

ENCIT-2018-0161 MODELLING AND ANALYSIS OF CLOSED LOOP TWO-PHASE THERMOSYPHON

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Abstract. *This study aims to model and analyze closed loop two-phase thermosyphons. The influence of the system parameters on the performance of thermosyphons such as working fluid, geometry, operating temperature, heat flux and filling ratio are analyzed by numerical simulation. The model comprises conditions of single and two-phase flow, with corresponding predictive methods for each case. It consists in a one-dimensional model with averaged properties and parameters for each section. The concept of thermosyphons applied for heat management system has potential for applications in a variety of devices, such as in power electronics, electrical machines and nuclear reactor cooling applications. The present study presents modelling and analysis for a two-phase close-loop thermosyphon operating with R245fa, by evaluating local parameters along the circuit, such as flow pattern, void fraction, pressure gradient and heat transfer coefficient for single-phase flow, and for convective boiling and condensation conditions, as well as global parameters as mass flow rate estimation.*

Keywords: *thermosyphon, natural circulation loop, two-phase flow, theoretical modelling, flow patterns*

1. INTRODUCTION

A closed loop thermosyphon is a heat transport device capable of transferring heat from a heat source to a separate heat sink over a relatively long distance, without the use of active control instrumentation and any mechanically moving parts such as pumps or compressors. Thermosyphons are proven to be very effective, low cost and reliable heat transport devices with very different engineering applications such as solar water heaters, emergency cooling systems in nuclear reactors, electronic devices cooling, gas turbine blades cooling, thermoelectric refrigeration systems, geothermal energy harvesting systems and energy storage systems (Franco and Filippeschi, 2012).

The main reasons for their wide use in many energy and industrial applications are they simple design, passive nature and high heat transport capabilities. Two-phase loop thermosyphons for power applications below 1 kW are mainly related to the development of environmentally friendly systems such as solar collectors, photovoltaic modules cooling systems, heat exchangers for geothermal sources exploitation, and thermal control of conventional and small scales devices (Franco and Filippeschi, 2012).

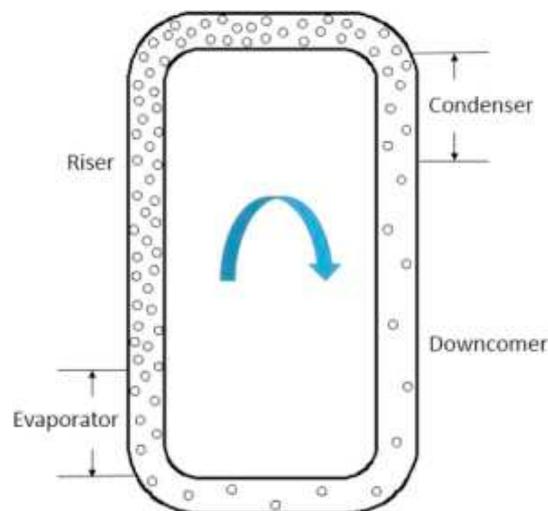


Figure 1. Closed loop two-phase thermosyphon scheme (Tong *et al.*, 2016).

The closed-loop two-phase thermosyphon can be visualized for simplicity as a long hollow pipe bent and the ends joined to form a continuous loop, usually oriented in a vertical plane and filled with working fluid. The hydrostatic pressure difference resulted from density gradient between the cold and hot sides drives the fluid flows (Dobson and Ruppertsberg, 2007).

The thermosyphons evaluated in this paper consist in evaporators and condensers connected by two adiabatic tubes, the down-comer and the riser. Heat is transferred to the working fluid in the evaporator, resulting in transition to liquid-vapor two-phase flow with consequent increment of mixture specific volume. Then, the fluid is directed to the condenser by flowing along the riser that is approximately adiabatic. In the condenser, where the heat is rejected to ambient, the fluid is completely condensed reducing the specific volume, and then the liquid is directed to the evaporator by flowing through the down-comer. It is desirable that a thermosyphon has reduced hydraulic resistance in order to favor flow circulation, which is imposed by relatively low pressure difference produced by natural convection (Dobson and Ruppertsberg, 2007).

