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DEVELOPMENT OF AN EXPERIMENTAL METHODOLOGY OF BOILING PHENOMENON IN THE EVAPORATOR OF A GLASS THERMOSYPHON

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Abstract. *The use of solar energy has become popular since recent years as an alternative clean source of energy, especially on home environments heating. Solar collectors assisted by thermosyphons can enhance the heat transfer to the water which should be heated. A thermosyphon is a vertical sealed tube filled with a working fluid and is characterized, from bottom to top, by an evaporator, an adiabatic tube and a condenser region. Depending on the heat load and the fill charge ratio applied to the thermosyphon, vapor bubbles nucleate on the evaporator region and may grow and coalesce, pushing up the liquid phase and thus forming a two-phase liquid-vapor slug flow. The slug flow pattern is characterized by the intermittent succession of an elongated bubble and a liquid slug and it was experimentally observed occurring inside the glass thermosyphon setup here presented for different heat loads with 100% of water fill charge ratio of evaporator volume. An experimental procedure is presented for estimating the heat transfer coefficient related to the boiling phenomenon that occurs in evaporator of a glass thermosyphon.*

Keywords: *Glass Thermosyphon, Experiment, Boiling Heat Transfer Coefficient.*

1. INTRODUCTION

The use of solar energy has become popular since recent years as an alternative clean source of energy, especially on home environments heating. Solar collectors assisted by thermosyphons can enhance the heat transfer to the water which should be heated. Thus, for the development of this kind of solar collectors, the heat transfer coefficient of the thermosyphon's intern flow is required.

A thermosyphon is a vertical sealed tube filled with a working fluid and is characterized, from bottom to top, by an evaporator, an adiabatic tube and a condenser region. So, the phase change phenomena which occur inside the evaporator and condenser sections are boiling and condensation, respectively. These phenomena have been studied and, as a result, correlations for two-phase flow heat transfer have been obtained for pool boiling (by Cooper, 1984) and for saturated flow boiling in vertical and horizontal tubes and annuli (by Gungor and Winterton, 1987; Kandlikar, 1990; Khodabandeh, *et al.*, 2002 and Khodabandeh, 2005).

According to Mantelli (2013), different heat transfer regimes (pool boiling, vapor flow and slug flow) can be found in the evaporator, depending on the heat load and the fill charge ratio. The thermal characteristics of thermosyphons under different fill charge ratios have been studied (by El-Genk and Saber, 1999; Park, *et al.*, 2002; Jiao, *et al.*, 2008 and Shabgard, *et al.*, 2014). However, experimental results have shown that these correlations overestimated the real value of boiling heat transfer coefficient.

This paper presents a boiling characterization that occurs on the evaporator region of a glass thermosyphon using two resistive sensors. An experimental procedure is presented for estimating the heat transfer coefficient related to the boiling phenomenon that occurs into the evaporator of the thermosyphon and these results are compared with results obtained using the correlation published by Groll and Rösler (1992). The experimental tests have been accomplished with fill charge ratio 100% for different heat loads.

2. EXPERIMENT

The thermosyphons have three regions with distinct roles in their operation, they are evaporator, adiabatic section, and condenser. The evaporator, the lower region of the tube, is heated by a hot source and the working fluid undergoes an evaporation process. This vapor, due to the pressure gradient, moves up to the colder region (condenser). In this region condenser, the vapor generated in the evaporator loses energy as heat and is condensed. The working fluid, in the liquid phase, flows back to the evaporator by gravity action, closing the cycle. The adiabatic section is located between the evaporator and the condenser. In this section, there is not heat transfer between the thermosyphon and the environment. In some cases, the adiabatic section is absent. A schematic diagram of the thermosyphon operating principle is shown in Fig. 1.

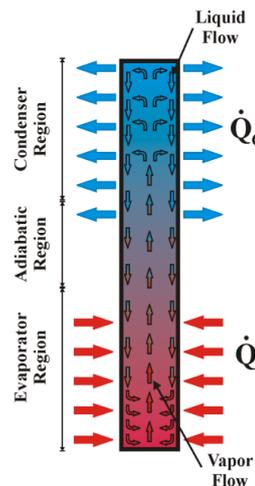


Figure 1: Sketch of the thermosyphon operating principle.

