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Numerical Simulation of Oscillating Heat Pipe Using VOF Model

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ABSTRACT

In this study, a CFDmodel was built to simulate the two-phase flow and heat transfer inoscillating heat pipe. Water was used as the working fluid with filling ratio equal to 50%. The volume of the fluid (VOF)method in ANSYSFLUENT was used in the simulation. The evaporation-condensation processes were dealt with by adding a user-defined function (UDF) to the FLUENT code. The simulationresults where compared with experimental data obtained at the same condition. The simulation was successfulin reproducing the heat and mass transfer processes in the oscillating heat pipe. Fairly agreement was obtained between CFD results and experimental data.

INTRODUCTION

Oscillating heat pipes (OHP's, also called Pulsating heat pipes PHP's) are an important heat transfer devicesworking without wick. A common design for OHP is a capillary-sized tube forming a closed, continuous flow loop and containing a liquid and its vapor, which is heated at one end (evaporator), while cooled at the opposite end (condenser), as shown in Fig.1. If the capillary diameter is not too large, the fluid distributes itself into an arrangement of liquid slugs separated by vapor bubbles (often referred to as plugs). During operation, heat input to the evaporator expands existing bubbles and/or nucleates new bubbles, driving liquid and bubbles toward the cooler condenser region, where vapor bubbles contract or collapse via condensation. The evaporation/condensation cycle provides the motive force for the convective motion, though heat is mainly transferred sensibly by the movement of hot liquid from evaporator to condenser. These devices are called "pulsating" heat pipes because the evaporation/condensation process happens as a non-equilibrium chaotic process, whose continuous operation requires non-equilibrium conditions to exist in some, but not necessarily all, of the parallel channels at any given instant of time (Fasula, C., 2009).

The parameters affecting the operation of OHPhave been summarized by (Nagvase, S.Y. and P.R. Pachghare, 2013) as: internal diameter, working fluid,total tube length, length of condenser, evaporator and adiabatic sections, number of turns or loops and inclination angle.

The most important parameter that affecting the performance of OHP is the internal diameter, because it contains the condition for discrete bubbles and liquid slugs sustained in the tube when the fluid is stationary and thus determines whether confined bubbles exist during the two-phase flow in a channel. The internal diameter must be less than a critical value to generate the pulsating effect (Khandekar, S. and M. Groll, 2003).

Several researchers have been studied the effect of the above operation parameters, among them; Zirong *et al* (2013), studied the heat transfer mechanism of miniature oscillating heat pipes (MOHPs) and predict the heat transport capability of MOHPs.Both evaporation section and condensation section length were 20 mm, the

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internal diameter was Di=1.3 mm. Water was used as the working fluid.The volume of fluid and mixture model in FLUENT were used for comparison in the simulations.

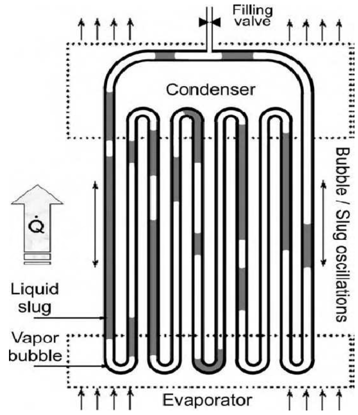


Fig. 1: Oscillation heat pipe[1]

The phase change process in a MOHP was deal with by adding a user-defined function source term in each phase. The continuum surface force model was used to consider the effect of surface tension. The result showed that the mixture model was good to simulate the two-phase flow in a MOHP, but did not visualize the internal phenomena of evaporation condensation and phase change inside the MOHPs.The simulation with unsteady model was successful in reproducing the two-phase flow process in a MOHP, including the bubble generation in evaporator section and the oscillations caused by the pressure difference.

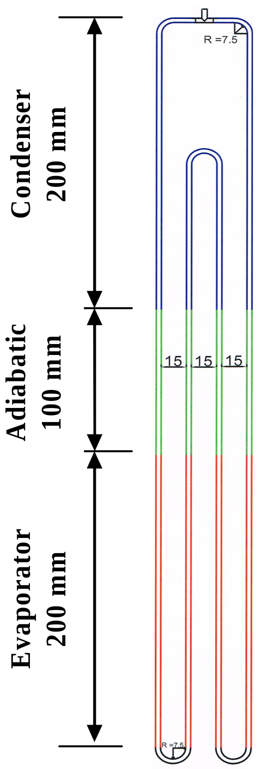
Mathematical and physical model of OHP was built by Wanga *et al* (2012), to simulate the process of flow and heat transfer in vertical bottom heating oscillating heat pipes. The four-turn OHP was selected as a typical shape. Pure water was used as working fluid. The operational orientation was vertical bottom heating mode.Two types of copper tubes with outer diameter Do= 2.5 mm and 3mm, inner diameter Di= 1.3 mm and 1.8 mm were used.The filling ratio was controlled around 50±5%. The length of evaporation section was 20 mm, the same as the condensation section; and the condensation section was cooled by water 25±0.05°C. Mixture model was used for flow and heat transfer simulation. The user defined function was added to calculate mass and energy transport, in order to achieve heat transfer process of evaporation and condensation. The result showed that the numerical simulation was successful to reproduce the behavior of the internal flow of OHP, including vapor generation in evaporation section, oscillation phenomena caused by the pressure difference and heat transfer due to oscillation. Comparing with the experimental tests, the simulation results agreed fairly well.