Differently than the condition of forced circulation systems, the mass flow rate is not previously known, nor can be directly determined by the evaluation of pressure drop along the system and pump or compressor curve, but depends on the heat transfer processes, pressure drop along the circuit, possibility of abrupt bubble formation, among other aspects. Therefore, this study aims to model and simulate two-phase closed-loop thermosyphons.

2. BACKGROUND THEORY

This section presents a brief bibliographic review about the available studies undertaken to model theoretically thermal and hydraulic aspects of the working fluid in closed loop systems with natural circulation.

One of the first studies available in the open literature was presented by Welander (1966), who considered the pressure difference and buoyance force as the flow driven forces, and the frictional force as retarding one to the fluid. This author considered single-phase flow and made some assumptions such as one-dimensional heat transfer from the walls to the fluid, or vice-versa, by considering thermal resistances and that the frictional force was proportional to the instantaneous flow rate.

Later, Knaai and Zvirin (1990, 1993) showed that the single-phase loop theory could be extended for two-phase theory by some considerations as two-phase frictional multiplier for frictional pressure drop parcel, as well as relationships for void fractions, and predictive methods for single and two-phase heat transfer coefficients.

Dobson (1993) concluded that the finite difference discretization formulation scheme is able to capture the transient and the non-linear aspects of the loop for single-phase laminar flow. Vincent and Kok (1992) captured the transient performance of a two-phase closed loop thermosyphon and used the control volume approach. They concluded that this approach was a powerful tool to describe the performance of the thermosyphon.

Lee and Rhi (2000) considered methods for computer simulation of two-phase loop thermosyphons with heat transfer rates ranging from 60 to 100,000 W. They concluded that computer simulation alone does not give meaningful results unless they are accompanied with empirical correlations using experimental test results.

Alizadehdathel *et al.* (2010) performed a CFD modelling and analysis of heat transfer and fluid flow in a two-phase closed-loop thermosyphon. The simultaneous evaporation and condensation phenomena and vapor-liquid two-phase flow were considered in the model using the volume of fluid (VOF) technique. The predicted temperature profile was compared with experimental results and a good agreement was obtained.

3. CLOSED LOOP TWO-PHASE THERMOSYPHON THEORY

From the literature survey, it can be concluded that the following factors affects the thermal hydraulic performance of two-phase thermosyphons: thermophysical properties of working fluid, volumetric filling ratio, length of various sections (evaporator, adiabatic and condenser sections, and height difference between evaporator and condenser, also known as pump lift), heat flux, operating pressure and sub-cooling degree at evaporator inlet (Jadhav, 2016). Additionally, the heat exchangers are not necessarily in vertical position, hence, the channels inclination also play significant role on the performance of thermosyphons, such as indicated by Karthikeyan *et al.* (2010).

The parameters above listed affect the mass flow rate throughout the system and heat transfer coefficients in the evaporator and condenser, which are important for defining the system efficiency.

From the analysis of flow boiling, the heat transfer during convective flow is dominated by convective and nucleate boiling effects, with higher contribution of the second parcel due to the reduced flow velocity. Convective effects are related to the fluid flow close to the duct wall and those characterized by significant influence of flow velocity, which can be imposed by higher mass velocity or higher vapor fraction. Otherwise, nucleate boiling mechanisms present reduced influence of the flow velocity, where the phase change occurs due to bubble formation and growth in the channel wall (Kanizawa and Ribatski, 2016b).

The volumetric filling ratio, as defined by Franco and Filippeschi (2012), can be identified as the ratio between the driving head and the vertical distance between evaporator and condenser, corresponding to the submergence ratio. Agbo

and Unachukwu (2007) says that the thermosyphon driving head is evaluated from the difference between pressure head in the down-comer and riser.

