Flat Plate Pulsating Heat Pipes with and without separating grooves: experimental investigation on adiabatic length, coolant temperature and orientation

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Abstract. Two Flat Plate Closed Loop Pulsating Heat Pipes have been tested under various operating conditions, working fluids and different evaporator-to-condenser distances. The devices have the same geometrical features; nevertheless, on one FPPHP plate external grooves between adjacent channels have been engraved to minimize the transversal thermal conductive spreading whilst the other is a simple smooth plate. The working fluids are ethanol and FC72. Both FPPHPs have shown better performances when tilted in favorable vertical position. It has been found that higher secondary fluid temperatures within the condenser provide better thermal performances. The fluids’ different thermophysical properties and critical diameters have remarkable effects on FPPHPs behavior. Moving the condenser towards evaporator increases the heat transfer capabilities, reducing the influence of gravity. Finally, the higher thermal insulation between adjacent channels, provided by the external grooves, seems to degrade the thermal performance, especially on horizontal position and on the edge.

1. Introduction
Conceived by and patented by Akachi [1] in 1990, Pulsating Heat Pipes (PHPs) are passive, two-phase heat transfer devices that have been intensively studied over the last two decades. Their cheap and simple structure which consists of one bended tube arranged in a coil shape (for classics PHPs) or in a channel engraved within a thin conductive plate (FPPHPs), makes the Pulsating Heat Pipes highly appealing solutions for some industrial purposes, in particular electronic cooling with high heat flux densities and reduced scale heat exchangers [2]. However, their chaotic and pulsating flow nature makes the thermo-hydraulic coupling difficult to predict numerically and experimentally.

Nevertheless, important steps forward have been done both from experimental [3] and numerical [4] point of views. Indeed, visualization campaigns like those carried out by Khandekar et al. [5] and more recently by Xue et al. [6] allowed identifying the main flow regimes within the PHP: slug, semi-annular and annular ones. Furthermore, literature provides a number of parametric studies on Pulsating Heat Pipes that attempt to correlate geometrical features, operative conditions to the thermal performance.

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These studies are of primary importance because they can outline some key features of the PHPs, giving helpful guidelines for their design and addressing future researches in the field.

For example, Gi et al. [7] analyzed the influence of filling ratio on thermal performance: they found that, depending on the working fluid, the best occurred for values within the range 25-65%. Charoenasawan et al. [8] studied the effects of the channels number on the PHP behavior: they argued that a minimum number of 16 channels is able to reduce the effects of gravity and ensures the functioning of the PHP in horizontal position, for both internal diameters of 1 and 2 mm. Mameli et al. [9] correlated the PHP inclination with respect to gravity to thermal performance at different heat inputs. Bonnenfant et al. [10] compared open and closed loop PHP configurations: they found that at low input heat powers, the latter provides a better thermal performance compared to the former.

Ayel et al. [11] investigated the behavior of a FPPHP in a variable gravity field: they observed a quick thermal response during the gravity module variation and a satisfactory thermal performance on horizontal position compared to the one obtained on ground tests.

According to different authors [11-12], in FPPHP made of full plates, the thermal inertia of the latter generally prevents the device from experiencing abrupt temperature peaks during operation. Moreover, conduction thermal spreading occurs through the full plates. As a result, lower temperature gradients are generally observed as well as pressure homogenization between the channels. Since in a PHP pressure differences among adjacent channels are the main driver of oscillations under slug flow, FPPHP are potentially more exposed to a premature dry-out in horizontal inclination.

Since this latter aspect has been rarely investigated in the literature, the present paper reports the results of a multi-parametric study performed on a special FPPHP having external grooves machined on one plate to partially separate internal channels. The present work reports a quantitative analysis of the PHP steady-state thermal performance as function of the heat power inputs, heat sink temperature, orientation with respect to gravity and evaporator-to-condenser lengths. Finally, a comparison with a full plate FPPHP that has the same geometrical dimensions is also provided.

Most of the results presented in this work agree with those available in the literature. Nevertheless, some divergences emerged, underlyline, once again, that the complex thermohydraulic behavior of PHPs is still far from being fully understood.

