Condenser Temperature Effect on the Transient Behavior of a Pulsating Heat Pipe

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Abstract

A Pulsating Heat Pipe filled up with distilled water has been designed and tested at different condenser wall temperatures and heat power inputs. The device consists in a copper tube (internal and external diameters of 3.18 and 4.76 mm), bended into a planar serpentine with five U-turns in the heated zone. The tube is closed in a loop, evacuated and partially filled with pure water, with a filling ratio of 50%. The heating section is equipped by means of two heating elements able to dissipate up to 350W as total. A cold plate, directly connected to a thermal bath, keeping the condenser at a constant temperature in the range of 10 °C up to 60 °C, permits to perform tests at different condenser temperature levels. The temperature evolutions recorded both at the condenser and at the evaporator zone allow to evaluate the overall PHP thermal performance for all the configurations tested. The experimental results show different temperature superheats at different heat fluxes as the condenser temperatures and the input powers are changed. Based on experimental data, a theoretical analysis of the heat transfer mechanisms for the dynamic behavior of the PHP has been made. The most relevant equations of boiling and evaporation have been compared with the experimental results. Based on Roshenow equation, a simplified conductive model is proposed, which is compared with experimental data. It is found that the model is in good agreement with the experimental results, especially after the full activation.

Keywords: Condenser temperature, Pulsating heat pipe, Roshenow correlation, Start-up.

1. INTRODUCTION

Closed Pulsating Heat Pipes (CPHP) are heat transfer devices that function via thermally excited oscillated motion, induced by the cyclic phase change of an encapsulated working fluid [1]. The thermo-mechanical behavior of liquid in motion within a closed volume, when surface tension are the dominant forces, yields a unstable two-phase bubble plug-liquid slug system movement, as shown in Fig 1[1][2]. This unsteady behavior constitutes the main CPHP operation principle. When compared to conventional heat pipes, CPHP can be considered a high reliability and a simple fabrication device [2][3].

Heat is added at evaporator zone and the vapor generated is transported along the pipeline to be rejected in the condenser zone due to pressure perturbations. So, two simultaneous phenomena occur in a CPHP: bubble-growing, that causes the pressure increase at the evaporator, and bubble-collapsing, that induces the pressure decrease at condenser. This growing-collapsing process yields fluctuation of liquid slugs, trapped between vapor plugs. Heat is transferred by two means: liquid-vapor and vapor-liquid phase change and sensible heat, due to the displacement of liquid slugs [4][5].

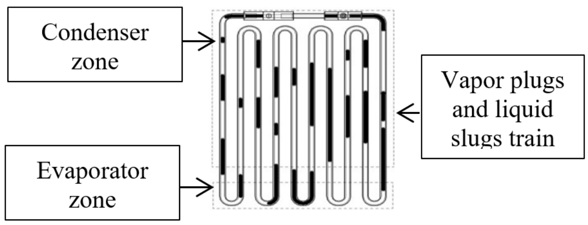

Fig 1. Schematic of a CPHP

The start-up and the working fluid circulation can be theoretically influenced by hydraulic and thermal parameters. At the low power levels, the vapor slugs and liquid plugs are separated and no liquid films are expected within the PHP until the fluid starts to oscillate. In this transient condition, the physical phenomenon that drives the oscillation plays its major role, as high amplitude oscillation are necessary for the circulation start-up of the fluid in the CHPH.

Parameters that influence the performance of a PHP are: critical diameter, heat flux, orientation, filling ratio and number of turns [5][6][7][8]. Those parameters have been already analyzed by the

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literature. However, few studies consider the condenser temperature as a fundamental parameter that directly affects the start-up in CPHP. Hansen et al [9] studied the effect of condenser temperature in a PHP and observed a significantly reduction of thermal resistance of the device, by increasing the condenser temperature. Also they found that, for a given heater input, the evaporator temperatures were lower for a condenser temperature of 30°C than of 20°C. They hypothesized that this behavior was the result of more favorable fluid properties, such as viscosity and vapor pressure at lower temperature levels.

The aim of this work is to understand the effect of the condenser temperature on the thermo-fluid behavior of a CPHP. The results obtained experimentally points out that the condenser temperature is a key parameter on the start-up of such heat transfer device. In order to find a simple way to predict this dependency, a conductive model that takes into account the Roshenow correlation is compared with the experimental results. In spite of the simplifications adopted, the conductive models, especially after the full activation, is found to be in agreement with respect to the experimental data, making it a suitable elementary tool to design of PHP devices.