Xiangdong *et al*(2014),investigatedathree-dimensional numerical on the vapor–liquid twophase flow and heat transfer performance in a flat plate oscillating heat pipe FP-OHP. The FP-OHP is fabricated on aluminum plate as a meanderingclosed loop (square cross-section 2 mm × 2 mm) with dimensionof 110 mm × 140 mm. It is set vertically and contains three parts:evaporator, condenser and adiabatic section. It was charged with ethanol as the working fluid with filling ratio 30%, 50%, and 70%, respectively and for differentheat load.In order to verify the mathematical model, anexperimental verification was conducted to investigate thermo-hydrodynamics phenomena inside the FP-OHP. The condenserwas cooled by a cold aluminum blocksupplied with cooling water, and the inlet cooling water temperature was kept at 20 ± 0.1◦C.It was indicated that the dispersed bubblesin FP-OHP are generally produced by the nucleate boiling in evaporator and the condensation of shortvapor plugs in condenser. The short plugs are usually formed by the self-growth and coalescence ofdispersed bubbles. Additionally, the long plugs occur due to further coalescence of the short plugs. Theproportion of dispersed bubbles decreases and then increases with increasing heat load, and the averagesize of total bubbles is inversely proportional to the filling ratio. The optimal filling ratio for the thermalperformance of FP-OHP is determined by most adequately combining the advantages of the sufficientbubbles pumping action for driving the heat transport along with the motion of working fluid, and thesensible heat transfer of the liquid, which is shown around 50%.

The purpose of the present study is to develop a CFD model to cover the details behavior of two phase flow and heat transfer accompanied the operation of an OHP with water as a working fluid, especially the phase change inside the OHP, which did not visualized before.

Model Geometry and Computational Mesh:

Two dimensional model was built to simulate the two-phase flow and heat transfer phenomena in oscillation heat pipe, the total length is 2050 mm shaped as two turns with 500mm length, for each turn, its set verticallyand including three sections evaporator with 200 mm length, adiabatic with 100mm length and condenser with 200 mm length. Theinside diameter of the capillary tube is 3mmand outside diameter 6mm, as shown in Fig.2.The working fluid is water and the filling ratio 50%from the total volume of the OHP.



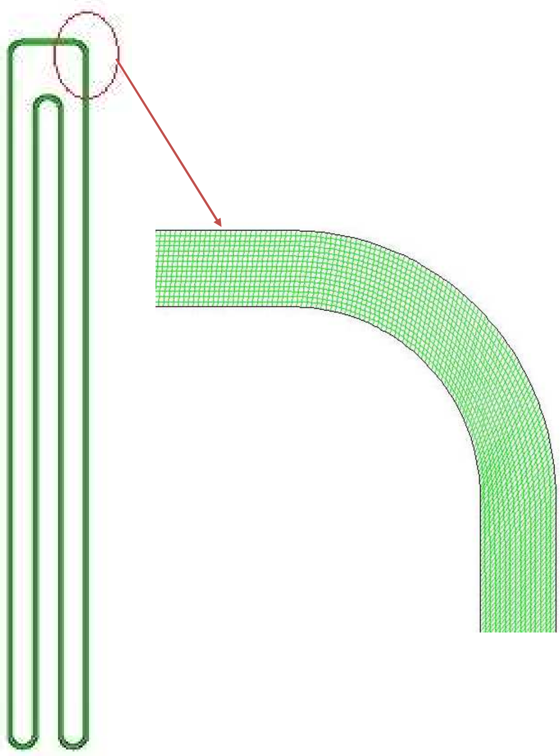


Fig. 2: Schematic of Oscillation Heat Pipe with its Dimension. **Fig. 3:** Mesh of Oscillation Heat Pipe.

The geometry was constructed and meshed using the GAMBIT grid generationSoftware, The fluid regions contain 153,525 Quad cells, as shown in Fig.3.

CFD Model:

A two-dimensional physical model was developed to simulate the internal flow and heat transfer in OHP.In this model, the commercial code ANSYS FLUENT 14.5 and the volume of fluid (VOF) method has been applied.VOF is used for more than two immiscible fluids, where their interface position is of importance.It has only one set of momentum equation and single energy equation for the entire system.The only disadvantage is that requires fine mesh.The VOF model relies on the fact that each cell in the domain is occupied by one phase or combination of the two phases. In other words if α_l is a volume fraction of liquid and α_v is a volume fraction of vapor, the following three conditions are possible:

- α_l=1 ; the cell is fully occupied by liquid
- α_l=0 ; the cell is fully occupied by vapor
- 0<α_l<1 ; the cell is at interface between the liquid and vapor phases.

When the third condition occurs, the volume fraction of all phases sum to unity (Bandar Fadhl, *et al.*, 2013).