2. Experimental setup

2.1. Closed Loop Flat Plate PHPs

![Figure 1](image_url)

**Figure 1.** Front view (bottom) and schematic channel cross section (top) of the FPPHPs: with external grooves ‘GP’ (left), smooth plate ‘SP’ (right).
Two closed loop Flat-Plate Pulsating Heat Pipes have been tested under various conditions. The FPPHPs have been manufactured from a 120x200x2 mm³ copper plate in which square grooves of 1.6x1.7 mm² have been machined to obtain 24 parallel channels (23 U-turns). Channels are then closed by means of a second plate (1 mm thick) that has been brazed on top of the latter. The brazing operation provides a perfect sealing between the adjacent internal channels and from the external environment. External grooves of 1x1.6 mm² have been machined on one FPPHP to increase the transversal thermal resistance between adjacent channels, as shown in figure 1. For the sake of simplicity, as the presence of external grooves on one FPPHP represents the unique geometrical difference between the two devices, these will be named, from now on, as Grooved-Plate (GP) and Smooth-Plate (SP) -PHPs, respectively. The former device has been previously tested in a variable gravity field during the 60th ESA parabolic flight campaign.

![Figure 1. FPPHP schematic diagram.](image)

**Figure 1.** Overview of the grooved and the smooth plates FPPHPs. The numbers on the plate indicate the location of the thermocouples.

Indeed, the GP-PHP internal channel dimensions have been sized for microgravity conditions through the Weber criterion, in the range of flow speeds between 0.1-0.2 m.s⁻¹, FC72 as working fluid and temperatures up to 60°C, more details can be found in [11]. All the tests performed in this work has been conducted on ground, it follows that the internal channel dimensions should be sized following the critical Bond number, as shown in Eq. (1). As with FC72, the hydraulic diameter of the internal channel exceeds the critical one (equation (1)) for temperatures above 30°C, ethanol has been also selected as primary working fluid ($d_{crit}(100°C) ≈ 3$ mm).

$$d_{crit} \sim \left( \frac{\sigma}{(\rho_l-\rho_v)g} \right)^{1/2}$$  \hspace{1cm} (1)

2.2. Test apparatus

The FPPHP evaporator consists of an electrical wire (Thermocoax, 1 mm external diameter, $R = 3.81 \Omega$) embedded in the grooves of a thin copper plate of 10x120x2 mm³ fixed on one edge of the FPPHP using clamps (figure 2 (left)). A controlled heat power is delivered by means of a power supply (EA® 8360-10T).

The heat sink consists of an engraved aluminum cold plate of 80x120x10 mm³ in which a mixture of water/glycol (5% in volume) flows at a controlled bath temperature, by means of a high precision thermoregulation device (HUBER® CC240wl). The cold plate is screwed on the FPPHP plate. To enhance the thermal contact between the hot source, the heat sink and the FPPHP plate, a thermal gap
filler (Tflex Series Lard Technologies®, thickness \( \delta = 0.5 \text{ mm} \), \( \lambda \approx 6 \text{ W.m}^{-1}\text{K}^{-1} \)) and a thin layer of silicon paste (Heat Sink Compound, \( \lambda = 2.9 \text{ W.m}^{-1}\text{K}^{-1} \)) have been applied at their interfaces, respectively.

Thirteen T-type thermocouples of 0.5 mm diameter have been glued on the plate to measure the PHP temperatures: five thermocouples in some grooves on the back side of the evaporator (\( T_{ev1} \) to \( T_{ev5} \)), five within grooves on the back side of the condenser (\( T_{c1} \) to \( T_{c5} \)) and the last three in the adiabatic region between evaporator and condenser (\( T_{adia1} \) to \( T_{adia3} \)), as shown in figure 2. A pressure sensor (GE P TX5076, 5 bar absolutes, 0.04% accuracy on full scale) has been connected to FPPHP through a stainless-steel pipe brazed on the plate outer groove at the condenser end. The devices have been thermally insulated during the tests with insulating Rockwool (\( \lambda = 0.04 \text{ W.m}^{-1}\text{K}^{-1} \)) and an aluminum tape coating. The thermocouples, pressure sensor and power supply are all connected to a data acquisition system (NI-cRIO-9073, NI-9214). Data recording frequencies have been set to 1 Hz for the temperature signals and 100 Hz for pressure ones.