2. EXPERIMENTAL APPARATUS

The CPHP consists of a capillary tube made of copper with 4.76 mm of outer diameter (OD) and 3.18 mm of inner diameter (ID) bent in a planar geometry in order to have 5 U-turns at the evaporator. The device is firstly vacuumed down in a RV8 Vacuum pump to $1x10^{-6}$ [mbar]. The sealing of the device is checked by means of Edwards Spectron 5000 Leak Detector equipment. Finally, the CPHP is charged with distilled water in a volume ratio of 50% [8.2±0.1ml], firstly degassed by continuous vacuuming cycles utilizing a volumetric pump. The internal pressure is monitored by means of a pressure transducer (Kulite XTEL-190M-15A) located at the condenser zone. CPHP was assembled in a cold plate formed by two aluminum blocks as shown in Fig 2. The cold plate is connected to a thermal bath (Lauda® Proline RP855), able to keep constant the condenser temperature (±0.01°C). In order to assure the contact, circular cross section channels were milled on both heat sink plates. Thermal contact between the device and the cold plate are improved spreading on the heat exchange surface a thin layer of Omegatherm® 201 thermal grease. Heat is provided to the evaporator by means of two High performance cartridge heaters (HLP type, OD 10 mm and 100 mm length) embedded in copper block, which are insulated by means of mineral wool. Heaters were connected to a power supply (TDK-Lambda GEN300) that can provide an electric power input up to 350W.

The temperature distribution of CPHP tested was monitored by means of 16 type T thermocouples (±0.9°C of uncertainty), distributed according to Fig 3. A DAQ-NI SCXI-1000 data acquisition system was utilized to record data during experiment.

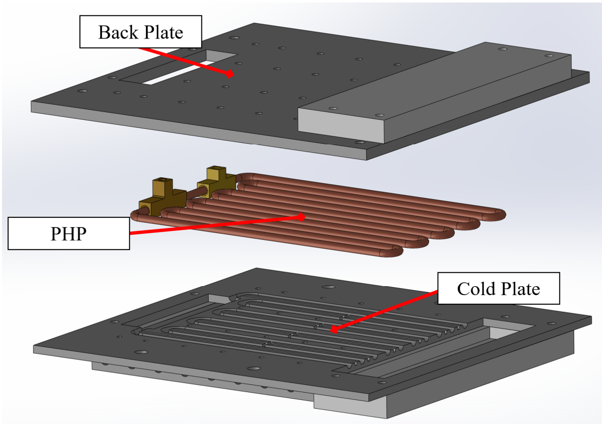

Fig 2. Back Plate and the Cold Plate.

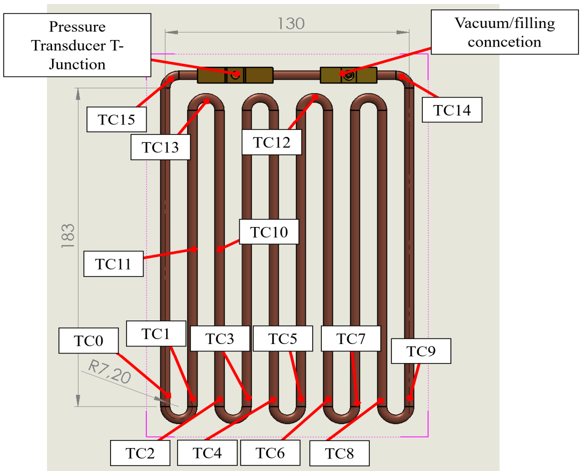

Fig 3. Thermocouples location along the CPHP tube

The experimental parameters considered in these tests were the condenser temperature [°C] and the heating power [W] supplied. The same elapsed time was considered for each experimental power level test. The measured quantities were the temperatures [°C] along the condenser zone and the evaporator zone. Pressure transducer showed direct indications in mV and its calibration equation was used to convert these measurements in [Pa] units. Thermal power was measured, based on voltage and supplied current readings, and compared with theoretical values. Thermal losses were neglected in the evaporator and condenser zone.

2.1 Experimental procedure

The experimental procedure adopted was the following:

- The condenser temperature was selected by means of the thermal bath control system;
- It was waited until all the temperatures, both at the evaporator and at the condenser zone stabilizes.
- The power supply is switched on.
- The power levels during tests were: 20, 40, 60, 80, 100, 140, 180, 230, 290, 350, 290, 230, 180, 140, 100, 80, 60, 40, 30 and 20 [W].
- Each power level was maintained for 900 seconds (resulting in 17200 seconds for a full cycle, for each condenser temperature).

