

Numerical Simulation of Helically Coiled Closed Loop Pulsating Heat Pipe

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ABSTRACT

This paper addresses the numerical simulation of helically coiled closed loop pulsating heat pipe which is carried in ANSYS Fluent. The values of thermal resistance for constant heat fluxes vs. transient heat fluxes are analyzed. Phase change visualization after the end of simulation is carried out to observe the phenomenon in liquid at its saturation temperature and pressure. Finally, helical heat pipes are found to have thermal resistance less by 2.7K/W, 0.56 K/W, and 0.227 K/W for 8W, 40W and 80W heat inputs than circular pipes. Helical heat pipes are found more efficient than circular heat pipes.

Keywords— Helical, pulsating heat pipe, simulation, thermal resistance

crust has been declining and is towards extinction. In addition to this, the global warming caused by the use of fossil fuels has compelled the researchers to find an alternative solution which can address the problem. In the area of automobile, an alternative to combustion engine can be high voltage battery which could deliver the same amount of power and torque. Similarly, the electric generation via use of coal may be possibly replaced by the use of solar energy to produce electricity. The storage and effective transfer of thermal energy system is thought to be crucial in near future.

Heat pipes are recognized as one of the most efficient passive heat transfer technologies available. A heat pipe is a structure with very high thermal conductivity that enables the transportation of heat whilst maintaining almost uniform temperature along its heated and cooled sections. In general, heat pipes are passive thermal transfer devices able to transport large amounts of heat over relatively long distances, with no moving parts, using phase-change processes and vapor diffusion. The main structure of a heat pipe consists of an evacuated tube partially filled with a working fluid that exists in both liquid and vapor phases.

The application of the heat pipes is widely known. The high voltage battery in hybrid-electric, electric vehicles is said to have maximum efficiency at the temperature of 35 °C. In order to maintain this temperature, the heat generated from the battery has to be dissipated to the surrounding which needs an efficient heat transfer medium. Heat pipes are being experimented in this field to solve the issue. Also, in order to transfer the solar thermal energy the people working in the area of cryogenics fluid is trying to build the heat pipe system. Industries are challenging the heat transfer community to provide solutions to enable systems that can run reliably at extremely cold temperatures. Theoretically, a heat pipe can operate at any given temperature, as long as the operation temperature is between the triple and the critical points of the working fluid utilized.

The use of heat pipe technology in heat exchange and thermal management is expanding fast due to their advantageous characteristics compared with conventional heat exchangers and temperature control systems. Advances in the design and capabilities of heat pipes have led to the development of cost-effective manufacturing techniques for both wicked and wickless heat pipes and this, in turn, is

Table 1: List of Abbreviations

Abbreviations	
Q	Heat
T _e	Evaporator temperature
T _c	Condenser temperature
R	Thermal resistance
p	pressure
Nu	Nusselt number
Pr	Prandtl number
Re	Reynold's Number
S	Source Term
Coeff.	coefficient to be tuned
v	vapor
V	Velocity
T	Temperature
h _s	condensation heat transfer coefficient
x	vapor quality
C _{p,w}	effective heat capacity of wall
k _w	thermal conductivity of wall

I. INTRODUCTION

An efficient heat transfer technology the subject of research globally. The amount of fossil fuels from the earth

creating new areas of implementation for heat pipe based systems. In addition, with advances in automation and development in material sciences, new heat pipe materials has to be investigated to deal with challenging areas that have so far been out of reach for conventional solutions, particularly in dealing with high temperature and strongly contaminated flows. The addition of heat pipes in a range of temperature scales and applications are a clear indication of the potential for such a technology, but there are significant modeling issues which are hindering the performance. The lack of advancements within commercial models will result in the increase of bespoke codes created in open source software. The use of nano fluids can be considered important for the future of heat pipes, but the validity of such claims is questionable with the lack of validation. Extensive research still needs to be conducted on Newtonian fluids before progression onto other fluids.