An experimental setup was built on the UTFPR facilities in order to study the heat transfer phenomena of boiling in evaporator of thermosyphons. Therefore, a thermosyphon was manufactured using a glass tube of 26 mm ID, 31 mm OD and 875 mm length. The methodology for the thermosyphon manufacturing, tests and analysis was based on (Peterson, 1994 and Reay and Kew, 2006). One fill charge ratio was considered, being 100% of volume of evaporator. The glass thermosyphon has an evaporator length of 320 mm and a condenser length of 475 mm. The evaporator region was heated using an electrical resistor in wire form (metal alloy made of copper and zinc). The wire was directly connected with a stabilized power supply in order to supply the heat loads.

The temperatures along the thermosyphon were measured using thermocouples, attached to the external surface of the evaporator, insulation and condenser. Two thermoresistors were used to measure the internal temperature of the vapor flow and two pressure transducers were used to measure the internal pressure at two points inside the evaporator. The temperatures and pressure were measured using a data acquisition system (Agilent™ Data Logger) and a computer. The experimental apparatus used for the experimental tests is schematically shown in Fig. 2.

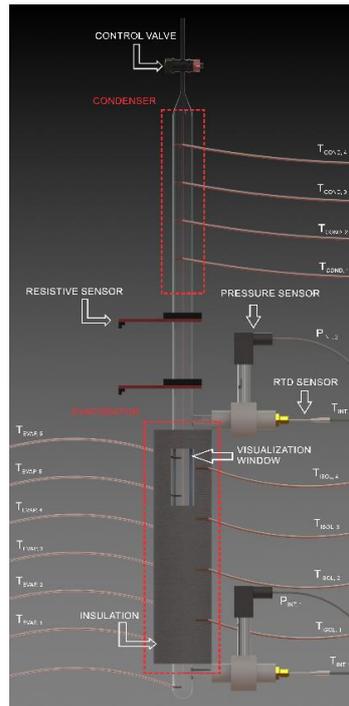


Figure 2: Diagram Schematic of Experimental Apparatus.

Still in the region of the evaporator, two holes that give access to the inner side of the glass tube, were equipped with sensors for acquisition of data of internal temperature and pressure of the fluid. In each extension a platinum thermistor type sensor (RTD Pt100) was installed for temperature measurement and a high precision pressure transducer (Omega PX419) measuring absolute pressure up to 50 psi (3.45 bar). In addition, in order to measure the surface temperature of the glass thermosyphon, 10 T-type thermocouples were installed in the outer region of the tube: 6 in the evaporator region and 4 in the condenser. The positions of the thermocouples and other sensors and the configuration of the thermosyphon assembly are shown in Figure 3.

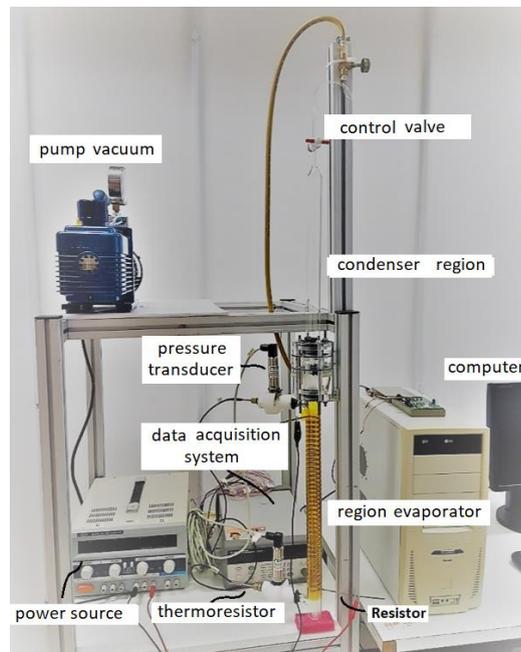


Figure 3: Experimental Bench

The experiment is started by evacuating the thermosyphon. For this purpose, a vacuum pump connected to a flexible rubber hose connected to a flow control valve and to a silicone hose is used, which directly access the inside of the thermosyphon (configuration shown from the upper region of the thermosyphon in the Figure 3). With this procedure, the

non-condensable gases can be withdrawn from the inside of the tube, in addition to reducing the internal pressure and, consequently, the boiling temperature of the working fluid.

Next, taking care that the air did not return to the interior of the equipment, the working fluid was inserted with the aid of a burette connected to a silicone hose. The working fluid used was mineral water (pH of 6.7 and electrical conductivity of $256 \mu\text{S} / \text{cm}$ at 25°C).