Governing Equations for VOF Model:

The governing equations of the VOF formulations on vapor-liquid flows with phase change are as follow(ANSYS FLUENT, 2010):

$$\frac{\partial \alpha_l}{\partial t} + \vec{v}.\nabla\alpha_l = -\frac{S_m}{\rho_l} \quad (1)$$

Where;ρ is the density, v is the velocity and t is the time,S_m is the source term of mass used to calculate the masstransfer during evaporation and condensation.

$$\frac{\partial}{\partial t}(\rho\vec{u}) + \nabla.(\rho\bar{u}\vec{u}) = \rho\vec{g} - \nabla P + \nabla.\left[\mu(\nabla\vec{u} + \nabla\vec{u}^T) - \frac{2}{3}\mu\nabla.uI\right] + F_{CSF} \quad (2)$$

Where$g$ is the acceleration of gravity, P is the pressure, and I is the unit tensor.

The forces acting in the fluid are gravitational,pressure, friction and surface tension. In order to considerthe effect of surface tension along the interface between the twophases, the continuum surface force (CSF) model has been added to the momentum equation:

$$F_{CSF} = 2\sigma_{lv}\frac{\alpha_l\rho_l C_v\nabla\alpha_v + \alpha_v\rho_v C_l\nabla\alpha_l}{\rho_l + \rho_v} \quad (3)$$

whereσ_{lv} is the surface tension coefficient and C is the surface curvature.

The energy equation for the VOF model:

$$\frac{\partial}{\partial t}(\rho e) + \nabla.(\rho e\vec{u}) = \nabla.(k.\nabla T) + \nabla.(P\vec{u}) + S_E \quad (4)$$

Where S_E is the energy source term used to calculate the heat transfer during evaporation and condensation.A user-defined function (UDF)has been used with the existing FLUENT codetocalculate the mass and heat transfer betweenthe liquid and vapor phases during the evaporation andcondensation processes.The source terms in thegoverning equations, particularly the continuity and energyequations for this UDF was proposed by De Schepper, *et al*. (2009)

It is assumed that the phase change occurs at saturation temperatureT_{sat}. Therefore the mass source term iscalculates as follow:

$$S_{m,l\to v} = \begin{cases} -0.1\ \rho_l\alpha_l\frac{T_l - T_{sat}}{T_{sat}} if T_l > T_{sat} \\ o if T_l < T_{sat} \end{cases} \text{Evaporationprocess} \quad (5)$$

$$S_{m,v\to l} = \begin{cases} 0.1\ \rho_v\alpha_v\frac{T_{sat} - T_v}{T_{sat}} if T_v < T_{sat} \\ o if T_v > T_{sat} \end{cases} \text{Condensationprocess} \quad (6)$$

where$T_l$and T_v are the liquid and vapor temperatures, respectively, and $\alpha_l$and α_v are the volume fraction of the liquid and vapor phases, respectively.

Energy sources S_E in the energy equation used are determined by multiplying the calculated mass sources in Eq. (5) and Eq. (6) by the latent heat of evaporation for the working fluid, and can be expressed as follows;

$$S_{E1} = S_{m,l\to v}\,.LH \quad (7)$$

$$S_{E2} = S_{m,v\to l}\,.LH \quad (8)$$

Where LH is the latent heat of evaporation.

Boundary and Operating Conditions:

The heat flux in evaporator section depended on the power input; a constant temperature is defined at the wall boundaries of the evaporator section (hot-wall), a zero heat flux is defined as boundary condition on the adiabatic section (adiabatic-wall). The condenser section is cooled as a result of heat released when vapor condenses. It is assumed that the condenser is cooled by water, applied constant temperature along the condenser section (cold-wall) (Richard, L., *et al*., 2014).According to the experimental data, Fig.4shows the details of boundary conditions.Thetime step is set equal to 0.0001.The time step has been selected based on the global courant number($\text{courantnumber} = \frac{u\ .\Delta t}{\Delta x}$), where Δt is the time step, Δx is the space between the cell and u is the velocity. For VOF models, the maximum Courant number allowed near the interface is 250. For a time step that been selected, the Courant number is less than 3.

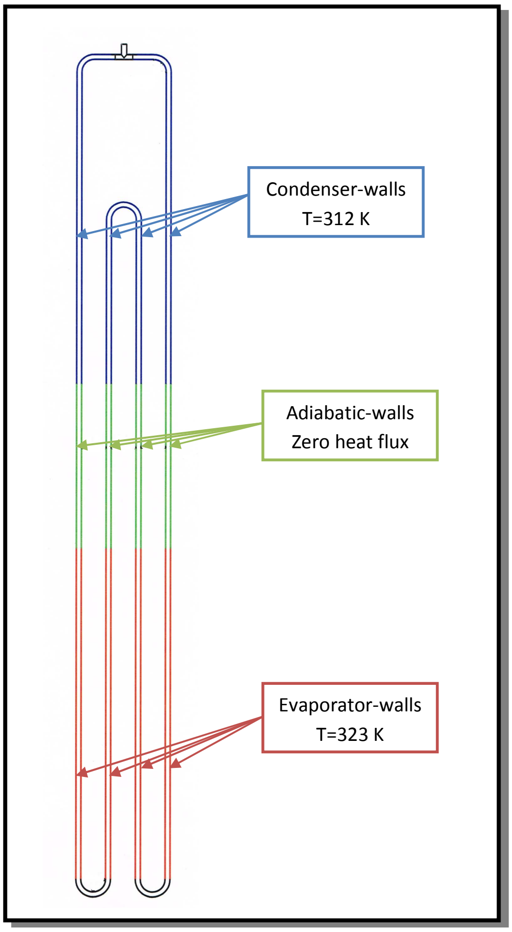
Water vapor is defined as the primary phase (vapor) and water liquid is defined as the secondary phase (liquid),the liquid density is given as polynomial; (Xiangdong Liu, Yongping Chen, 2014)

$$\rho_l = 859.0083 + 1.252209T - 0.0026429\,T^2 \quad (9)$$

The effect of surface tension was considered using the following equation;

$$\sigma_{lv} = 0.09805856 - 1.845\times 10^{-5}T - 2.3\times 10^{-7}T^2 \quad (10)$$

In all cases, the continuum surface force (CSF) model was used for analyzing the effect of surface tension.