The search for cost effective, reliable and efficient thermal management system is constantly growing. Likewise, research works trying to establish PHP as a novel device to manage the heat flux problem in wider range of application has attracted many scientists. PHP is seen as potential heat controlling devices in the field of microprocessors, solar panel, fuel cells, space exploration and many more. Once the operational features of PHP are fully established, PHP is certain to get a commercial platform

Theoretical and Numerical Modeling of PHP

Assumptions

In order to model heat transfer and flow characteristics in the theoretical model, the following assumptions are made:

- The vapor quality in the evaporator and condenser is in linear variation along the flow direction, and no variation in adiabatic section.
- The liquid is incompressible and the vapor is assumed to behave as an ideal gas, both in the saturated state for ammonia working fluid during experiment temperature area.
- The heat losses causing by the convection and radiation heat transfer between the ambient to the tube wall of whole PHP are neglected

Governing Equations

Lee Model

The Lee model is used with the mixture and VOF multiphase models. In the Lee model, the liquid-vapor mass transfer (evaporation and condensation) is governed by the vapor transport equation:

$$\frac{\partial(\alpha_v \rho_v)}{\partial x} + \nabla \cdot (\alpha_v \rho_v \vec{v}_v) = \dot{m}_{lv} - \dot{m}_{vl}$$

If $T_l > T_{sat}$ (evaporation):

$$\dot{m}_{lv} = coeff * \alpha_l \rho_l (T_l - T_{sat}) / T_{sat}$$

If $T_l < T_{sat}$ (condensation):

$$\dot{m}_{vl} = coeff * \alpha_v \rho_v (T_{sat} - T_v) / T_{sat}$$

Where,

α_v = vapor volume fraction

α_l = liquid volume fraction

ρ_v = vapor density

\vec{v}_v = vapor phase velocity

$\dot{m}_{lv}, \dot{m}_{vl}$ = the rates of mass transfer due to evaporation and condensation

Coeff. = coefficients to be tuned which is termed as evaporation and condensation frequency

Volume Fraction Equation

The tracking of the interface(s) between the phases is accomplished by the solution of a continuity equation for the volume fraction of one (or more) of the phases. For the ‘w’ phase, this equation has the following form:

$$\frac{1}{\rho} \left[\frac{\partial}{\partial t} (\alpha_l \rho_l) + \nabla \cdot (\alpha_l \rho_l \vec{v}_l) \right] = S_{\alpha_l} + \sum_{v=1}^n (\dot{m}_{vl} - \dot{m}_{lv})$$

where, \dot{m}_{vl} is the mass transfer from phase ‘v’ to phase ‘l’ and \dot{m}_{lv} is the mass transfer from phase ‘l’ to phase ‘v’. By default ‘S’, the source term on the right-hand side of above equation is zero

The volume fraction equation will not be solved for the primary phase; the primary-phase volume fraction will be computed based on the following constraint:

$$\sum_{l=1}^n \alpha_l = 1$$

Condensation

The value of condensation heat transfer is determined by the two factors, which are condensation heat transfer coefficient h_s and vapor quality x . While h_s has a direct relationship with the velocity of fluid flow in the condenser, and the quality x relates the distribution of liquid and vapor phase. Obviously, these two items are all varied as the flow patterns change.

One of the most widely used has been the shah’s correlation, covering three flow regimes, such as the If the flow pattern is in ‘‘Slug Flow’’, the vapor will flow through the condenser section slowly, and the mechanism of condensation is used following heat transfer equation,

$$h_s = 1.32 \cdot Re_{LS}^{-1/3} \left[\frac{\rho_l (\rho_e - \rho_g) \cdot g \cdot \sin\beta \cdot k_l^3}{\mu_l^2} \right]^{1/3}$$

where, Re_{LS} is Reynolds number assuming liquid phase flowing in tube alone.

Above figure shows a schematic of annular flow pattern features at high input power in PHP tubes.

If the flow pattern is in ‘‘Annular Flow’’, the vapor flowing velocity increases, and the vapor bubble becomes long and long, the condensation heat transfer coefficient also grows. In this case, the mechanism of condensation is used with annular flow correlation,

$$Nu = 0.023Re_{ls}^{0.8} Pr_l^{0.4} [(1-x)^{0.8} + 3.8x^{0.76} (1-x)^{0.04} / Pr^{0.38}]$$

Where, the ‘Pr’ is the reduced pressure of vapor in the entrance of condenser.