Thereafter, it will be presented the predictive models and correlations for flow pattern, void fraction, pressure drop and heat transfer coefficient used to the numerical modelling of thermosyphons with phase change in closed loop.

3.1 Flow pattern

The predictive method for flow pattern during vertical upward flow proposed by Taitel *et al.* (1980) is adopted in this study. This method considers the transitions between slug, churn, bubble, dispersed bubble and annular flow patterns based on fluid properties, slip velocities, phases geometry, and channel geometry, with the void fraction as the main geometrical parameter. Taitel *et al.* (1980) presented the dominant mechanisms in the transitions, and then described the governing equations.

In the case of downward flow, the method proposed by the same research group and presented by Barnea *et al.* (1982) was adopted. Basically, three patterns are expected to occur concurrent vapor-liquid flow: annular, bubble and slug flows.

3.2 Void fraction and pressure drop

Woldesemayat and Ghajar (2007) presented an extensive review for void fraction predictive methods. They classified them as slip correlation, general correlations, drift flux models and correction factors for the homogeneous model for void fraction.

The homogeneous method is the simplest one but does not capture the effect of velocity and phases distribution profiles, and slip between the phases. Kanizawa and Ribatski (2016a) proposed a semi-empirical method based on the minimum kinetic energy method, with the form factor ratio correlated by using an extensive database from experimental studies available in the open literature, comprising hydraulic diameters between 0.5 to 89 mm and mass velocities between 52 to 2030 kg/m²s, for several working fluids. This method has shown reasonable agreement with experimental results for distinct operational conditions and was implemented in the present study for prediction of void fraction along the circuit.

During single-phase flow, the frictional pressure drop parcel was predicted assuming smooth tube for estimation of friction factor. In the case of two-phase flow, the pressure drop was evaluated by considering two-phase multiplier, with an approach similar to the proposal of Chisholm (1967), with the *C* parameter evaluated according to Zhang *et al.* (2010), appropriate for reduced pressure fluids.

3.3 Heat transfer coefficient

For single-phase flow in the evaporator and condenser sections, the correlation proposed by Dittus and Boelter (1930) was adopted due to its proofed validity.

The correlation proposed by Kandlikar (1990) for heat transfer coefficient during convective boiling was adopted in this study due to its wide range of validity. This author used about 10,000 data points for different fluids, including water, refrigerants and cryogenes for vertical and horizontal tubes. This correlation considers the contribution of nucleate boiling and convective boiling mechanisms, in a similar way to Chen (1966), however the estimated value corresponds only to the highest estimation for both mechanisms.

The correlation proposed by Katto (1984) was adopted for the estimation of vapor quality corresponding to dryout in the evaporator, which corresponds to the condition that successive increments of the fluid enthalpy results in heat transfer coefficient reduction. For dryout conditions, the method proposed by Saitoh *et al.* (2007) was adopted to estimate the heat transfer coefficient.

In the case of convective condensation, the method proposed by Bivens and Yokozeki (1994), who modified the method proposed by Shah (1979) in order to fit data with a wide range of mass flows, was used to estimate the heat transfer coefficient.

4. METHODOLOGY

This study consists in the mathematical and numerical modelling for analysis and simulation of thermal and hydrodynamic behavior of closed-loop two-phase thermosyphons, evaluating variations of channels geometry, and operational conditions for R245fa. The model is one-dimensional, consisting in discretization along the loop length, and is based on the energy and force balance. The energy balance gives the variation of fluid enthalpy along the thermosyphon loop, and depends mainly on the heat transfer process, as well as pressure drop. On the other hand, the pressure along the circuit is evaluated based on the force balance, given as pressure drop, which in turn is also affected by the heat transfer process. Therefore, both aspects are interrelated and must be solved simultaneously.