The resistive metal tape that evenly surrounds the outside of the thermosyphon evaporator was subjected to a potential difference at its ends by the power source. Thus it was possible to adjust the electric voltage in the resistor as well as the electric current supplied. With the passage of current, the region was heated by Joule's effect, simulating solar irradiation. The applied power (heat load) q can be related to the electric current i and the voltage V through the law of Joule through equation (1):

$$q = iV \quad (1)$$

The heat load was varied from 50 to 100 W, at intervals of 10 W, for an interval of approximately 15 minutes in order for the system to reach steady state. The temperature and pressure sensors were connected to the data acquisition system and to the computer. Through the software supplied by the company, the data from the sensors were stored.

A simplified scheme of the thermosyphon evaporator region is shown in Figure 4, where $\bar{T}_{p, ext}$ is the average external wall temperatures measured by thermocouples, $\bar{T}_{p, int}$ is the average of the internal temperatures of the tube, T_{sat} is the saturation temperature of the working fluid at a given pressure measured through the thermistor and q_{evap} is the heat transfer rate added to the thermosyphon evaporator through the power source. Note that the heat transfer to the environment (insulated tube) is being neglected and the electrical power being applied to the evaporator is equal to the energy transferred to the fluid inside the thermosyphon.

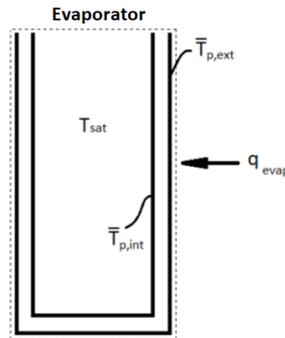


Figure 4. Simplified evaporator scheme of a thermosyphon

With the configuration shown in Fig. 4, it is possible to obtain an expression that adequately represents the average internal temperature of the evaporator wall. With the evaporator internal temperature and other initial parameters, the boiling heat transfer coefficient inside the thermosyphon evaporator is obtained experimentally.

2.1 Determination of experimental biphasic heat transfer coefficient

According to Newton's Law of Cooling for convection:

$$h_z^{exp} = \frac{q_{evap}}{(\bar{T}_{p,int} - \bar{T}_{sat}) A_{sup,e}} \quad (2)$$

The average convective coefficient for the entire surface of the evaporator, and local is related as follows:

$$\bar{h}_z^{exp} = \frac{1}{A_{sup,e}} \int_{A_{sup,e}} h_z dA \quad (3)$$

According to Incropera (2003), the thermal conduction resistance can be defined as the ratio between a driving potential and the corresponding heat transfer rate, as shown in equation (4):

$$R = \frac{\bar{T}_{p,ext} - \bar{T}_{p,int}}{q_{evap}} \quad (4)$$

For radial systems, as in the case of hollow cylinders, exposed to fluids with different temperatures on the inner and outer surfaces, the thermal resistance has the form:

$$R = \frac{\ln \frac{r_e}{r_i}}{2\pi k_{\text{vidro}} L_e} \quad (5)$$

Where r_e and r_i refer to the outer and inner radius of the tube wall, k the thermal conductivity of the thermosyphon material in the case of glass and L_e is the length of the evaporator section. Combining equations (4) and (5):

$$\bar{T}_{p,\text{int}} = \bar{T}_{p,\text{ext}} - \frac{\ln \frac{r_e}{r_i}}{2\pi k_{\text{vidro}} L_e} q_{\text{evap}} \quad (6)$$

3. RESULTS

A series of measurements were made for external surface temperatures and saturation temperatures for the powers of 50, 60, 70, 80, 90 and 100 W. From these data, the internal surface temperatures were determined through equation (6), and then the local convective coefficient was calculated through equation (2). With the use of a cubic extrapolation the most suitable functions for the heat transfer coefficient as a function of z (h_z) were determined which are shown in Fig.5.

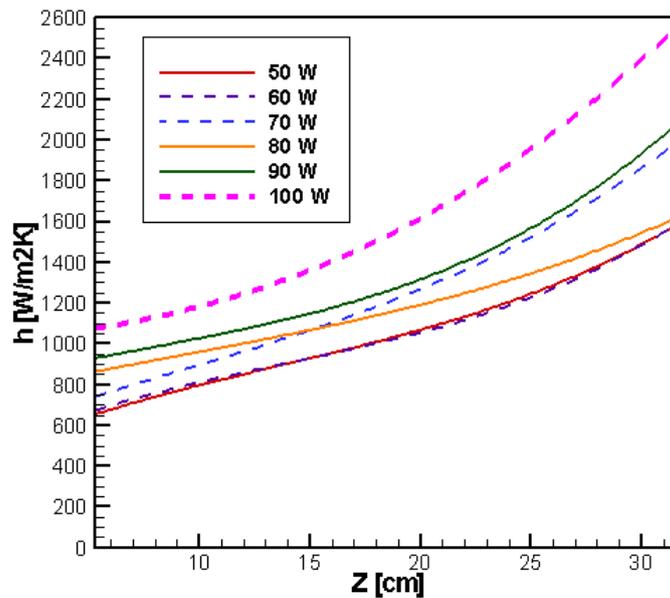


Figure 5. Local heat transfer coefficient as a function of height z and heat load.