Characteristics of Helical Coil

Inner diameter of pipe is 2r and the coil diameter is 2Rc (measured between the centers of pipes). Dean number is used to characterize the flow in helical pipe as same to Reynolds number for flowing pipes. Dean number (De):

$$De = Re \sqrt{\frac{r}{R_c}}$$

where Re is the Reynolds number.

Mathematical Model

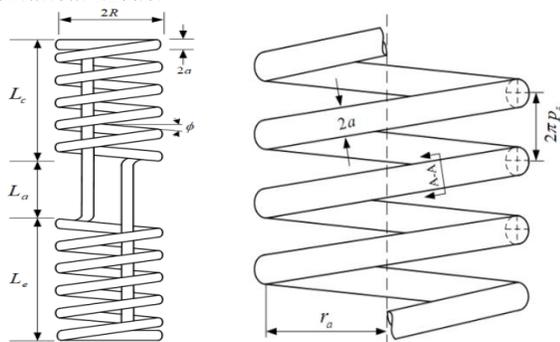


Figure 1: Schematic diagram of helical oscillating heat pipe

Same as circular pulsating heat pipe HOHP consists of three parts: the evaporator section (L_e), the adiabatic section (L_a) and the condenser section (L_c), with the coil radius r_a , the pitch p_s , the radius a (defined by the increase in elevation per revolution of coils $hg = 2 \pi p_s$), the curvature ratio κ and the torsion τ , which can be calculated using

$$k = \frac{r_a^2}{r_a^2 + p_s^2}$$

$$\tau = \frac{p_s^2}{r_a^2 + p_s^2}$$

Figure shows an orthogonal helical coordinate system. The basic governing equations for helical tubes can be represented in an orthogonal helical coordinate system, as suggested by Germano (1982). An orthogonal helical coordinate system can be introduced with respect to a master Cartesian coordinate system (x, y, z), by using the helical coordinates s for the axial direction, r for the radial direction and θ for the circumferential direction.

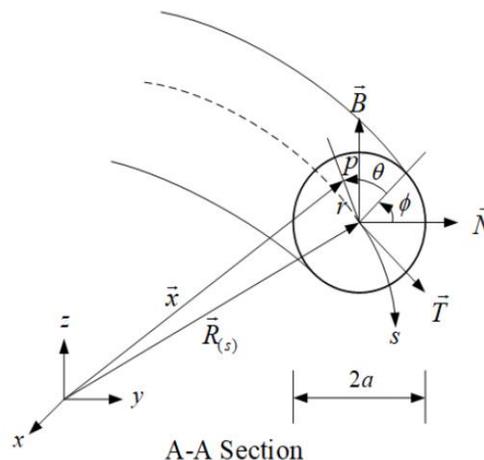


Figure 2: Cylindrical coordinate system for heat pipe

The vector in the orthogonal coordinate system of a HOHP is $\vec{R}(s)$, as calculated by

$$\vec{R}(s) = a \cos(s) \vec{i} + a \sin(s) \vec{j} + b(s) \vec{k}$$

The position of any given point p inside the helical pipe can be described by the vector $\vec{X}p$, which can be calculated as

$$\vec{X} = \vec{R}_s - r \sin(\theta - \tau s) \vec{N}_s + r \cos(\theta - \tau s) \vec{B}_s$$

Where $\vec{T}, \vec{N}, \vec{B}$ are the tangential, normal and bi-normal directions to the generic curve of the pipe axis $\vec{R}(s)$ at the point of consideration, respectively. The metric of the orthogonal helical coordinate system is given by:

$$d\vec{x} \cdot d\vec{x} = (1 + \kappa r \sin(\theta - \tau s))^2 ds^2 + dr^2 + r^2 d\theta^2$$

where dr, ds and $d\theta$ are the infinitesimal increments in the radial, axial and circumferential directions, respectively. With this metric, one obtains the scale factor h_s as given by

$$h_s = 1 + \kappa r \sin(\theta - \tau s)$$

Governing Equations

Governing equations for the calculated of the HOHP are the governing equation at the pipe wall and the vapor core. In addition, there is the governing equation for calculation of the heat transfer. All of which can be describe as in the following.