3. RESULTS AND DISCUSSION

Fig 4 shows mean temperature profiles at the evaporator zone for all the tests performed. As one can observe, startup conditions were reported in all tests, including those were the bath temperature varied between 10°C to 30°C. The results show that higher bath temperatures reduce the time for the system to reach start-up conditions. Decreasing the power input, the evaporator temperature decreases for all bath temperatures. Also, at low power levels, the oscillations stop abruptly, without showing a partial stop motion, as illustrated in the right side of Fig 4.

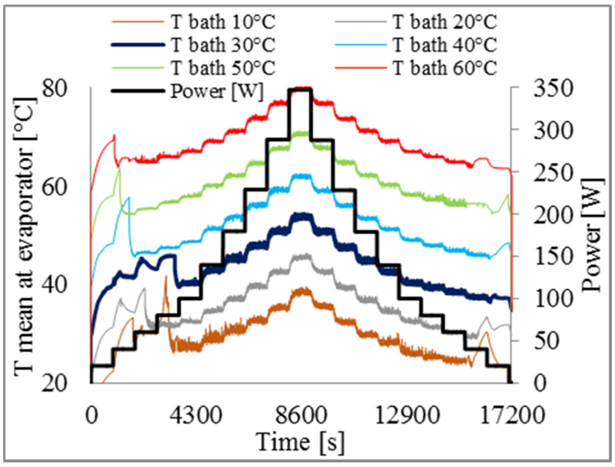

Fig 4. Mean temperature at evaporator zone for each thermal bath temperature.

Fig 5 presents pressure oscillations measured at the condenser zone. Oscillation starts after a peak of high pressure that characterize the CPHP activation.

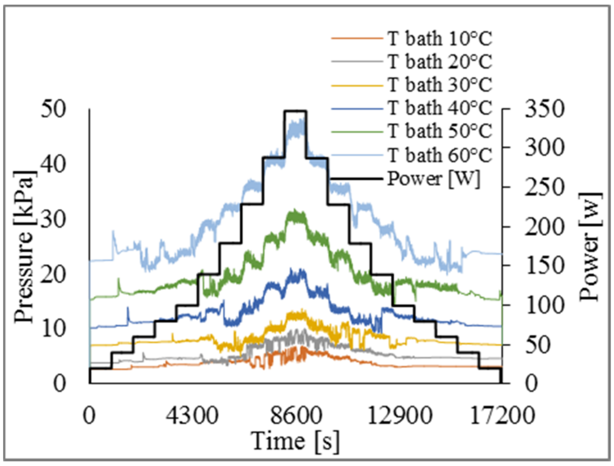

Fig 5. Pressure at condenser zone.

Fig 6 presents the temperature difference between evaporator and condenser. One can notice that the CPHP oscillation is strongly affected by the condenser temperature (or bath temperature).

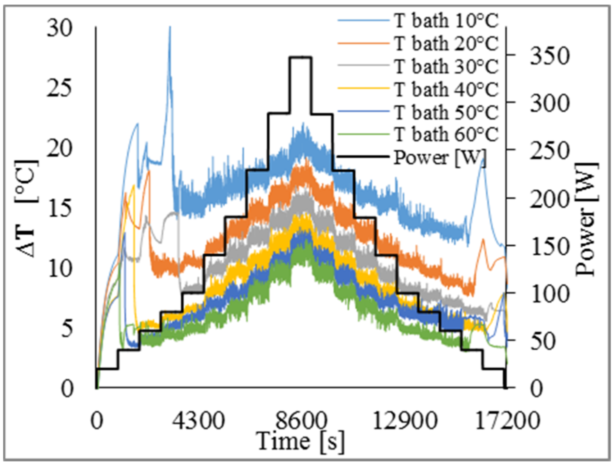

Fig 6. Mean **ΔT** for each power level at different condenser temperature.

In addition, the comparison between the power input in a ramp up and down mode, showed some degree of hysteresis. The tests results are summarized in Table 1.

Table 1. Start-up condition under thermal power effect

	Condenser temperature [°C]					
Power [W]	10	20	30	40	50	60
20	N	N	N	N	N	N
40	N	N	N	S	F	F
60	N	N	N	F	F	F
80	N	F	N	F	F	F
100	F	F	F	F	F	F
140-350-140	F	F	F	F	F	F
100	F	F	F	F	F	F
80	F	F	F	F	F	F
60	F	F	F	F	F	F
40	N	N	F	F	F	N
20	N	N	N	N	N	N

N= not working. S=start-up, F=full activation.