The circuit is discretized along the channels axial direction in short sections, and for each element the heat transfer coefficient, pressure drop, void fraction and flow pattern are evaluated. From these estimations, the properties in the inlet

of the following element can be evaluated. This process is solved for all elements in the circuit, attaining for distinct boundary conditions of each region, which includes heat flux, wall temperature, channel orientation, and so on, by adopting the corresponding appropriate predictive method.

The mass flow rate along the system is not previously known, which depends on the mixture specific volume gradient and pressure drop along the circuit. Therefore, an iterative method is needed to solve for the mass flow rate, which is performed by using a dynamic model of the system.

The false position method was selected for estimation of mass flow rate because of its simplicity and for having a reasonable convergence time. This numerical method is based on the bisection method, and uses appropriate weighting to estimate the corresponding increment value. The convergence criteria is based on the fluid pressure along the system, whereas the total pressure variation along the circuit should be null for steady-state condition.

Finally, it will be analyzed the effects that the system parameters have on the thermosyphon performance, which will be inferred based on simulation of some operational conditions, including system pressure, heat flux and overall circuit geometrical characteristics.

5. MODELLING

This section presents the physical and mathematical modelling approach. The model depends on the number of phases of the flow and the type of heat flux on the wall of each section to represent the energy and momentum transfer along the circuit.

5.1 Pressure drop

The pressure gradient (dP/dz) imposed to the flow can be related to the friction between the phases, friction between the fluid and the wall and variation of height or velocity of the phases. These contributions can be divided to frictional, gravitational and accelerational parcels. The accelerational and gravitational parcels are conservative, while the frictional is dissipative, hence cannot be recovered. Mathematically, these parcels are derived based on the Reynolds transport theorem for the conservation of mass and momentum quantity for an infinitesimal duct segment for each phase (Kanizawa and Ribatski, 2016b).

$$\frac{dP}{dz} = -\left(\frac{dP}{dz}\right)_{frictional} - \left(\frac{dP}{dz}\right)_{gravitational} - \left(\frac{dP}{dz}\right)_{accelerational} \quad (1)$$

The gravitational parcel for single-phase flow depends on the inclination of the duct (θ), gravitational acceleration (g) and the density of the working fluid (ρ), as follows:

$$\left(\frac{dP}{dz}\right)_{gravitational,sp} = \rho g \sin \theta \quad (2)$$

$$\left(\frac{dP}{dz}\right)_{gravitational,tp} = \{[\alpha\rho_v + (1 - \alpha)\rho_l]g \sin \theta\} \quad (3)$$

Where the subscript v refers to vapor phase, l to liquid phase, sp to single phase, and tp to two-phase flow.

Besides that, for two-phase flow the specific mass is function of the area averaged void fraction (α). For vertical conditions with reduced flow velocities or reduced friction, the gravitational parcel is dominant and can correspond to more than 90% of the total pressure drop. Therefore, the correct estimation of the void fraction is critical for evaluation of flow development along the channels (Kanizawa and Ribatski, 2016b).

The accelerational parcel in steady-state flow along a channel with uniform cross section and null mass transfer through the wall is related to variation of phases velocities, which in turns is related to phases specific volume. For two-phase flow, it is function of variation along the cross section, denoted by the void fraction α , which ultimately is related to velocity variation of the phases. The accelerational pressure gradient parcels are given as follows:

$$\left(\frac{dP}{dz}\right)_{accelerational,sp} = \frac{d}{dz} \left[\frac{G^2}{\rho} \right] \quad (4)$$

$$\left(\frac{dP}{dz}\right)_{accelerational,tp} = \frac{d}{dz} \left[G^2 \left(\frac{x^2}{\alpha\rho_v} + \frac{(1-x)^2}{(1-\alpha)\rho_l} \right) \right] \quad (5)$$

Where x is the thermodynamic vapor quality.