Heat Conduction of the Pipe Wall

Under normal operation, the heat applied to the evaporator section by an external source is conducted through the pipe wall. The three-dimensional transient condition heat conduction equation that describes the temperature in the heat pipe wall from conservation of energy is

$$\rho_w C_{p,w} \frac{\delta T_w}{\delta t} = \frac{k_w}{r} \frac{\delta}{\delta r} \left(r \frac{\delta T_w}{\delta r} \right) + \frac{k_w}{r} \left(\frac{\delta^2 T_w}{\delta \theta^2} \right) + k_w \left(\frac{\delta^2 T_w}{\delta z^2} \right) + \dot{Q}_w$$

where $C_{p,w}$ is the effective heat capacity of the pipe wall, and k_w is the effective thermal conductivity of the pipe wall.

Vapor Core

The continuity, the momentum and the energy equations used in the calculation at the vapor core of the HOHP are given

Continuity Equation:

$$\frac{\partial rw}{\partial s} = 0$$

Momentum Equation

$$\frac{\partial w}{\partial t} + \frac{1}{h_s} \frac{\partial w}{\partial s} + \frac{k \sin(\theta - \tau s)}{h_s} w + \frac{k \cos(\theta - \tau s)}{h_s} w = - \frac{1}{h_s} \frac{\partial p}{\partial s} + \frac{1}{Re} \left[\frac{1}{h_s} \frac{\partial}{\partial s} \left(\frac{1}{r h_s} \frac{\partial rw}{\partial s} \right) \right]$$

Energy Equation

$$\frac{\partial T}{\partial t} + \frac{w}{h_s} \frac{\partial T}{\partial s} = \frac{1}{RePr} \left[\frac{1}{h_s} \frac{\partial}{\partial s} \left(\frac{r}{h_s} \frac{\partial T}{\partial s} \right) \right]$$

II. METHODOLOGY

Geometry

Since the simulation is expensive in terms of computation, the number of turns is taken as four. The geometry for the figure shown above is taken from the experimental work performed by Pachghare (2014) in order to valid and verify the results before proceeding with the helical structure because much researches has not been done in the area of helical oscillating heat pipe.

- A Water
- B Steam
- C Evaporator
- D Adiabatic water
- E Adiabatic steam
- F Condenser

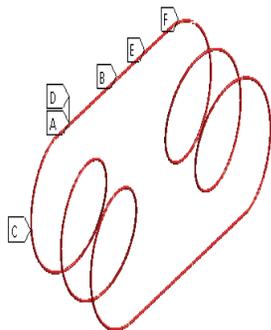


Figure 3: Schematic of model for simulation

Table 2: Design parameters for helical heat pipe

Length of Evaporator	100mm
Length of Condenser	100mm
Number of turns	2.5
Diameter of coil	80mm
Diameter of pipe	2mm
Pitch	40mm

Physics Setup

On the basis of the experiment performed by Pachghare, et al. (2014) following simulating criteria is selected.

Table 3: Physics setup

SN	Domain/Boundary/Physics	Type
1	Fluid Model	Volume of Fluid (Lee Model)
2	Time step for Transient Analysis	0.0003
3	Materials: Water	Liquid & Vapor
4	Evaporation and Condensation Model	For phase change
5	Saturation Temp & Pressure	308K & 4kPa
6	Convergence Criteria	10 ⁻³
7	Solver: Solution Method	SIMPLE
8	Wall material	Copper

Boundary Conditions

First simulation is tested with 80W heat input on the evaporator in which the heat fluxes are provided directly to the walls of evaporator. This is the Neumann boundary condition because it involves the value obtained from the derivate of thermal energy per unit area with respect to time. In some of the experiments performed by researchers (Yeboaha et. al (2018)), the walls of evaporator is heated by a hollow cylindrical pipe in which a hot liquid flows.