According to the analysis of the results of the graphs of Figure 5, the values for the average boiling heat transfer coefficients were determined through equation (3). These values will be compared in Table 1 with the correlation of Groll and Rösler (1992) dealing with pool and liquid film boiling shown in equations (7) and (8):

$$R_{\text{pool}} = \frac{1}{g^{0,2} \phi q_{\text{evap}}^{0,4} (\pi d_i L_e)} \quad (7)$$

$$\phi = \frac{\rho_l^{0,65} k_l^{0,3} C_{pl}^{0,7}}{\rho_v^{0,25} h_{lv}^{0,4} \mu_l^{0,1}} \left(\frac{p_v}{p_{\text{atm}}} \right)^{2,3} \quad (8)$$

Where g is the local gravity acceleration and d_i is the internal diameter of the thermosyphon ρ_l is the specific mass of water, ρ_v is the specific mass of water vapor, k_l is the thermal conductivity of water, C_{pl} specific heat of water, μ_l is the

viscosity of water, h_{lv} is the vaporization enthalpy, p_v is the pressure measured at the sensor at the evaporator outlet and p_{atm} is the atmospheric pressure. Properties were determined at the saturation temperature. The results can be seen in Table 1.

Table 1. Comparison between experimental and theoretical biphasic convective coefficients

$q_{evap}(W)$	$h_{exp}(W/m^2.K)$	$h_{teorico}(W/m^2.K)$
50	1062	5910
60	1072	6420
70	1196	6447
80	1284	6521
90	1379	6539
100	1670	6555

For all heat loads applied to the evaporator under 100% filling ratio, the slug flow was predominant as shown in Figures 6.

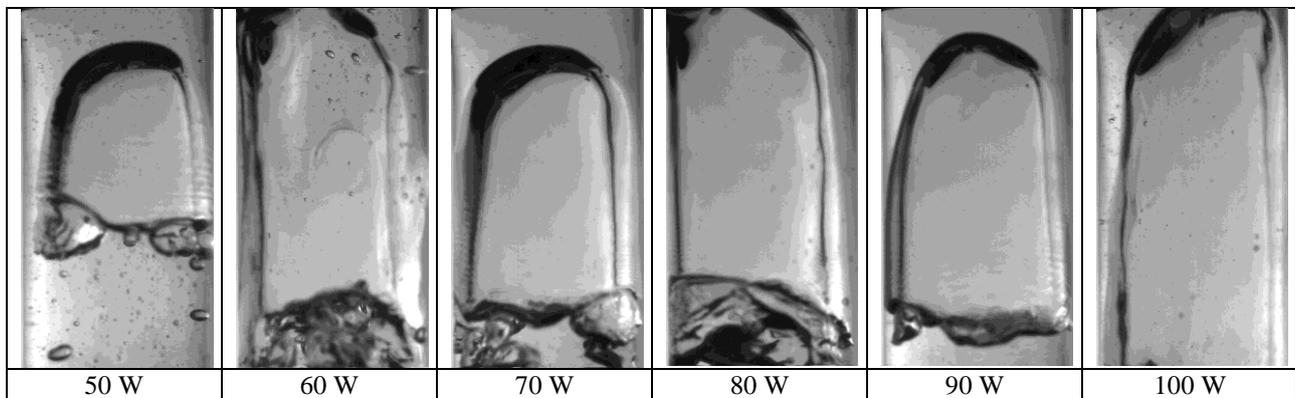


Figure 6. Photos of slug flow for all heat loads.

4. CONCLUSIONS

The results obtained experimentally for the average boiling heat transfer coefficient are in agreement with reasonable values found in the literature. On another hand, comparing the experimental results obtained here with the correlation proposed by Groll and Rösler (1992), there was a discrepancy between the experimental values because these researchers emphasize the boiling in pool and liquid film. In this way, the values obtained using the correlation are overestimated. As a result, it is necessary to develop a correlation that can better estimate the boiling heat transfer coefficient of slug flow in thermosyphons.

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6. RESPONSIBILITY NOTICE

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