For single-phase flow, the frictional pressure drop parcel depends on mass velocity (G), fluid viscosity and geometrical characteristics, like diameter of the duct (d). It can be written as follows.

$$\left(\frac{dP}{dz}\right)_{frictional,sp} = 2f \frac{G^2}{\rho d} \quad (6)$$

Where the Darcy friction factor f for turbulent flow was evaluated based on Blasius (1913) correlation.

For two-phase flow, the frictional parcel depends on the shear stress between each phase with the duct wall, as well as the contact area between each phase. Due to its naturally complicated nature, the frictional pressure drop parcel during two-phase flow is usually evaluated based on correlations or models available in the open literature (Kanizawa and Ribatski, 2016b). The most common approach is based on the two-phase multipliers, firstly proposed by Lockhart and Martinelli (1949). After them, several authors suggested improvements and adjustments, such as Zhang *et al.* (2010), who proposed alternate correlations for the parameters C of Chisholm's (1967).

$$\left(\frac{dP}{dz}\right)_{frictional} = \Phi_l^2 \left(\frac{dP}{dz}\right)_l \quad (7)$$

Where the liquid single-phase pressure gradient $(dP/dz)_l$ and the two-phase multiplier (Φ_l^2) , are respectively calculated as follows.

$$\left(\frac{dP}{dz}\right)_l = 2f \frac{G^2(1-x)^2}{\rho_l d} \quad (8)$$

$$\Phi_l^2 = 1 + \frac{C}{\hat{\chi}} + \frac{1}{\hat{\chi}^2} \quad (9)$$

Where the term $\hat{\chi}$ is the Lockhart and Martinelli (1949) parameter given as follows:

$$\hat{\chi} = \sqrt{\left(\frac{dP}{dz}\right)_l / \left(\frac{dP}{dz}\right)_v} \quad (10)$$

5.2 Energy equation

The energy equation can be derived from the First Law of Thermodynamics for a control volume. Along the circuit, no mechanical work is performed by or to the fluid and the variations of kinetic and potential energy can be neglected when compared to the variation of enthalpy (i). Hence, by considering circular channel with internal diameter d , the gradient of enthalpy i along the channel is given as follows:

$$\frac{di}{dz} = \frac{4\phi}{Gd} \quad (11)$$

The heat flux term (ϕ) is modelled as uniform for the evaporator section, null for the adiabatic section and for the condenser section is function of heat transfer coefficient (h), wall temperature (T_w , modelled as uniform) and the mean temperature of the fluid to respect of cross sectional area (T_f) as follows.

$$\phi = h(T_w - T_f) \quad (12)$$

Where the heat transfer coefficient is evaluated according to the method described in section 3.3.

6. RESULTS

This section presents results for the default simulation that consists of selected parameters in order to remark the general performance of the modelled thermosyphon. Then, results for variation of some parameters that affect the performance of the thermosyphon.

6.1 Default simulation

For the default simulation the channel diameter d , vertical length L_V (riser and down-comer), horizontal length L_H , evaporator length L_E , condenser length L_C , riser inclination θ_R , down-comer inclination θ_D , evaporator heat flux ϕ_E , condenser wall temperature T_w , degree of subcooling in the evaporator inlet ΔT_{sub} , pressure of saturation at the beginning of saturated flow boiling in the evaporator P_{sat} and working fluid were established.

The simulation provides results (output parameters) that are mass velocity (G), heat transfer rate (Q), driving head (DH) and volumetric filling ratio (ξ). All those parameters are presented in Table 1, and it can be noticed from this table that the mass flux is considerably high if we consider that the fluid circulation along the circuit is natural.

Table 1. Parameters of the studied thermosyphon.

