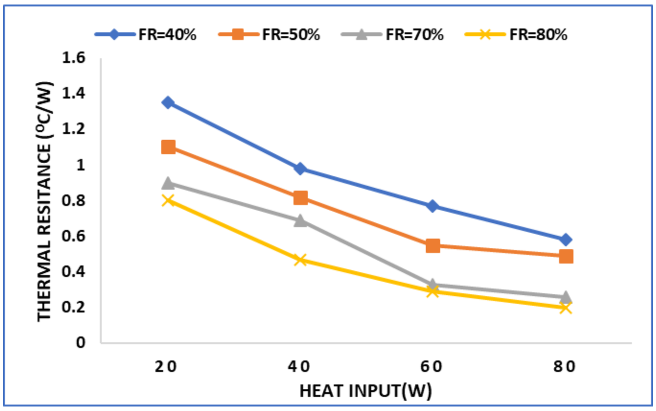

Fig. 2(b) Variation of thermal resistance in DI water with different Fill ratio, vertical position of PHP

Water thermal resistance at different angles at the heat inputs with a 50 % filling ratio. As the heat input rises, at an inclination of 45 degrees the thermal resistance decreases, in Fig. 2(a) i.e. for heat input between 40w and 60w.The flow was begun organized, and the fluid trends were described as churning with wispy annular air. The flow increased vigorously over 56 W of heat input. Thus, thermal resistance reduces again, and stream patterns shift from wispy annular to annular air. Due to thermal resistance the graph pattern trends to decrease with each fill ratio increases heat input. With increasing heat input, more fluid is vaporized so that heat can be expelled by phase change. This can be noted that HI=60w is highly consistent with filling ratio FR=50 %is shown in Fig. 2(b), showing relatively good repeatability of the test.

3.1.2. Benzene

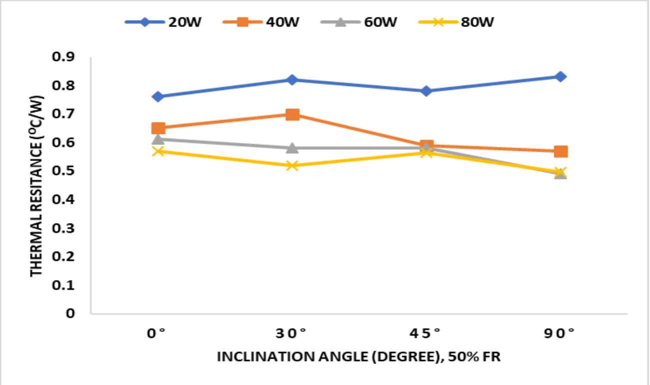

Fig. 3. Variation of thermal resistance in Benzene with different Inclination angles at 50% Fill ratio

Thermal resistance of butanol at heats input at Inclination angles with fill ratio 50%.

As heat input rises, at an inclination of 45 degrees thermal resistance reduces, in Fig. 3. The variation in graphs by increasing heat input thus decreasing thermal resistance. More fluid becomes vaporized as the heat input is increased, so that heat can efficiently be expelled by phase change. The vaporization rate is relative to the volume of heat input, so that thermal resistance reduces the molecular bonding.

3.1.3. Comparison of Water and Benzene

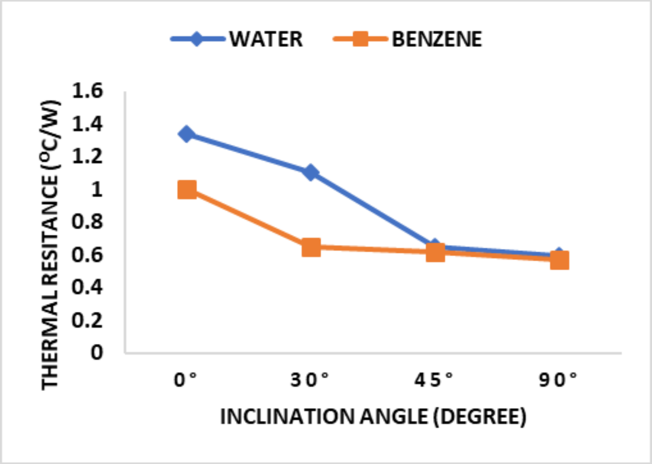

Fig.4. Variation of thermal resistance in DI water and Benzene with different heat input

Thermal resistance of two working fluids are plotted, benzene is low with compare to water Fig. 4. The surface tension is high at higher temperatures due to the self-reverting fluids because it sticks to the walls of the heat pipe so that more heat is being transmitted.