A convenient way to express the performance of CPHPs is the thermal resistance measurement as the ratio between the mean temperature difference between the evaporator and condenser positions and the heat transported by the tube, as given by the following equation:

$$R_{eq} = \frac{\Delta T}{Q} \quad (1)$$

In Fig 7, the overall thermal resistance is plotted against the power input. One can see from this plot that the thermal resistance decreases with the increase in the transported heat power, until the maximum experimentally applied power input (350W). Khandekar [10] also observed that when the power level decreases, the thermal resistance increases. The thermal resistance also increases with the bath temperature. Unfortunately, dry out condition was not reached, due to limitations in the electrical resistance capacity of generating power.

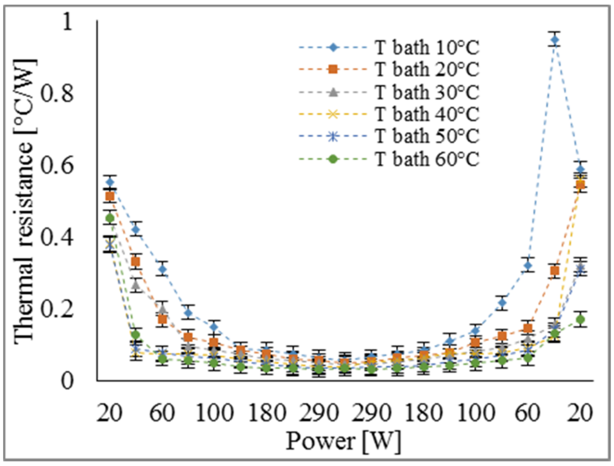

Fig 7. Thermal resistance for different condenser temperature varying thermal power.

After the CPHP two-phase flow motion full activation, the R_{eq} values tend to stay within a narrow value range, independently by the ambient temperature because, in this condition, the convective heat transfer plays a dominant role and the performance does not vary with respect to the ambient temperature.

Nevertheless, the ambient temperature has a relevant impact between start-up and the full-activation. In Fig 8, experimental data is compared with Roshenow correlation for different bath temperature. Thermodynamic properties were calculated for saturation state using EES® library. Saturation temperature was considered equal to condenser temperature and the prescribed heat fluxes, (q'') were the same as those power input levels tested.

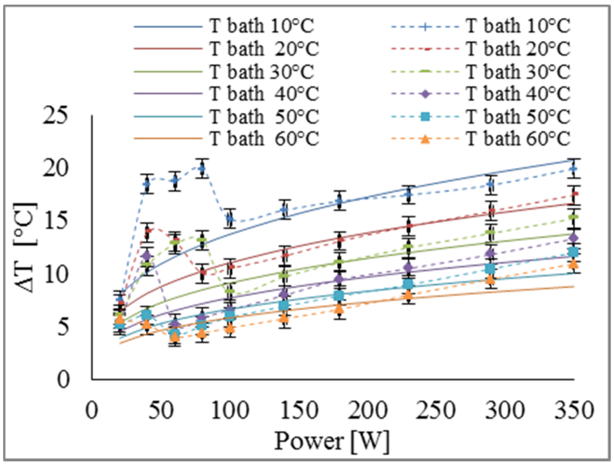

Fig 8. **ΔT** Calculated by means of Roshenow correlation and compared with experimental data.

To conduct the theoretical analysis, the evaporator section was divided in two zones, before start-up conditions: I- liquid-wall interface and II- vapor-wall interface, as shown in Fig 9.

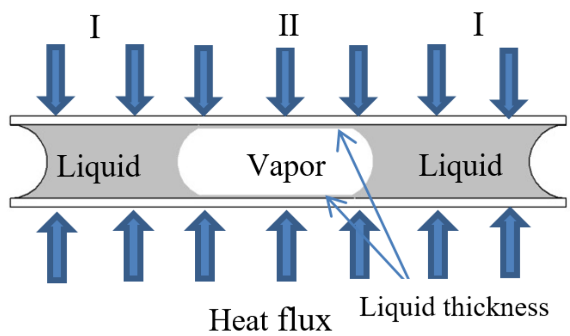

Fig 9. Train of liquid slug-vacuum plug sketch

According Qu et al [11], the working fluid wets the entire surface wall completely and creates a liquid film with a thickness large enough avoiding the non-wetting regions. This assumption allows to use pool boiling equations in both interfaces. The nucleate boiling due to the thickness reduction can be neglected before the startup occurrence.