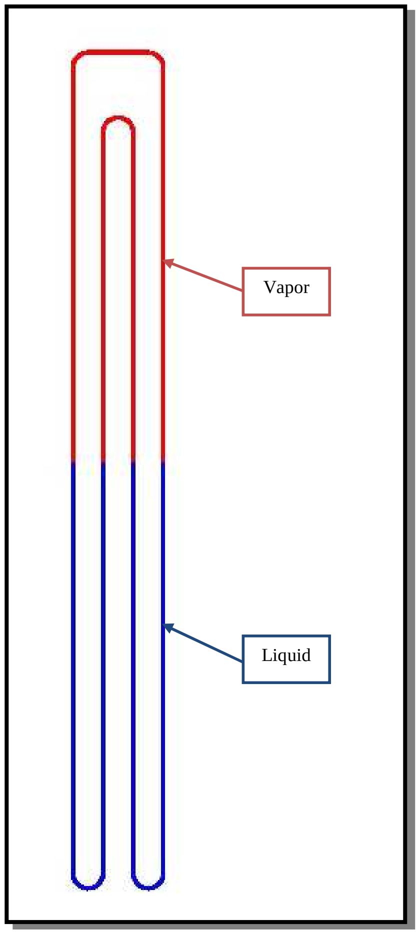


Fig. 4: Boundary conditions of OHP.

Fig. 5: Solution initialization of OHP

In the model solution, the pressure–velocity coupling was SIMPLE. The relaxationfactors of pressure and momentum were set to 0.3. Under pressure interpolation schemewas set to PRESTO; momentum and energy changed to Geo-Reconstract.The initialized liquid filling ratio was 50%, as shown in Fig.5.The numerical computation is considered to have converged when the scaled residual of the mass component and velocity is less than $10^{-4}$and energy less than 10^{-8}, as shown in Fig.6.

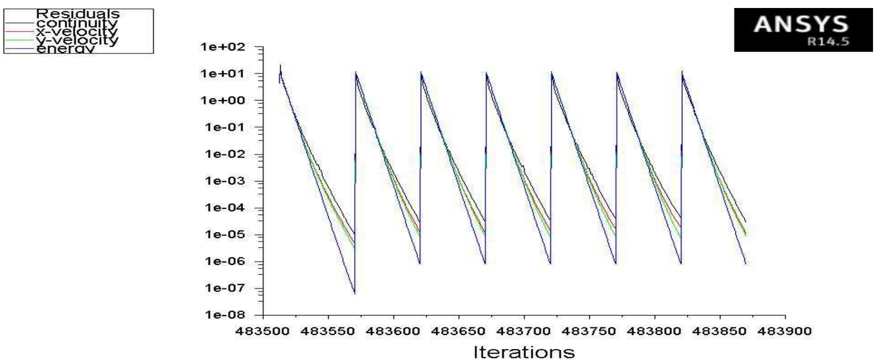


Fig. 6: Convergence history for continuity, momentum, and energy.

RESULT AND DISCUSSION

In order to understand the heat transfer process during the OHP operation, the phases and temperature contours at different times have been recorded and compared with the experimental results in the following paragraphs:

Experimental flow visualization:

In the visual observation, the two-turn OHP with heating power of (95.0 W) was selected for comparison. At the beginning, most liquid slugs located in the bottom of channels in vertical mode due to gravity effect. When the heat is added to the evaporator section the fluid began to move up till reach the condenser section, as shown in Fig.7(a), where its condensed and back to the evaporator section.Here the oscillating motion is began, as the temperature of evaporator section increased the bubbles began to generate, as shown in Fig.7(b), and the flow began to oscillate in the channel and circulated in an anticlockwise direction, as shown in Fig.7(c).Fig.7(d), show the flow back from the condenser section in a fast movement due to two forces capillary plus gravity. The flow inside the channel is liquid slug and vapor bubble, as shown in Fig.7(e). As the time increased the flow began to change from bubble flow to slug flow as shown in Fig.7(f).The bubble pumping action plays an important role in driving the circulation of fluidinside oscillating heat pipe, which is dependent on theformation of vapor bubbles and liquid slugs; also the sensible heat transferof liquid has an important function in the total heat transfer inthe OHP, the circulation of flow inside the heat pipe showed in Fig.7(g-i).