Input Parameters							
W. Fluid	R245fa	d [mm]	5	θ_R [°]	90	θ_D [°]	90
L_V [m]	4	L_H [m]	0.5	L_E [m]	2	L_C [m]	1
T_w [°C]	9	ΔT_{SUB} [°C]	2	P_{sat} [kPa]	177	ϕ_E [kW/m ²]	16.8
Output Parameters							
G [kg/m ² s]	140	Q [W]	528	DH [m]	1.5	ξ	0.38

For the default conditions, it is possible to evaluate the variation of pressure, vapor quality and void fraction along the thermosyphon. Figure 2 depicts the variations of pressure, vapor quality and enthalpy along the circuit (Figure 2.a, c and d, respectively), where $z = 0$ m corresponds to the evaporator inlet, and $z = 4.5$ m corresponds to the condenser inlet. Figure 2.b presents the variation of pressure with enthalpy, with the processes in clockwise direction. According to this figure, it can be noticed that the model properly captures the vaporization of the fluid, as well as complete condensation. Figure 2.e and f shows the heat transfer coefficient along the evaporator and condenser, respectively.

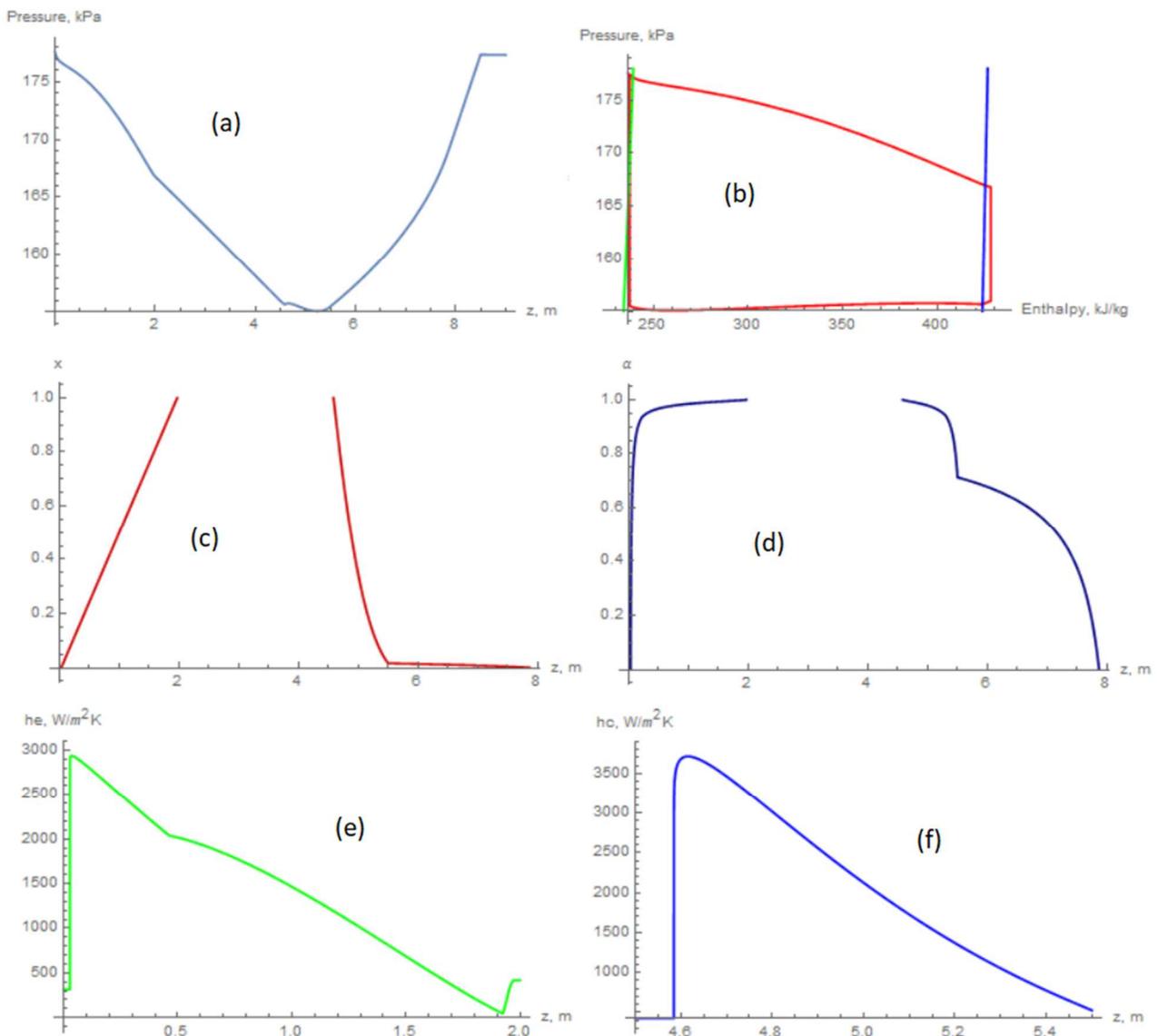


