Design and Thermodynamic Analysis of Single-Loop Thermosyphon

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Abstract. A bibliographical review of the literature on heat and mass transfer in gravity-aided closed Loop Two-Phase Thermosyphon (CLTPT). To prevent the dissolution of the liquid layer, it is advised to use a thermosyphon that is optimally filled with a modest amount of working fluid. The main task is to develop a CFD model to examine temperature variations and stage changes in thermosyphons. As a result, thermodynamic analysis is carried out for void factors between 40% and 80% and also calculates pressure drop, superficial velocity, heat-transfer coefficient, and heat transfer under various filling conditions.

Keywords: Closed Loop Two-Phase Thermosyphon (CLTPT) · void factor · Lockhart Martinelli correlation

1 Introduction

1.1 Thermosyphon

Thermosyphon is a passive heat exchanger that circulates a fluid without the use of a mechanical pump and is based on natural convection. For the circulation of liquids and flammable gases in heating and cooling systems including heat pumps, water heaters, boilers, and furnaces, thermosyphon is utilized. Thermosyphon also takes place across air temperature gradients, such as those found in a solar chimney or a chimney used for a wood fire.

1.2 Working of Thermosyphon

An evaporator, condenser, riser, and downcomer make up this system (see Fig. 1). Fluid is converted from a vapor to a liquid condition using a condenser. After rejecting heat, liquid fluid moves gravitationally to the input of the evaporator. This procedure was repeated until the system's requisite temperature was maintained. In these applications the circle thermosyphons normally work at low tension and temperatures contain into reach 20–120 °C.

The working rule of those gadgets as for the notable single cylinder thermosyphon manages the division of the pathways of the recharging fluid from that of the fume getting away from the vanishing zone. In such a manner a significant increment of

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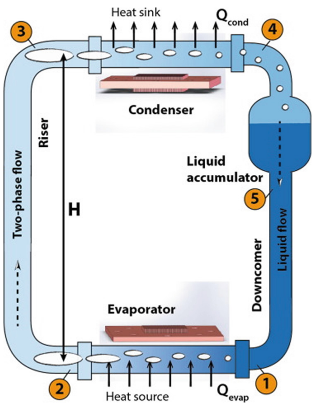

Fig. 1. Two-phase loop thermosyphon

intensity transport concerning the traditional two stages shut thermosyphons is gotten and the old-style employable constraints like flooding or entrainment restrictions are survived. These impacts are more amazing as the size of the gadgets diminishes.

1.3 Literature Study

A thermosyphon's warm execution has been the subject of numerous tests, many of which have examined the various factors that influence its display. The accompanying is in line with these. A few gleam exchange packages benefit from using 2-degree thermosyphons (TPTs).