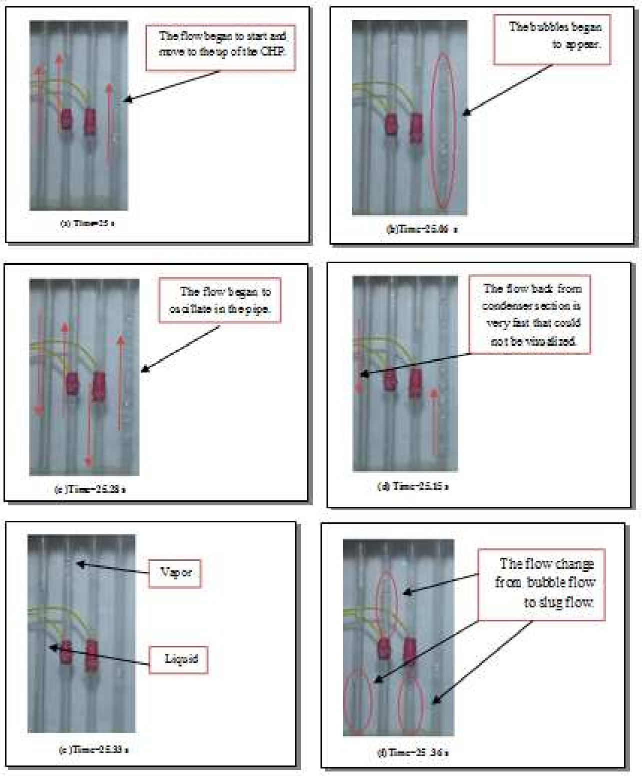


Fig. 7: Internal flow visualizationwith heating power = 95.0 W.

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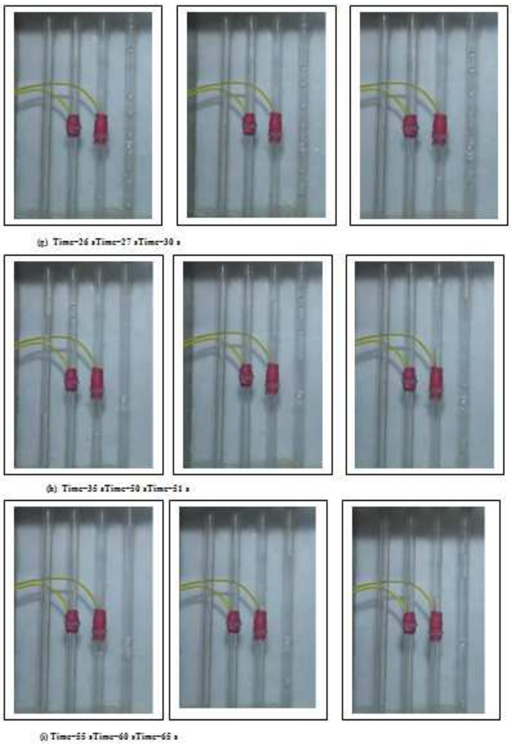

Fig. 7: Continue.

- ***Flow visualization of CFD simulation:***

In the simulation case, the two-turn OHP with same sizes of the visualization case was selected to build the two dimensional model. The heating power was 95.0 W. As shown in Fig.8 and 9, the contours of temperature and phase change in OHP operation had been recorded, which indicated the process of heat transfer and mass transfer. As shown inFig.8 the region of high temperature in evaporation section expanded, corresponding to the accelerated growth of bubbles Fig.8 (0.1 s). As time increased the bubbles jetted across the evaporator section to the condenser section in Fig.8(1s). As shown in Fig.8 (2 s), the hot and cold region exchanged. The high-temperature region appeared in the condenser section, while the evaporation section was cooled down, corresponding to bubbles condensed into liquid in the condenser section. With the help of gravity and capillary forces, the condensed liquid returns to the evaporator section. By then, one cycle of heat transfer from evaporator section to condenser section was finished. After that, a new cycle began, and the stable temperature oscillation occurred. The cycle period of the subsequent oscillations became smaller.

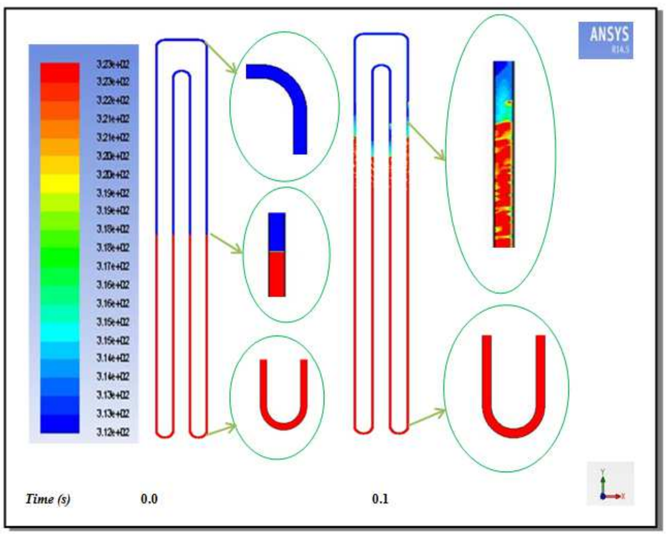

Fig.8: Temperature contours at different times in OHP operation process.