3.2 CFD simulation of PHP

Procedure for CFD numerical analyzes, it requires five stages such as: Geometry creation, Grid generation, Fluent setup, Solution methods, Results.

The 3-D analysis of fluid flow and heat transfer for the tube is performed on ANSYS FLUENT software based on the control volume method.

3.2.1 Geometry Creation

The PHP design consists of 5 copper tubes of inner2 mm diameter and outer 3.6 mm diameter, the channel's total length is 283 mm respectively, and the copper pipe is used. For geometry simulation, the copper domain is not measured, as only the subject of the liquid domain of concern. The PHP consists of three portions: evaporator, adiabatic and condenser Region as length 42mm, 50 mm and170mm.

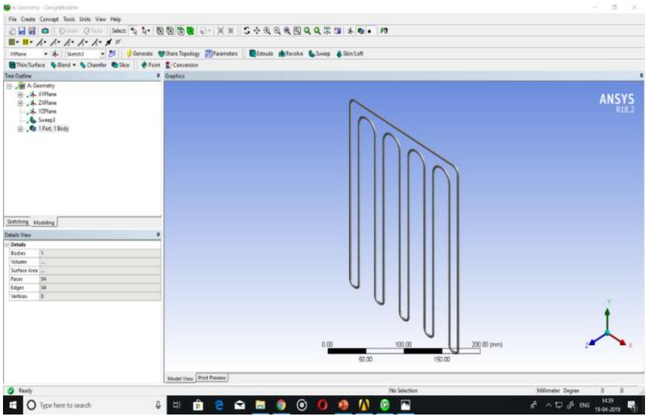

Fig4.1: Geometry of pulsating heat pipe created in ANSYS CFX

3.2.2 Meshing

The automated mesh form in this scenario, the program is used; furthermore, the mesh is manually designed as shown in Fig.4.1. With this control setting, the mesh dimension is 0.0002 m, meshing is generated with 343419 numbers of nodes and 356658 numbers of elements.The automatic mesh shape method will be selected in this case as shown in Fig 4.2. The mesh is done manually.

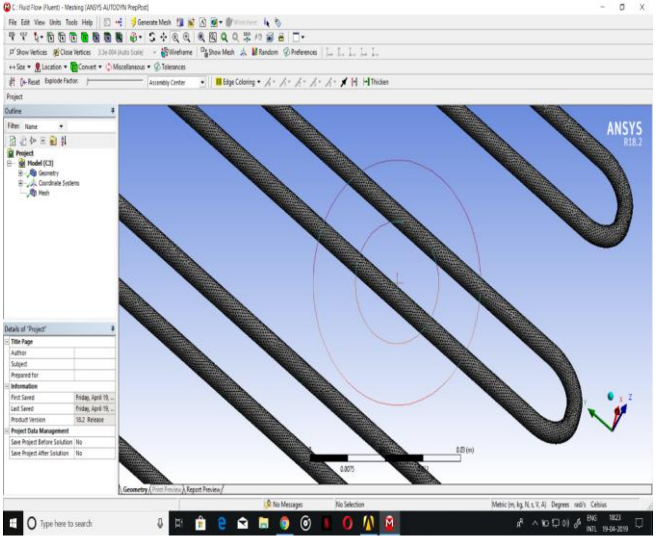

Fig.4.2 depicting discretization in a part of the geometry

3.2.3 Analysis

A steady state analysis is performed on CFX in the first phase. While most of the thermal analysis with phase change phenomenon is predominantly a transient case, steady state analysis is performed just to make sure this case is either a steady state analysis or a transient one. Most of these aspects are included in the transient simulation. Usually this means that the stable state system has a simpler convergence because model and some temporary non-linearity is eliminated, but in some systems this non-linearity aids convergence (but this is rare).The transient state analysis of simulations in different ways, viz. Time steps, time steps for running and methods of adaptation were made as a trail. Given the number of meshes in the domain and the fluid flow speed, it was not feasible to use larger time measures. There was no external coupling of the solver, as all parameters are not required, including from geometry. In the same file meshing to the domain settings was performed; and the solver selected, i.e. CFX is able to solve the problem completely. For PHP's overall geometry, mainly three domains were described, i.e. Adiabatic area, condenser and evaporator. In contrast, the adiabatic area is split into two regions to integrate from the foundation evaporator the water level increase in the PHP. The air from the base of the adiabatic zone fills 50% by volume for the water ratio. The Evaporator region were given various heat fluxes, 20W, 40W, 50W and 80W.Heat flux is zero for the Adiabatic region. For CFD analysis, the VOF template is used. The walls are of no types of slip walls in all domains. Both walls are expected to be smooth. The thermal boundary state of 302 K is set in the condenser. Similarly, in adiabatic areas, the boundary state is adiabatic, describing the heat flux value as 0W/m²K.Likewise in the area of the evaporator, where the external heat supply is associated, maximum constant heat flux of 1000W/m²K.In terms of precision and robustness, the k-epsilon offers a good compromise. The turbulence system k-epsilon applies generic two-equation models within CFX. Those two algorithms are nothing more than the transport equations for turbulent kinetic energy (k) (or partial differential equations) and dissipation (ε) and uses a modular approach to wall feature to improve robustness and precision with a very strong near-wall mesh. All velocity components (u, v and w) are provided in the condenser region as 0 m/s and relative pressure as 0 kPa. Considering that the condenser will have less volatile flow as there is only high-pressure water in the initial state at low temperature. Upon heating the evaporator, if the thermal boundary state of the differential mixtures is set at a temperature of 302 K, shown below Fig 4.3.