According to Josh Charles et al., [1] examined heat transfer coefficients that were estimated across the single and two-phase sections of the loop to better forecast heat transfer within the evaporator and condenser and to estimate thermal losses from the thermosyphon. While Chen's modified Dittus-Boelter equation is used to determine the two-phase heat transfer coefficient in the non-boiling, two-phase zone, the Kandlikar correlation is utilized to forecast the saturated flow boiling heat transfer coefficient inside the evaporator. Heat loss from each portion of the condenser is calculated using the sum of thermal resistance method, and the Shah correlation is used to approximate the condensing heat transfer coefficient in the loop thermosyphon sections of the condenser. The evaporator, two-phase riser, and condenser all employ two-phase pressure drops correlation like Lockhart and Martinelli's because the accurate calculation of the loop pressure drops is essential for predicting LTS performance. Using sections of single-phase liquid or vapor, Darcy's equation is used to determine the frictional pressure decrease. Alessandro Franco et al. [2] focused on assessing the results of recent experimental studies on the heat transmission of two-phase flow in compact closed-loop two-phase thermosyphons (CLTPT). Despite the large number of experimental studies that have been carried out, few authors have examined the relationship between temperature and fluid dynamic performances. The most important discovery is the presence of a maximum mass flow rate as a function of heat input however this maximum is dependent on several experimental variables. Inconsistencies in data appear to be caused by measurement errors and experimental conditions, such as the presence of air, and can result in large changes in mass flow rate for the same heat input when small deviations from the design operating state exist. The practical use of CLTPT in systems with changeable heat input conditions, such as solar collectors or heat transformers, is hampered by the lack of general results. The report concludes that further thorough research is needed to learn enough about the transport mechanism in CLTPT. In conclusion, there is still much to learn about the principles governing fluid flow and heat transmission in small-scale closed-loop two-phase thermosyphons. Kyung Mo et al. [3] examined how annular and concentric thermosyphons with individual lengths of 215 mm and internal and outer distances of 22 mm and 25.4 mm performed well at high fill levels. The execution of both thermosyphons was centered on considering changes of a significant distance at high fill volume. Since their findings, concentric thermosyphons have performed significantly better than annular thermosyphon due to the increased fill extent. This is because a decrease in the pass-sectional area for fume flow causes an increase in the shear at the fume fluid connection component, which in turn leads to this development. Hasna Louahlia et al. [4] studied circumnavigated thermosyphon's evaporating and improved heat skip coefficients. The evaluation wrapped up by way of changing the strain load throughout the starter. The result indicates that the warm resistance of the condenser and evaporator decreases as the electricity load increases. Their findings are the same. This demonstrates that an increase in warmth load is an important factor in the path of thermosyphon execution, despite the use of specific strategies. Thanaphol et al. [5] focused on the results of including R134a as the working fluid in a versatile hose in a thermosyphon at the adiabatic fragment. The initial coordination was achieved by varying the flexible hose component under extreme pressure. In light of their investigation, it was discovered that bowing the road had little effect on the stage thermosyphon circulation version and the line strain drop. The temperature that goes away is better the higher the pressure drops. This well-known demonstrates that even though a plan change can alter thermosyphon execution with a new separator at the adiabatic fragment Khalid and Witwit [6] examined the onset of general-degree thermosyphon. The thermosyphon was constructed using a near-copper tube with a length of 1,000 mm and internal and exterior breadths of 26 mm and 32 mm, respectively. The adiabatic component time frame differs at 100 mm, 300 mm, and 700 mm. Evaluation and a typical segment thermosyphon emerged from the display of the altered-degree thermosyphons. The results showed that using an adiabatic separator improved the two-diploma thermosyphon performance across all lengths and orientations. That the emergence of the thermosyphon is influenced by converting the adiabatic problem via and large. As a result, this demonstrates that thermosyphon development and mathematics anticipate a crucial component for its display. Anjankar and Yarasu [7] focused on the exploratory evaluation results of condenser length on the presentation of a two-stage vertical close thermosyphon with unique flow speeds of 6, 8, and 10 liters per hour to the condenser and power to the evaporator, respectively. At a temperature input of 500 W and a stream rate of 0.0027 kg/s and a condenser length of 450 mm, the final result indicates that the warm display of thermosyphon is typically increased. To achieve better heat transfer, it is most likely assumed that the length of the condenser should be 1.5 times that of the evaporator. It showed that the demonstration of the thermosyphon is been invigorated by utilizing fluctuating perspectives. Initial research on the warm execution of anodized thermosyphon (TPCT) phases with lengths of 350 mm, 14.9 mm inside, and 16.5 mm outside has been completed by Brusly Solomon et al. [8]. The evaluation was carried out to determine the superior and relaxed resistance as well as the heat-consistent brilliance of the anodized surface prepared on the inside mass of the TPCT. The pressure circulates coefficient of anodized TPCTs is greater than that of non-anodized TPCTs at all electricity enter values, as measured by varying the fluid's strength input from 50 to 250 W. The results indicate that the anodized TPCTs are responsible for CH3)2CO. Additionally, the reduced heat resistance of the evaporator made it clear that the anodized TPCT's top and relaxed impedance is more fragile than that of non-anodized TPCT. The pleasant and comfortable constancy examination demonstrated the TPCT's feasibility of employing penetrable security. By employing CH3)2CO as the working liquid, Renjith Singh, et al [9] focused on the effects of anodization at the depth of the flow on degree thermosyphon. To ensure uniform protection, the anodization was carried out with the aid of a permeable defensive and a hole of three millimeters was stored up between a portion of the anode and cathode. The effects that were given were then compared to a barrel-formed thermosyphon to see how different they were. The thermosyphon's performance is largely influenced by fill percentage and tendency factor, and anodized thermosyphon has a 20 percent lower warm resistance than non-anodized thermosyphon. This result and conclusion that the anodized TPCT has a lower warm resistance than the non-anodized TPCT. When compared to thermosyphons that have not been anodized, the greatest increase in heat skip accumulation is 27%. When compared to spherical and hollow thermosiphons, the level of thermosyphon was generally regarded as performing better. The heat skip coefficient of the stage thermosyphon at the evaporator and condenser was higher than that of the tube-formed thermosyphon at the warmth transition, with values of 69% and 56%, respectively. By using R410A refrigerant as a walking liquid, Ong et al. [10] investigated the excellent and comfortable resistance of a thermosyphon at various fill proportions and tendency angles (30°, 60°, and 90°). The thermosyphon is made from a 930-millimeter-long copper tube with internal and external dimensions of 9.5 and 12.7 millimeters, respectively. According to their analysis, the temperature obstruction of thermosyphon indicates that fill percentage and tendency aspect did not affect the beautiful and comfortable display. However, it became clear that the trending issue of 60° performed better than the special tendency points of 30° and 90°. In addition, it was discovered that the thermosyphon wall temperature rises in tandem with the enter strength, and it is common knowledge that the dry-out effect may have occurred as a result of high inclinations, low fill proportions, and excessive energy inputs. It was hypothesized that water-filled thermosyphons performed better at lower fill proportions while R410A-stuffed thermosyphons performed better in the upward function at all fill proportions.