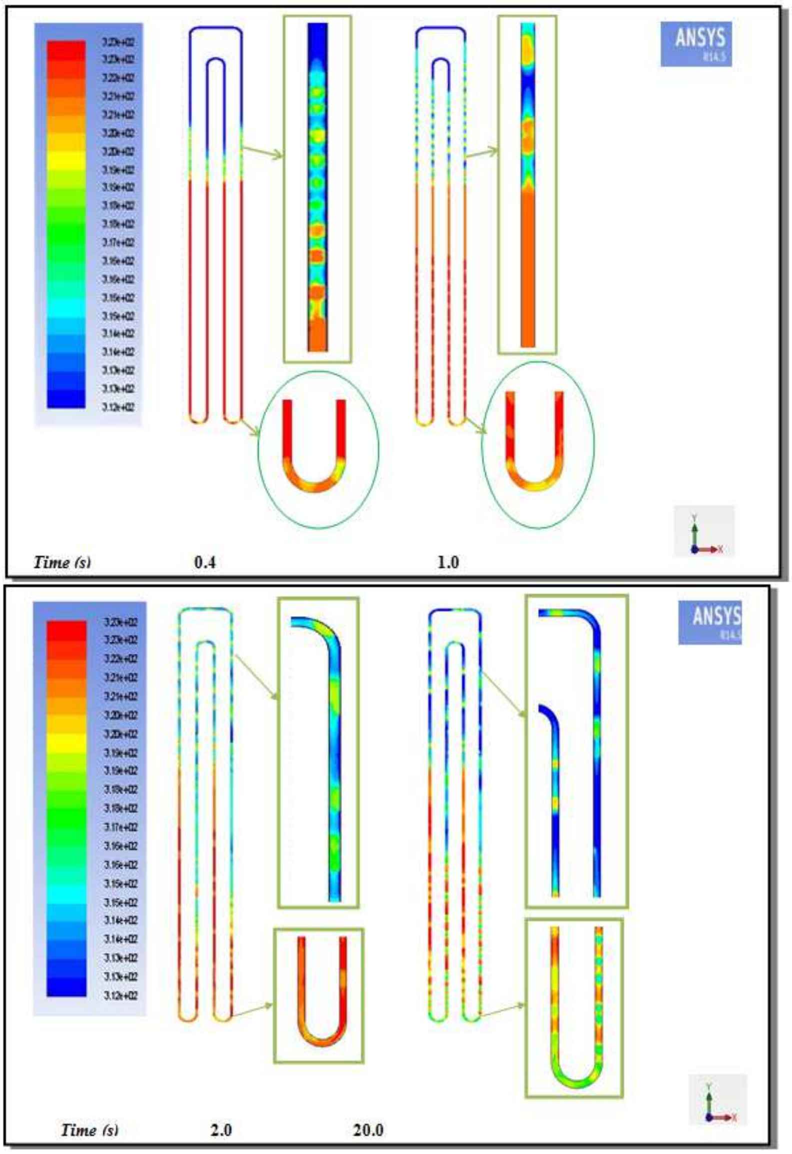

Fig. 8: Continue.

Fig. 9show the volume fraction contours in the OHP, for a heating power of 95.0 W. A red color illustratesvapor volume fraction = 1,while a blue color illustrates the liquid phase, vaporvolume fraction = 0.At the beginning ofthe process, the liquid pool filled half of the OHP. Whenthe liquid reached the boiling temperature, the liquid starts toevaporate and phase change occurs. Thiscontinuous evaporation of liquid results in a decrease in the liquidvolume fraction and an increase in vapor volume fraction. Atthose positions where the liquid evaporates, bubbles are formedand transported towards the top region of the liquid pool.Following the above process, the vapor is transported upwardto the condenser. As the vapor reaches the condenser's wall cooled; the vapor condenses along the coldwalls. So the evaporation-condensation process proceeded and the oscillation process occursand the working fluid in the OHP oscillates.

Numerical Simulation Compared with Experimental Results:

Fig.10 show theexperimental record of temperature variation in evaporator and condenser sections of OHP with time at input heating power 95.0 W. The two temperatures start from the ambient temperature and then began to increase as the heat added to the evaporator section. The temperature increased until the start-up point and then the oscillating motion in the heat pipe began. The system reaches a steady state with a temperature between 52 °C and 48 °C after certain time. The oscillation in temperature that recorded at any point with time at evaporator and condenser sections caused by the vapor bubbles and liquid slugs oscillation in each sectionsdue to the pressure difference in the system, that preventing the system from reaching a dry-out condition.