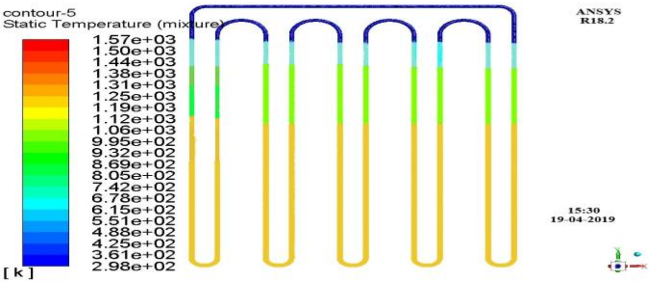

Fig.4.3 Showing the phase distribution in PHP

All velocity components are set to 0 m / s and relative pressure to 0 KPa in the evaporator as well. The walls are of no styles of slip walls in all domains. Likewise, the boundary state is adiabatic in adiabatic zones, which determines the heat flux frequency as 0 W / m²K. Similarly, in the region of the evaporator, where the external heat source is attached, total constant heat flux of 1000W/m²K value.

3.3 Comparison of CFD and Experimental Data

The results of experimental and CFD simulation of water and benzene CLPHP are shown in Fig. 5 & Fig. 6.

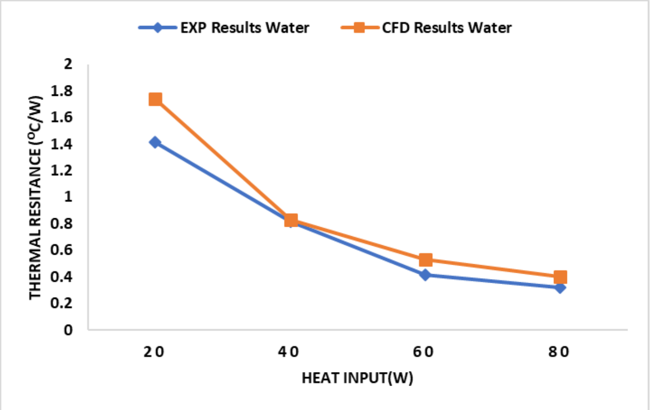

Fig. 5. Variation of thermal resistance in DI water with different heat input, fill ratio 50%

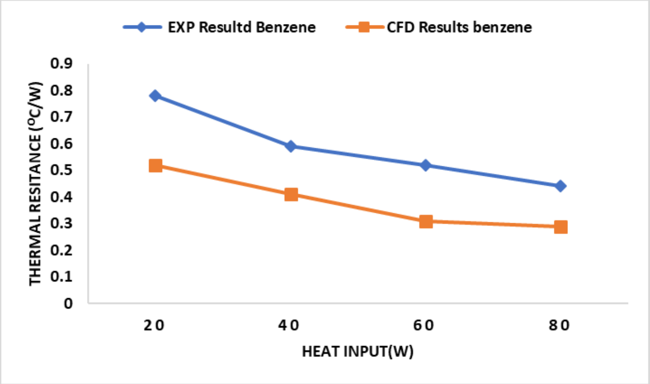

Fig. 6. Variation of thermal resistance in Benzene with different heat input, fill ratio 50%

Results indicate a near accord i.e. the research and simulation outcomes rates are the similar with negligible errors (around 5%-10%). In both cases, thermal resistance initially decreases heat input is drastically up to 40Wand subsequently decreases steadily. The reasons for simulation variation and experimental results are convection heat loss in the section of condenser and evaporator, as well as some errors in CFD simulation such as discretion error.