From the above study, it is understood that different boundaries that influence the intensity move rate are cooling liquid properties, filling proportion, and so on. The examinations were completed on the water as the working medium. We understood the intensity move rate by involving the above techniques for various void components might assume a crucial part for planning thermosyphon. To improve a CFD model that

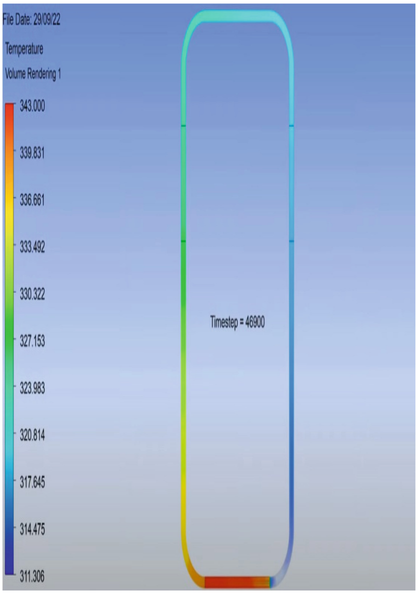

Fig. 2. Temperature Profile at 46900 steps

allows for the analysis of temperature variations and stage changes in thermosyphons. As a result, thermodynamic analysis is carried out for void factors between 40% and 80%. The calculation is done for pressure drop, superficial velocity, heat transfer, and heat transfer under various filling conditions.

The primary objectives are to find the pressure effect in closed single loop thermosyphon from 40% to 80% void factor, to find the heat transfer effect in the riser, downcomer, evaporator, and condenser in closed single loop thermosyphon from 40% to 80% void factor and to investigate on the Gas velocity and liquid velocity effect in closed single loop thermosyphon from 40% to 80% void factor.

2 Modeling and CFD Analysis of Thermosyphon

The temperature profile of a closed single loop thermosyphon for a diameter of 10mm and length of 800mm is 343K as the input evaporator temperature and condensing temperature where gas changes gas to liquid is 311K (see Figs. 2 and 3).

3 Theoretical Calculation

The bond number can be used, specifically in pipes, as a connection indicator between the pipe's diameter and the two-phase flow pattern. This is defined by law as the square of the relationship between the capillary length (l) and the characteristic dimension (pipe dia (d)) of a pipe, which is proportional to bubble departure diameter:

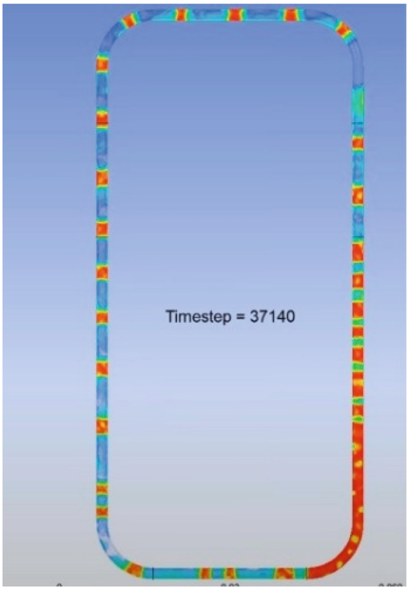

Fig. 3. Water and Gas volume at 37140 steps

Capillary Length

$$L = \frac{\sigma}{g(\rho_L - \rho_V)} \tag{1}$$

$$D_{\text{critical minimum}} = 2 * L = 5\text{mm}$$

$$D_{\text{critical maximum}} = 19 * L = 47.5\text{mm}$$

Filling Ratio or Void Factor

$$(\in) = \frac{1 - \frac{H}{L}}{\frac{\Delta\rho}{\rho_L}} \tag{2}$$

Lockhart and Martinelli suggest that 2-phase frictional pressure drop per unit length can be predicted in an evaporator, 2-phase riser, and condenser where empirically obtained, two-phase pressure drop correlations are most appropriate.