2.3. Test campaign
Both FPPHPs have been tested under the operative conditions resumed below (see figure 3 as reference):

- Working fluids: FC72, ethanol (filling ratio \( FR \approx 50\% \) at 20°C);
- Heat power applied: from 20 W to 260 W by steps of 30 W;
- Coolant temperatures (\( T_{II} \)): 5°C, 20°C, and 40°C;
- Orientations: horizontal (\( \alpha = 0^\circ \)) with gravity vector perpendicular to the FPPHP plate, with a favorable tilting angle (\( \alpha = 45^\circ \)) with respect to ground reference, vertical favorable (\( \alpha = 90^\circ \)) and on the edge, which corresponds to the case of \( \alpha = 0^\circ \) with gravity vector parallel to the FPPHP plate;
- Different evaporator-to-condenser distances: \( L_1 = 10.8 \text{ cm} \), \( L_2 = 8.7 \text{ cm} \) and \( L_3 = 5.8 \text{ cm} \).

For each of these operating conditions, heat power has been applied to evaporator by steps of 30 W starting from 20 W up to 260 W or less if the evaporator’s dry-out limit was encountered and/or temperatures exceeded 120°C. Each power step has been maintained until the FPPHP reached a relatively steady state mode for both temperatures and pressure signals.

![Figure 3](image)

**Figure 3.** Illustration of the various FPPHP operating configurations tested: Orientation with respect to gravity (left) and different adiabatic lengths (right).

2.4. Data processing
The steady state thermal performances at each heat power step has been quantified through the overall thermal resistance, computed as in equation (2); the time and space averaged temperatures of evaporator and condenser, respectively, over the net heat power applied to the evaporator. For the latter, an
estimation of the heat leaks have been performed using the data from the empty PHP test; assuming a
two resistance model and minimizing the sum of the mean squared deviation of the computed and
measured input heat powers, it was possible to find the thermal resistances of the empty PHP (pure
conductive) and the one corresponding to the heat losses with the external environment.

\[ R_{th} = \frac{\bar{T}_{ev} - \bar{T}_{e}}{q_{in} - Q_{loss}} \]  \hspace{1cm} (2)

An uncertainty analysis has been performed for both temperatures and the measured input heat
powers, in order to estimate the impact of the systematic uncertainties on the thermal resistance values.
Assuming, for small perturbations, a linear relation between the input quantities and the observed
variable, the propagation of the uncertainties for the thermal resistance can be estimated as shown in
equation (3).

\[ e_{R_{th}} = \pm \left( \left( \frac{\partial R_{th}}{\partial Q} \right)^2 e_{Q}^2 + \left( \frac{\partial R_{th}}{\partial T} \right)^2 e_{T}^2 \right)^{1/2} \]  \hspace{1cm} (3)

In equation (3), the uncertainties associated to the heat powers and temperature measurements that
appear within the square root can be estimated from the accuracy and stability data of both the power
supply and thermocouples through an equivalent expression of equation (3). In the range of temperatures
and heat powers supplied, the systematic uncertainties for thermal resistances are in the range
\( \pm 0.02 \text{ KW}^{-1} \). It has been seen that the standard deviations of temperatures around their time-averaged
values have higher impact on the thermal resistances rather than the systematic uncertainties, as argued
by Ayel \textit{et al.} \cite{11}.

All tests were performed in a period of four months, the devices have been stored in temperature-
controlled room and the FPPHPs have shown a satisfactory level repeatability in terms of overall
temperatures and pressure trends.

3. Experimental results

3.1. Effects of PHP orientation

Results for SP and GP –PHPs with ethanol and FC72 are presented in figures 4 and 5, respectively.
According to Mameli \textit{et al.} \cite{9}, three different operating zones can be distinguished: the “Start-up”,
“Normal operating” and “Medium High heat inputs” ones. The first zone corresponds to the low heat
input levels (< 80 W) where the gravity head seems not to affect the thermal resistances that are generally
decreasing as the heat inputs increase in all orientations. On the other hand, in the normal operating zone
(between 80 and 200 W in figure 4, and 80 and 140 W in figure 5), a higher favorable inclination angle
brings a lower thermal resistance. Here, the trends of \( R_{th} \) are still slightly decreasing or almost constants
over the input heat powers.