The temperature in condenser should be less than the saturation temperature; otherwise, the phase change won't occur there. Thus condenser of the heat pipe is maintained at 302 degree Kelvin.

Zero heat flux is assigned to the adiabatic section because Fluent recognizes the insulation area with this value.

Copper, whose thickness is taken as 0.5mm, is selected as wall material as it is preferred as solid material for heat pipes due to its high thermal conductivity.

Solution Controls

The Courant number is adaptive and was not fixed for the simulation to run. The under relaxation factors for momentum and pressure were take as 0.3 each because solution converged soon as this value was decreased from the default value of 0.7.

Initial Conditions

The saturation temperature is taken to be 35 degree Celsius for simulation. All velocities are taken zero because at t=0, both fluids (air and water) are stationary enclosed in closed loop. As the thermal energy is transferred slowly, kinetic energy of the fluid particles increase and velocity is observed. Vapor particles are expected to move from evaporator to condenser, which is kept at a constant temperature of 29 °C, and condensate at 35 degree Celsius (308 K).

III. RESULTS AND DISCUSSIONS

Tuning of coefficients in Lee Model

The coefficients in the Lee Model has to be tuned which is the case of hit and trial to verify the case as similar to heat pipe. Therefore, before proceeding to the simulation in heat pipe, a simple case was studied in a cube which has the following case.

In a closed cubical cylinder 25% of the volume is occupied by saturated water of mass (ml) 1.4kg and rest by saturated steam at 200 deg Celsius. When “Q” amount of heat to be supplied to water the water should evaporate completely within 12.88sec.

Calculation of Q and Tuning of Lee Coefficients

Using iterative method for Lee coefficient we obtain [Source of Lee] value of 0.0008. The Lee coefficient signifies that the mass of water of 1.4 kg at saturation temperature of 200C completely converts to vapor phase with addition of Q = 3.9 MW in 14.9 sec.

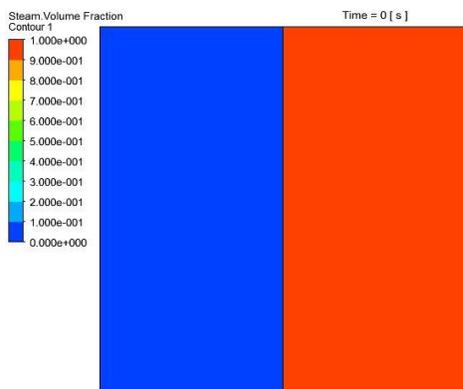


Figure 4: Steam volume fraction at t=0

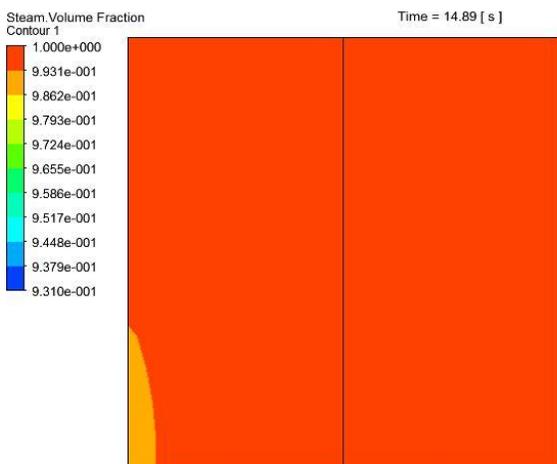


Figure 5: Steam volume fraction at t=14.89

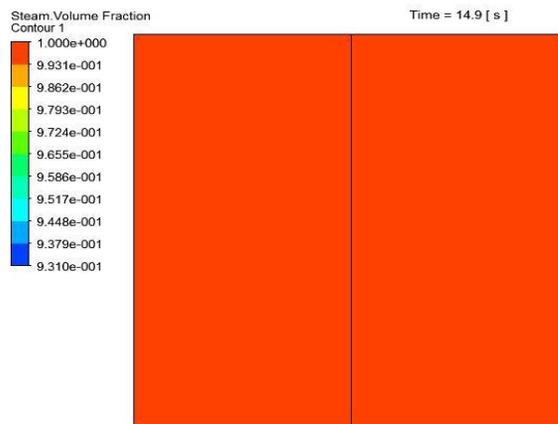


Figure 6: Steam Volume fraction at t=14.9

Different to the circular structure, the thermal resistance curve in helical first increases to the certain amount of heat input and decreases. It is not the area of interest beyond the value greater than 90W because heat pipe is not suitable to transport this amount of heat from evaporator to condenser that can result in dry out condition.