Figure 2. (a) Pressure; (b) Pressure vs. enthalpy; (c) Vapor quality; (d) Void fraction; (e) Heat transfer coefficient along the evaporator; (f) Heat transfer coefficient along the condenser.

Figure 3 depicts the variations of flow patterns for the evaporator and condenser. According to this figure, it can be noticed that annular flow pattern is predominant for both devices.

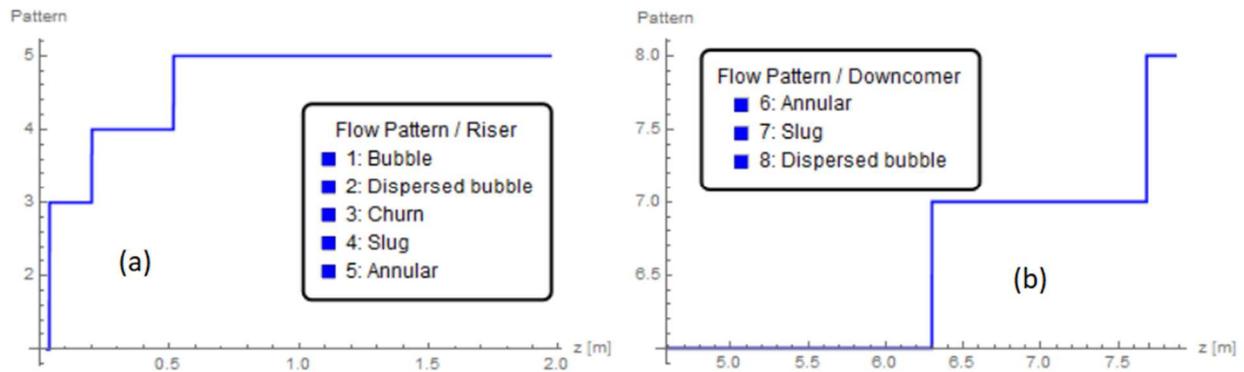


Figure 3. (a) Flow pattern / riser; (b) Flow pattern / down-comer.

6.2 Variation of parameters

Based on the previously described method, the thermosyphon parameters can be modified to evaluate the effect on the system response. Therefore, this section shows results for the variation of heat flux in the evaporator, internal diameter of the duct and inclination of the riser, and their effects on other parameters like mass velocity and driving head. Figure 4 depicts the variation of thermosyphon parameters, where all parameters are kept constant and equal to the description of Table 1 except the parameter in analysis.