Compared to the empty case, for the SP-PHP with ethanol (see figure 4), the values of thermal
resistances in the normal operating zone decrease down to 64% in horizontal orientation, 69% on the
edge, 78% at \( \alpha = 45^\circ \) and 82% in vertical one. Finally, in the “Medium high thermal inputs” zone (from
200 W in figure 4), evaporator dry-out occurs in horizontal and on the edge orientations, whilst for
\( \alpha = 45^\circ \) and 90° the SP-PHP provides a continuous operation up to the maximum heat power step with
no thermal crisis observed. As shown in figure 5, higher inclination angles postpone the thermal crisis,
which leads to an increase of the thermal resistance (for \( \alpha = 45^\circ \)) followed by a dried out evaporator
condition.

A favorable hydrostatic pressure head (bottom heated mode device) enhances the PHP thermal
performance. Moreover, the presence of sharp edges in the inner channel section leads to supplementary
capillary forces; these tend to break up the liquid menisci and to promote an annular flow regime, like
in a grooved heat pipe. Under such operating condition, the PHP reaches the highest thermal
performances \cite{12}, and the highest heat power tested (260 W) does not here correspond to the thermal
crisis of the FPPHP.
When oriented on the edge, the values of $R_\theta$ are generally smaller compared to those obtained in horizontal: this belongs to the transversal hydrostatic pressure head across the liquid menisci within the interconnected channels [13]. With ethanol as working fluid, this pressure head is in the order of 30 Pa for adjacent channels and 800 Pa for the channels at the PHP edges, whilst for FC72 they are of 70 Pa and 1800 Pa, respectively. With FC72 as working fluid, both FPPHP have provided a sustained and stable functioning only when gravity assisted, but not in horizontal orientation, as shown in figure 5 for the GP-PHP at $T_{II} = 20^\circ$C: this is due to the inner channel dimensions that exceeds the critical ones (see Eq. (1)) above 30°C.

**Figure 4.** Effects of the PHP orientation with respect to gravity (GP-PHP, FC72, $T_{II} = 20^\circ$C).

**Figure 5.** Effects of the PHP orientation with respect to gravity (SP-PHP, ethanol, $T_{II} = 20^\circ$C).
3.2. Effects of the coolant temperature

The use of a cold plate as heat sink with a controlled coolant bath temperature allows precisely defining the minimum PHP working temperature. As outlined in figure 6, which shows $R_{th}$ values for the GP-PHP on the edge with ethanol and FC72 as working fluids, a higher secondary fluid temperature brings lower thermal resistances. This trend has been observed for all input heat powers (with only few exceptions within the Start-up zone), all PHP orientations and both working fluids. It is obvious that the temperature-dependence of the fluid thermophysical properties can remarkably affect the PHP thermal performance.

![Figure 6. Effects of $T_{II}$ on the thermal performance with ethanol and FC72 as working fluids (GP-PHP, on the edge orientation).](image)

Indeed, switching from 5°C to 40°C, liquid dynamic viscosity of ethanol decreases of about 50% and 38% for FC72. Obviously, this means that viscous pressure losses are also decreased and the PHP performs better. However, it is interesting to observe that with FC72, $R_{th}$ change rate is smaller compared to ethanol. This could be due to the different temperature-variation rate of the thermophysical properties between the two fluids, as shown in Table 1, but also to a different flow regime induced by the internal channel dimensions with respect to the critical ones.