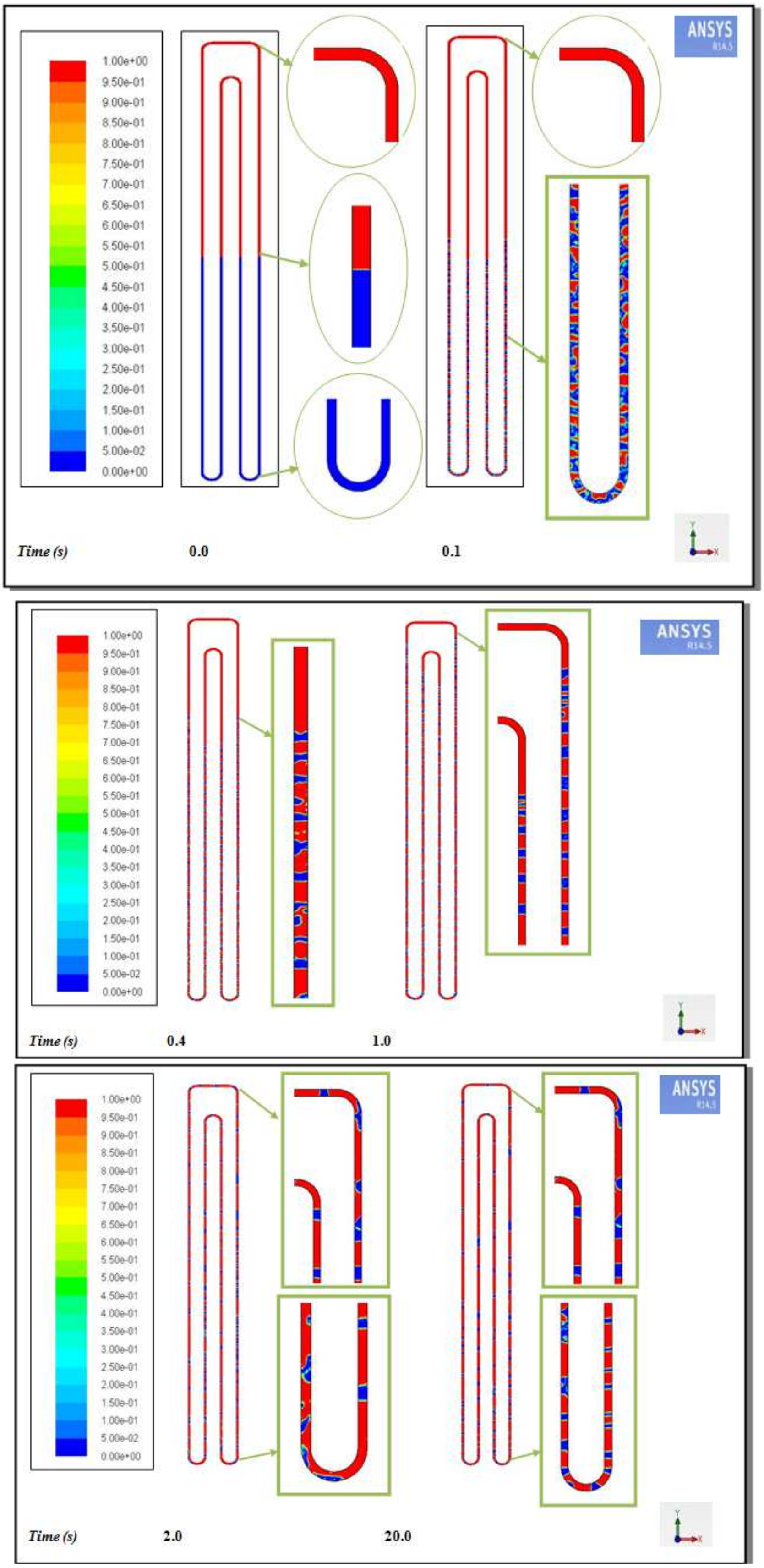

Fig. 9: Contours of volume fraction in OHP at several times.