IV. CONCLUSIONS

Study of computational fluid dynamics (CFD) and experimental work are used to study the performance of five loop pulsating heat pipe (PHP). The analysis was done for 50% filling ratios by varying heat loads at evaporator section. The PHP model is created, simulated using ANSYS FLUENT. The variation of thermal resistance was observed for varying heat load.

The experimentation was conducted for different working fluids (i.e., water, benzene at different heat inputs (i.e., 20W, 40W, 60W, 80w) the variation of thermal resistance was observed.

The summary of the present work as follows.

- The experimentation done at different heat inputs i.e. 20W, 40W, 60W,80W with different working fluids (i.e., water, benzene. show that as heat input increases thermal resistance value decreases.
- In vertical operating mode the thermal resistance is high at 20W heat input for both water and benzene. Then it gradually decreases at 40W and 80W for water and for benzene it decreases up to 40w and slightly increases at 80w as shown in graphs.
- For Benzene the Thermal resistance in 45^0 inclined case is considerably less when compared to other operating modes.
- At low heat input up to 40W, the thermal resistance decreases rapidly and the PHP performance is more sensitive to the inclination angle whereas high heat input i.e. above 40W, the thermal resistance decreases smoothly and less independent to the inclination angle.
- The Thermal performance is considerably improved from vertical to inclined position. This is because, higher heat fluxes allow the fluid to evaporate quicker compared to the lower heat flux hence shooting up the bubble train.

- From present study, we can observe that Benzene PHP has given the better thermal performance when it is operated at 45^0 inclination by comparing with water PHP.
- Maximum thermal performance is observed for Benzene PHP at 45^0 inclination.
- CFD results are compared with the experimental results at a fill ratio of 50% the PHP is exhibiting better heat transfer characteristics, in between the heat inputs considered in the analysis.

REFERENCES

1. H. Akachi, Structure of a heat pipe, U. S. Patent 4921041 (1990).
2. H. Akachi, F. Polášek and P. Štulc, Pulsating heat pipes. Proc. of the 5th Int. Heat Pipe Symposium (1996) 208-217.
3. M. Shafii, A. Faghri and Z. Yuwen, Thermal modeling of unlooped and looped pulsating heat pipes, ASME Journal of Heat Transfer, 123 (2001) 1159-1172.
4. B. Tong, T. Wong and K. Ooi, Closed loop pulsating heat pipe, Applied Thermal Engineering, 21 (18) (2001) 1845- 1862.
5. M. Shafii, A. Faghri and Z. Yuwen, Analysis of heat transfer in unlooped and looped pulsating heat pipes. International Journal of Numerical Methods Heat Fluid Flow, 12 (5) (2002) 585-609.
6. S. Khandekar, N. Dollinger and M. Groll, Understanding operational regimes of closed loop pulsating heat pipes: An experimental study, Applied Thermal Engineering, 23 (2003) 707-719.
7. S. Khandekar, P. Charoensawan, M. Groll and P. Terdtoon, Closed loop pulsating heat pipes part B: visualization and semi-empirical modeling, Applied Thermal Engineering, 23 (16) (2003) 2021-2033.
8. S. Khandekar and M. Groll, An insight into thermohydrodynamic coupling in closed loop pulsating heat pipes, Int. J. Thermal Science, 43 (1) (2004) 13-20.
9. P. Pachghare and A. Mahalle, Thermal performance of closed loop pulsating heat pipe using pure and binary working fluid, Frontiers in Heat Pipes, 3 (2012) 033002.
10. P. Pachghare and M. Mahalle, Effect of pure and binary fluids on closed loop pulsating heat pipe thermal performance, International Journal of Procidia Engineering, Elsevier Publications, 51 (2013) 624 -629.
11. S. Nagvase and P. Pachghare, Experimental and CFD analysis of closed loop pulsating heat pipe with DI-water, International Journal of IEEE, (10) (2013)185-190.
12. Pramod R. Pachghare and Ashish M. Mahalle,Thermo-hydrodynamics of closed loop pulsating heat pipe: an experimental study 528 (8) (2014) 3387~3394
13. Byeongchan Kim, Longnan Li, Jihoon Kim, Daejoong Kim A study on Thermal Performance of Parallel Connected Pulsating Heat Pipe 2017,S1359- 4311(17)33716-X
14. M. EbrahimiDehshali, M.A. Nazari, M.B. Shafii* Thermal performance of rotating closed-loop pulsating heat pipes: Experimental investigation and semi-empirical correlation 2017,International Journal of Thermal Sciences 123 (2018)14e26
15. J.Venkata suresh ,P. Bhramara,CFD Analysis of Copper Closed Loop Pulsating Heat pipe 5 (2018) 5487–5495

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