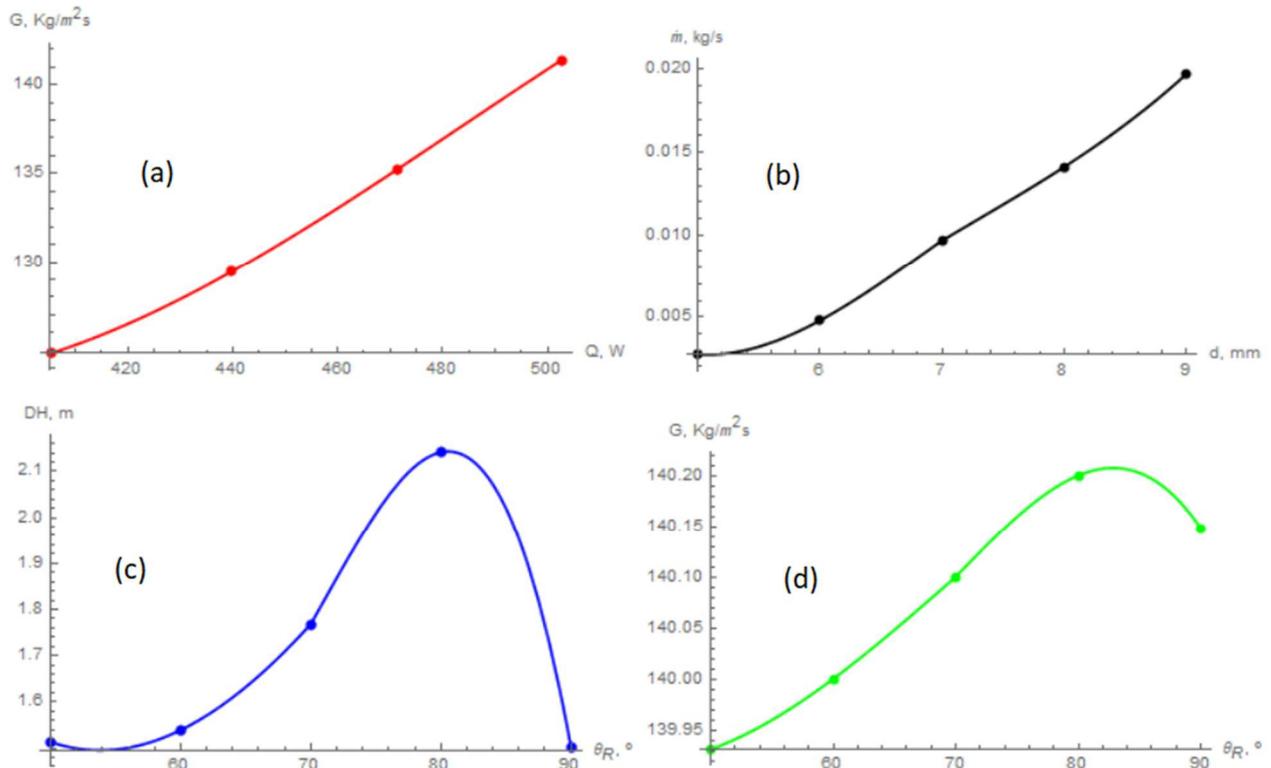


Figure 4. (a) Heat transfer rate vs. mass velocity; (b) Diameter vs. mass flow rate; (c) Inclination of riser vs. driving head; (d) Inclination of riser vs. mass flow rate

According to Figure 4, it can be concluded that the mass flow rate increases with heat flux and channel diameter due to higher vaporization rate and lower pressure drop, respectively. On the other hand, the driving head and mass flux

increases with heat exchangers inclination from 50° to approximately 80°, and then decreases due to increment of gravitational pressure drop parcel.

7. CONCLUSIONS

The following conclusions can be addressed for this study:

- A limited number of investigation of two-phase closed loop thermosyphon is available in the open literature, despite its common and potential use in several applications.
- The annular flow pattern is quoted to be predominant along both heat exchangers due to high vapor specific volume and flow velocity.
- The mass velocity increases and is approximately a linear function of heat flux imposed in the evaporator. Similarly, the mass velocity increases with channel diameter due to reduction of flow resistance.
- The mass velocity and driving head obtains a peak point at an optimum inclination of both heat exchangers close to 80°.

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