Lockhart Martinelli Correlation

$$\left(\frac{dp}{dx}\right)tp = \text{Ø}^2{}_{\text{L}}\left(\frac{dp}{dx}\right)\text{sl} = \text{Ø}^2{}_{\text{G}}\left(\frac{dp}{dx}\right)\text{sg} \tag{3}$$

The Lockhart and Martinelli parameter (Ψ) two-phase frictional pressure-drop multiplier and a C value

$$\text{Ø}^2{}_{\text{L}} = 1 + \frac{C}{\Psi} + \frac{1}{\Psi^2}\text{Ø}^2{}_{\text{G}} = 1 + \text{C}\Psi + \Psi^2 \tag{4}$$

$$\Psi 2 = \frac{\left(\frac{dp}{dx}\right)sl}{(\frac{dp}{dx})sg} \tag{5}$$

The single-phase pressure drops are calculated using the single-phase Darcy pressure drop equation for liquid or vapor flow

$$\left(\frac{dp}{dx}\right)Sg = \frac{\left(2f\,\rho_g v^2\right)\mathrm{air}}{\mathrm{D}} \tag{6}$$

$$\left(\frac{dp}{dx}\right)\mathrm{Sl} = \frac{\left(2f\,\rho_l v^2\right)\mathrm{water}}{\mathrm{D}} \tag{7}$$

Kandilkar Correlation (flow boiling heat transfer for evaporator)-Heat transfer coefficients were estimated across the 1 and 2-phase sections of the loop to estimate thermal loss from a 2-phase single loop thermosyphon and better forecast heat transfer within the condenser and evaporator. A correlation created by Kandlikar was utilized to forecast the saturated flow boiling heat transfer inside the evaporator.

$$\mathrm{h_{tp}} = \mathrm{C_1 C_0 (25Fr\ L_0)^{C5} h_{Liquid}} + \mathrm{C_3 B_0^{C4} h_{liquid} F_{fL}} \tag{8}$$

Dittus-Boelter Correlation (down corner)-The 2-phase heat transfer coefficient is calculated using Chen's modified Dittus-Boelter equation in the non-boiling, 2-phase zone.

$$\mathrm{h_{2\phi}} = 0.023(\mathrm{Rem})^{0.8}\left(\mathrm{Pr_{liq}}\right)^{0.4}\left(\frac{\mathrm{Kliq}}{\mathrm{d}}\right) \tag{9}$$

Shah Correlation (condenser)-Shah correlation to estimate condensing heat transfer coefficient for loop thermosyphon.

$$\mathrm{h_{2\phi}} = \mathrm{h_{liquid}}[(1-\mathrm{x})0.8 + (\frac{3.8\mathrm{x}^{0.76}(1-\mathrm{x})^{0.04}}{(\mathrm{Prliq}^{0.38})})] \tag{10}$$

Chen Correlation (riser)-The Chen correlation of saturated flow boiling heat transfer for flow boiling in the riser.

$$\mathrm{h_{forster\text{-}zuber}} = 0.00122(\Delta\mathrm{Te})^{0.24}(\Delta\mathrm{Pe})^{0.75}[\frac{(\mathrm{Kl}^{0.79} * \mathrm{Sl}^{0.49} * \mathrm{Cpl}^{0.45})}{(\sigma^{0.5} * \mu\mathrm{l}^{0.29} * \mathrm{hfy}^{0.24} * \rho\mathrm{v}^{0.24})}] \tag{11}$$

4 Results and Discussion

Investigated are overfilled and underfilled situations for which the working medium with the best case is present. The lowest thermal resistance and fastest response time are found in the optimally filled loop thermosyphon system; nevertheless, a minor increase in input power will result in condensate film disintegration. In comparison to an ideal condition, an overfilled thermosyphon exhibits increased heat resistance and slower responsiveness. It has a major effect on how much fluid is filled in a closed-loop thermosyphon system. In the system, the less working fluid is filled at a 50% filling ratio than at a 60% filling ratio.

In Fig. 4, it is evident that the pressure effect in the loop causes a decrease in pressure as the void factor rises from 40% to 80%. A smaller amount of water evaporates much faster than a larger amount.

Figure 5, shows that as the void factor increases by 40% to 80% heat transfer coefficient in the downcomer decrease. The closed-loop thermosyphon system's working fluid fill level is significantly affected the heat transfer coefficient.

Figures 6 and 7, shows that as the void factor increases 40% to 80% heat transfer in the condenser and evaporator increases. The loop thermosyphon system's total performance also depends on the heat transfer.