Effect of Transient Evaporator Heat Flux in Helical Pipe

In helical OHP, at a certain point in evaporator the value of different parameters such as thermal resistance, heat flux, steam volume fraction, difference in thermal potentials are observed. At t= 0.5, thermal resistance is found to be 0.54. As the rate of heat input from the evaporator increase, thermal resistance decreases until a constant value of 0.3 K/W is reached. At this stage the heat flux in the evaporator is 20252 W/m². After increasing the heat flux to a amount of 25390 W/m², the thermal resistance value increases drastically to 0.7 K/W. The increase in thermal resistance in that spot is due to the formation of vapor near to that region which pushes the liquid slug. The thermal energy in vapor phase is more than in liquid phase which heats up the wall of the evaporator. Hence, at this time, the circulation of the fluid commences resulting in the flow of liquid and vapor periodically in a particular region. The time of start of the circulation was found to be 2.52s from the simulation result. As the time elapses, the thermal resistance value at the particular region in evaporator shows the same behavior as in previous time period.

Table 4: Different thermal parameters value obtained from simulation

t(s)	Q _{in} (Watt)	T _e (K)	T _c (K)	R
0.5	12.2401452	308.6	302	0.53920929
1.02	24.4802904	310.067	302	0.329530405
1.5	36.000357	312	302	0.277775023

2.01	48.2405022	316.969	302	0.310299423
2.52	60.4806474	344.79	302	0.70749904
3	72.000714	330.433	302	0.394898862
3.5	84.2408592	338.336	302	0.431334632
4	96.4810044	350.411	302	0.501767164
4.2543	102.10443	357.972	302	0.548183855

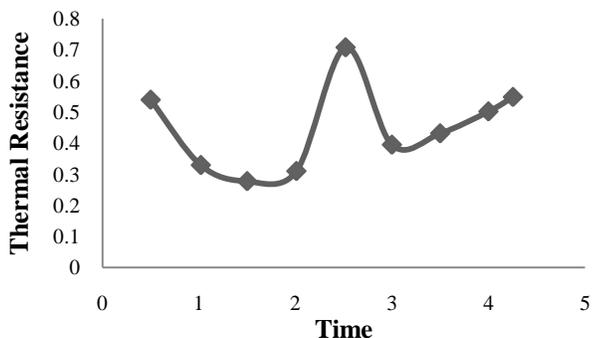


Figure 7: Thermal Resistance vs. Time

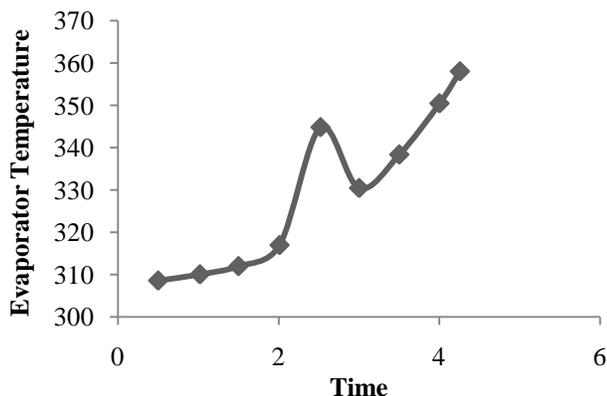


Figure 8: Evaporator Temperature vs. Time

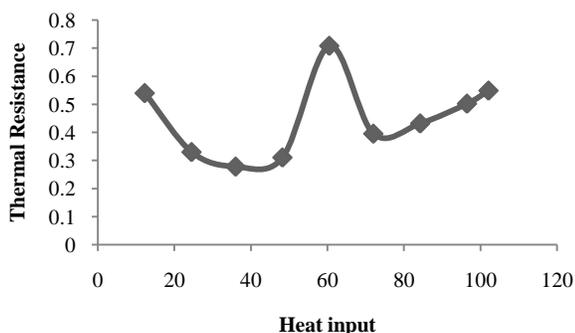


Figure 9: Thermal Resistance vs. Heat Input

The temperature of evaporator at a particular region increases linearly at first because the wall heats up taking the sensible heat flux at first and increases drastically due to the energy of the vapor phase formed by taking evaporative heat.