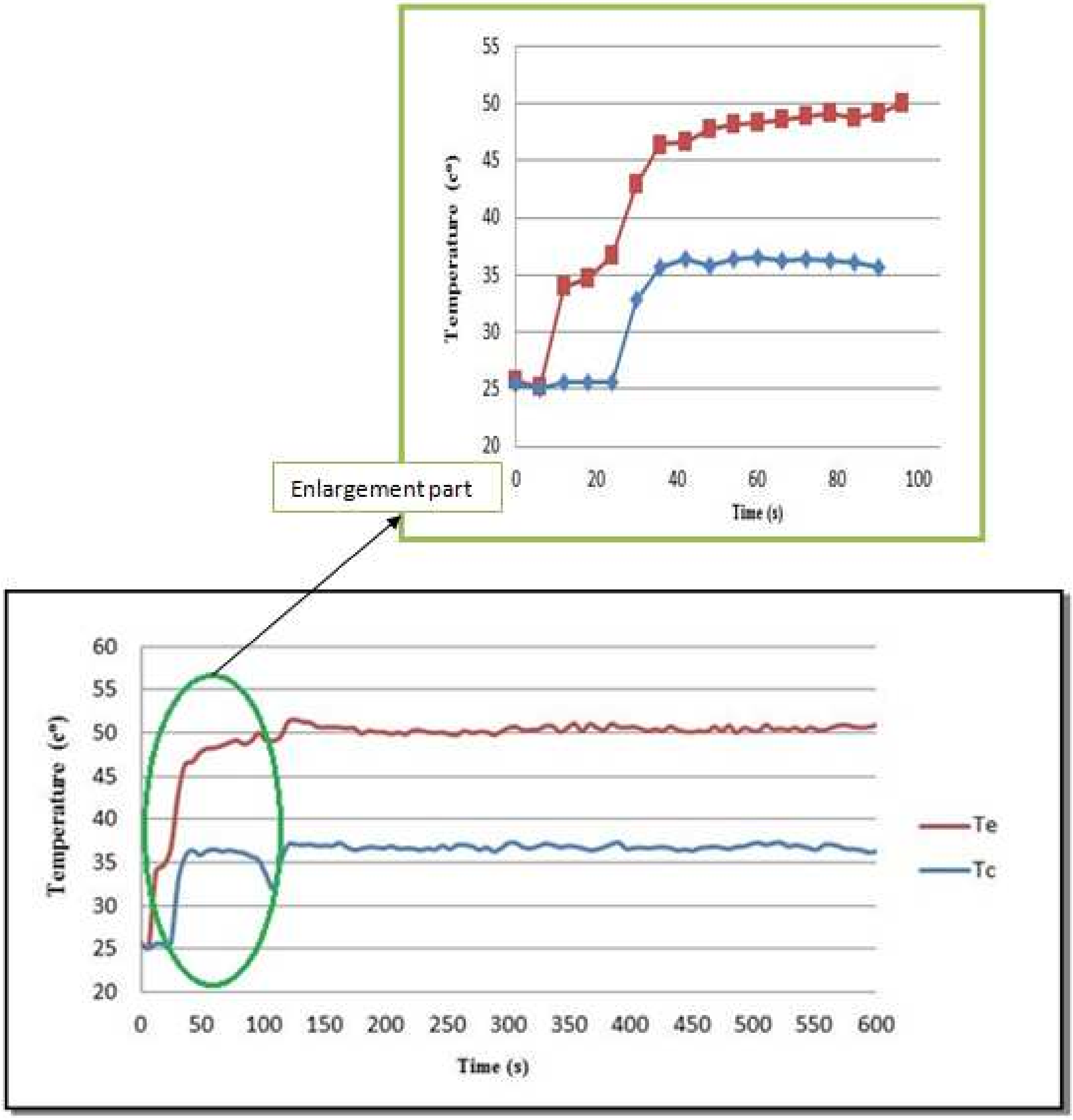

Fig. 10: OHP experimental temperature at 95.0 W.

Fig.11show the2D simulation record of temperature variation in evaporator and condenser sections of OHP with time at input heating power 95.0 W. The simulation time is limited to 60 second, because of the low time step 0.0001 that has been chosen for convergence restriction by courant number value. The computation time of than 60 second is about 3 months of CPU time.

As can been seen in Fig.11 the temperature oscillation range was consistent with the experimental record in Fig.10. Theaverage values of the experimental and simulation temperatures in the evaporator sectionsareabout (50^{o}C) and (56^{o}C) respectively; while in the condenser sectionsare about (36^{o}C), for both of them. Hence it can be said that the comparison of both results are fairly good.

Conclusion:

Anumerical and physical model was built to simulate the internalflow and heat transfer inside an oscillation heat pipe. Waterwas used as a working fluid with 50% filling ratio. The following was the major conclusion points:

- The volume of fluid (VOF) technique in FLUENT was successfully model the complex tow phase flow inside the OHP.
- The simulation withunsteady model was reproduce the operationof OHP, including vapor generation, vapor bubbles, liquid slugs and the oscillation phenomena caused by the pressure difference.

- The numerical simulation was found to agree fairly well with experimental data carried out for similar cases.

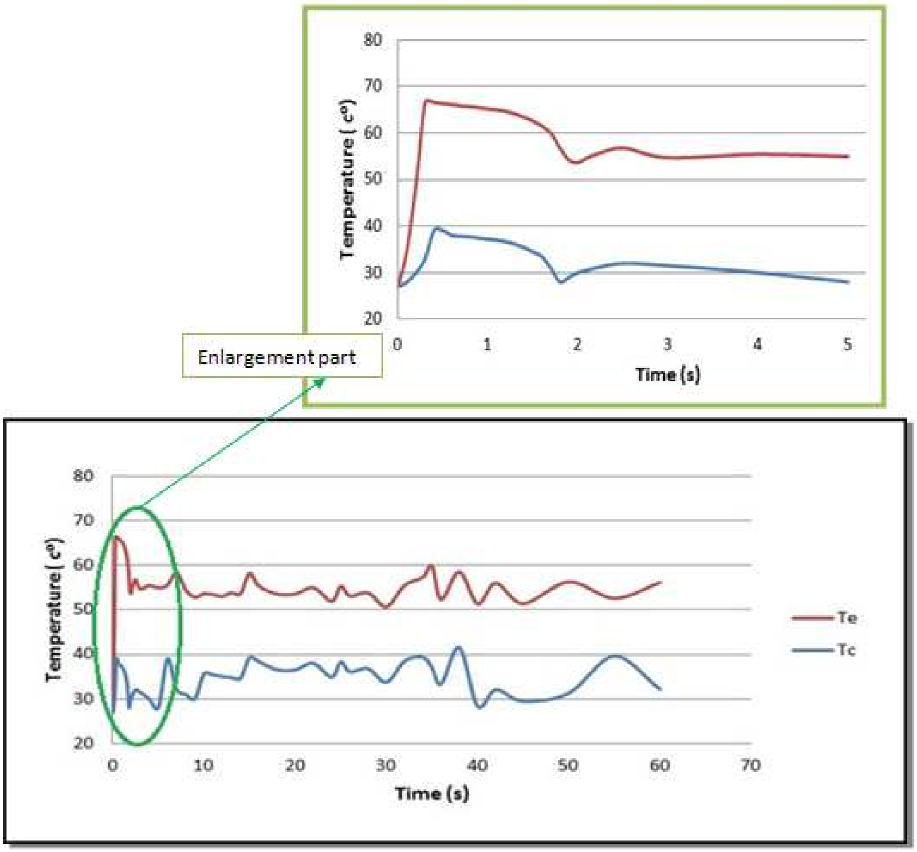

Fig. 11: Temperature record for OHP 2D simulation at 95.0 W.

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