In region I, when the wall temperature is higher than the minimum superheating required, the cavities are activated and nucleation starts. Before that, heat transfer mechanism can be considered pure conduction through the liquid film, neglecting internal convective phenomena. In this analysis the saturation temperature is equal to the working fluid bulk temperature before start-up. Bulk temperature is considered equal to the thermal bath temperature. Inner wall can be determined by:

$$T_w > T_{sat} + \frac{2\sigma}{R}\frac{T_{sat}}{\gamma_{lv}\rho_v}, \quad (2)$$

where R is the activated cavity radius, which, in this study, was assumed to be 2μm. The wall temperature estimated by means of equation 2 showed is two times larger than the temperature calculated by means of Roshenow correlation.

Fluid Bond number is expressed as:

$$Bo \equiv \frac{\rho g D^2}{\sigma} \quad (3)$$

In this work $D = D_{in} = 3.18$ mm. Therefore, Bo number calculated for D_{in} and the working fluid water takes the value of ~1.2. Taking Bo=2 [12] it is possible obtain the value for D_{crit}, which, for water from 10°C to 60°C, is larger than 5 mm. Due to equation 2, when $s \leq D_{crit}$, bubbles trend to be deformed, thus, confinement effect are always found in PHP.

Under pool boiling conditions, the classical Roshenow correlation can be used to determine the minimum ΔT needed to achieve the startup conditions [12]:

$$\frac{q''}{\mu_l h_{lv}}\left[\frac{\sigma}{g(\rho_l-\rho_v)}\right]^{1/2} = \left(\frac{1}{c_{sf}}\right) Pr_l^{-s/r}\left[\frac{C_{pl}[T_w - T_{sat}(P_l)]}{h_{lv}}\right]^{1/r} \quad (4)$$

where q'' is the applied heat flux, T_w and T_{sat} are the wall and saturation temperatures, respectively, g is gravity acceleration, μ_l is the dynamic viscosity, h_{lv} is heat latent of vaporization, Pr_l is Prandtl number for water, σ is surface tension, ρ_l, ρ_v are liquid and vapor density respectively, C_{pl} is specific heat at constant pressure. Carey [12] proposed constants of this correlation for water on scored copper, where$c_{sf} = 0.006$; $s = 1$ and $r = 0.3$

The comparison between experimental data and Rohsenow correlation for the ΔT necessary to activate the CPHP start-up, is less than 15% for the full activation. The larger difference between them is observed for the non-activated region, where the model underpredicts the data. This difference can be attributed to body and viscous forces that blocks the fluid motion. One should note that the dynamic viscosity for water at 10°C is approximately 300% highest that at 60°C.

4. CONCLUSIONS

The heat transfer performance of the CPHP partially filled with water has been investigated experimentally varying the condenser temperature. The data obtained showed the efficiency of the device, resulting in low thermal resistances.

The results indicate that Roshenow correlation predictions are in good agreement with the experimental data when CPHP reaches pseudo-steady state. However, start-up superheating is higher due to elevated thermal resistance of the CPHP.

The results also show that there is a minimum power level necessary to achieve full activation conditions, which depends on the condenser temperature. At higher power inputs, the working fluid is pushed to the condenser zone with higher kinetic energy, resulting in the increase of oscillations at the condenser zone. Also, the CPHP start-up is delayed for higher heat transfer rates. However, when it starts to oscillate, it is possible to maintain the CPHP working even with heat inputs lower than that power level necessary to achieve the start-up.

The effect of hysteresis of the minimum heat flux needed to maintain oscillation is observed as the power input decreased, during the power input variation tests. This effect can be attributed to the reduction of the viscous and inertial forces of the liquid slugs, resulting from the operation conditions of the CPHP.

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NOMENCLATURE

cp_l	: Specific heat of liquid (kJ/kg·°C)
h_{lv}	: Latent heat of evaporation (kJ/kg)
g	: gravity acceleration (m/s^2)
σ	: Surface tension (N/m)
c_{sf}	: Constant
CPHP	: Closed pulsating heat pipe
ID	: Inner diameter (mm)
HLP	: High performance cartridge heater
OD	: outer diameter (mm)
Pr_l	: Prandtl number for liquid (-)
P_l	: Liquid pressure
Q	: Thermal power (W)
q''	: Heat flux (W/m^2)
r	: Constant (-)
s	: Constant (-)
T	: Temperature (°C)
T_{bath}	: Thermal bath temperature
TC	: Thermocouple
ΔT	: Difference of temperature
$\gamma_{cb,ib}$	: Coefficient of heat transfer ($W/m^2 \cdot °C$)
μ_l	: Dynamic viscosity (Pa·s)
ρ_l	: Liquid density (kg/m^3)
ρ_v	: Vapor density (kg/m^3)

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