Start Up Performance of Helical Heat Pipe

After heat is supplied to the evaporator, the vapor generation starts as the saturation temperature is reached. Due to the uniform heat flux in the evaporator, the vapor generation starts everywhere downside and upside of the curve. Due to gravity, the vapor bubbles from the upper part descend to the lower region where it coalesce with the bubbles formed in the lower region resulting in the formation of liquid and vapor slug. As this process continues, the vapor pressure in the evaporator region rises till it's sufficient to drive the liquid and slug of vapor from evaporator to the condenser region. The amount of time taken for one single liquid slug to move from the evaporator to condenser region and back to evaporator is called the start up time.

Annular Flow

Annular flows in pipes are the flows that mostly occur in bends. The liquid and vapor slugs change the shape of the flow from circular to the angular due to the centrifugal force and secondary flows.

Slug Flow

Slugs of liquid and vapor are formed when the large amount of vapors combine which are formed at different intervals of time. When sufficient amount of heat is supplied to the evaporator, the change in the thermal energy to the vapor pressure energy starts the circulation of these slugs against the buoyant and surface tension forces.

Bubble Flow

The formation of bubble is seen at the initial stage of heating of the heat pipe. Some amount of bubbles strike the walls losing its kinetic energy; some starts growing in diameter due to coalesce of the vapors.

Stratified Flow

The flow in which the bubbles and slugs in conjunction is known as stratified flow.

The enhancement in heat transfer in helical pipe is due to the complex flow pattern existing inside the pipe. The helix angle and the pitch of the coil results in the torsion of the fluid and the curvature of the coil determines the centrifugal force. The centrifugal force develops a secondary flow inside the helical tube. The curvature effect makes the fluid in the outer side of the pipe to move faster than that present inside, which gives a difference in velocity setting up a secondary flow which changes correspondingly with the Dean number of the flow.

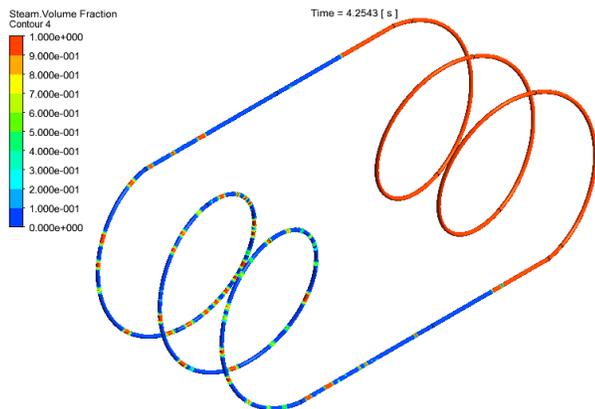


Figure 10: Steam volume fraction at $t=4.25$ for transient heat input with different bubbles and slugs shown

IV. CONCLUSION

From the simulation of the cubical cylinder, the evaporation frequency to carry out this simulation was found to be 0.0008 which is the case of hit and trial to validate the result. This frequency, which is also termed as coefficient, is used in the entire simulation to see evaporation and circulation process in the heat pipe.

Thermal resistance of the heat pipe decreases from the value gradually as heat input in the evaporator is increased whereas, in helical pipe thermal resistance value decreases at first and increases again decreases; thermal resistance value in helical pipe is oscillating in nature for different heat inputs.

Thermal resistance in helical heat pipe is less than in circular pipe which is found from constant heat input flux for both the geometry. From the simulation, helical structure is found as more effective heat transfer device compared to circular.

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