Figure 8, the gas velocity and homogeneous velocity in the loop fall when the void factor increases from 40% to 80%. It can be seen that water velocity in the loop when the void factor increases from 40% to 80%, water velocity remains constant as the void factor increases. The closed-loop thermosyphon system's working fluid fill level significantly affects the gas, water, and homogeneous velocity.

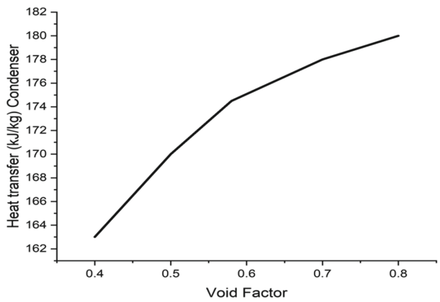

Fig. 4. Void factor vs Pressure

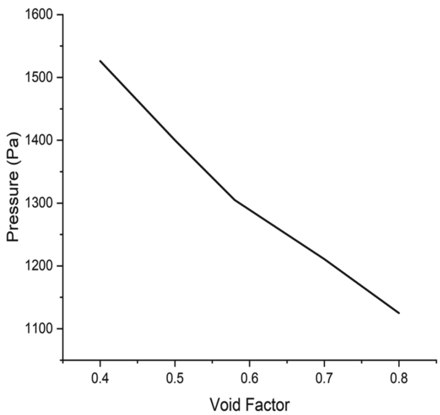

Fig. 5. Void factor vs Heat transfer coefficient

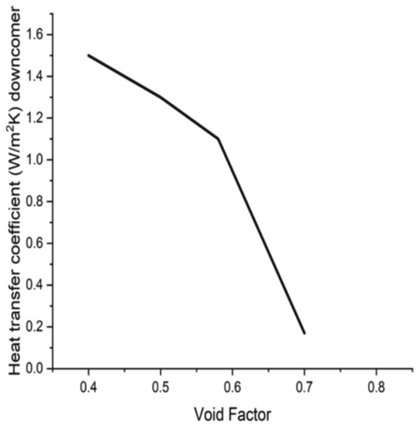

Fig. 6. Void factor vs Gas velocity

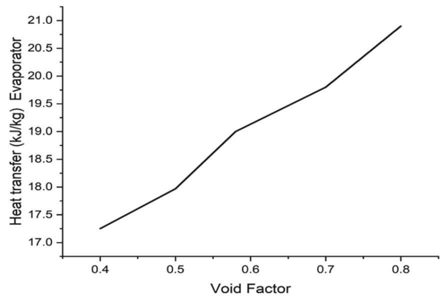

Fig. 7. Void factor vs Liquid velocity

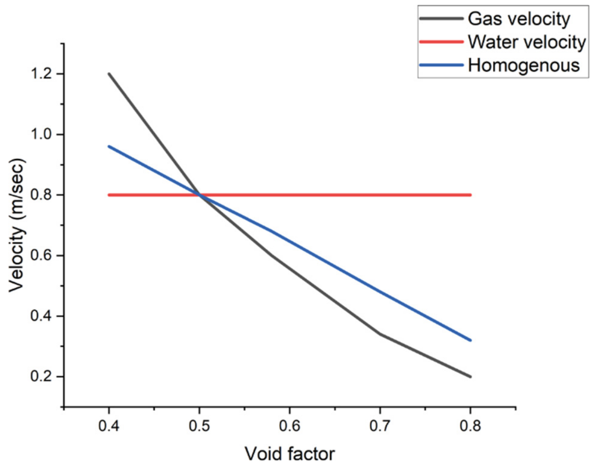

Fig. 8. Void factor vs Velocity

5 Conclusion

1. The pressure effect in closed single loop thermosyphon from 40% to 80% void factor gives that, drop in the pressure as void factor increases.
2. It can be seen that the velocity effect in the loop from 40% to 80% void factor gives, no change in the liquid velocity as the void factor increases.
3. The void factor increases by 40% to 80% heat transfer coefficient in the downcomer decrease and the condenser increases.
4. Through this paper, a void factor between 50% to 60%is prescribe for single loop thermosyphon, and the diameter of the loop is 5mm to 47.5mm for water as a working liquid.
5. It can be seen that the velocity effect in the loop from 40% to 80% void factor gives that, decreases in the gas velocity as the void factor increases.
6. So the optimum filling ratio is 50% to 60% when water is working fluid.

6 Nomenclature

Subscripts	Description
v	vapor
tp	two-phase
2Ø	two-phase
Bo	Boiling number

Symbol	Description
ρ_L	densityofliquid
ρ_V	densityofVapour
$\in$	fillingratio
μ	absoluteviscosity
σ	surafacetension
H	volumetric filling ratio

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