| $T_{sat}$ (°C) | $P_{sat}$ (kPa) | $h_{lv}$ (kJ.kg⁻¹) | $\rho_l$ (kg.m⁻³) | $C_{p,l}$ (kJ.kg⁻¹K⁻¹) | $\mu_l$ (mPa s) | $\lambda_l$ (W.m⁻¹K⁻¹) | $\sigma_l$ (mN.m⁻¹) |
|---------------|-----------------|--------------------|-------------------|------------------------|----------------|------------------------|-------------------|
| 5             | 2.35            | 11.3               | 1044              | 98                    | 802            | 1732                   | 2.23              |
| 20            | 5.8             | 23.6               | 1030              | 94.2                  | 790            | 1702                   | 2.21              |
| 40            | 18              | 54.8               | 1012              | 88.9                  | 772            | 1660                   | 2.58              |
| 60            | 47.2            | 112.4              | 989               | 83.6                  | 754            | 1614                   | 2.79              |
| 80            | 109             | 208.7              | 960               | 77.8                  | 735            | 1560                   | 3.04              |
| 100           | 226             | 358.5              | 927               | 71.4                  | 715            | 1490                   | 3.31              |

Table 1. Main thermophysical properties of ethanol (left column) and FC72 (right column) between 5°C and 100°C.
3.3. Effects of the evaporator-to-condenser distance

Evaporator-to-condenser distance represents the heat transport length of the PHP (generally called adiabatic zone in classic heat pipes); its variation implies also a change of the pressure losses generated in the flow. The results presented in figure 7 concern the GP-PHP with ethanol as working fluid, \( T_{II} = 20^\circ C \) and different evaporator-to-condenser distances (see figure 3): \( L_1 \) (corresponding to a classic configuration with condenser at top end of the plate), \( L_2/L_1 = 0.81 \) and \( L_3/L_1 = 0.54 \). Obviously, moving the cold plate towards evaporator has generated a secondary adiabatic region that, differently from the primary one, communicates only with condenser, as shown in figure 3. The temperatures recorded in this area have been considered for the estimation of the averaged condenser ones in equation (2).

![Figure 7](image-url)  
**Figure 7.** Effects of evaporator-condenser distance in horizontal and vertical position (GP-PHP, ethanol, \( T_{II} = 20^\circ C \)).

![Figure 8](image-url)  
**Figure 8.** Averaged evaporator temperatures for different evaporator-condenser distances (GP-PHP, ethanol, horizontal, \( T_{II} = 20^\circ C \)).
As shown in figure 7, adiabatic length plays an important role on the PHP behavior and performances: thermal resistances are decreased when condenser is moved towards evaporator.

Furthermore, with the minimal distance \( L_3 \), gravity seems not to affect the thermal performance anymore. Indeed, the gap between the horizontal and vertical positions almost disappear and all curves are well superimposed in the normal operating region (\( Q_{in} \geq 90W \)). Looking at space-averaged evaporator temperature curves for the three distances in figure 8, the GP-PHP oscillation start-up occurred always at 50 W (step #2) with an abrupt temperature drop. Then, a decrease in the heat transport length postpones the evaporator dry-out and provides lower temperatures for the same heat inputs. Similar results have been obtained with FC72.

3.4. Effects of the separating grooves

The presence of external inter-channel grooves on GP-PHP (see figure 9) increases the transversal conductive thermal resistance of about 65% compared to the SP-PHP. As argued by Khandekar et al. [12], the inter-channel heat balance plays a negative role in PHP operation because it reduces the local temperature gradient, which causes a decrease of the pressure instabilities degrading the thermal performance. As shown in figure 10, using ethanol as working fluid, the GP-PHP provides higher temperature difference between the middle and the right-hand points of the evaporator compared to SP-PHP; starting from 140 W they increase of almost 220%. A similar trend has been observed between other couples of points within the evaporator. However, figure 11 outlines clearly that the overall thermal performance of GP-PHP is degraded compared to SP-PHP one. Indeed, on the edge orientation and \( T_H = 20^\circ C \), the \( R_{th} \) values of the former are between 10% and 30% higher between 80 and 140 W, closer values are recorded at higher heat power steps.

![Figure 9. Schematic geometry of GP and SP –PHP channel cross-sections.](image)

![Figure 10. Effect of external grooves on the time-averaged, steady state, evaporator’s temperature differences in two points (ethanol, on the edge, \( T_H = 20^\circ C \)).](image)
Moreover, in horizontal position, the GP-PHP evaporator was totally dried-out and its performances were close to those of the empty case, whilst the SP-PHP provided a steady thermal performance keeping evaporator’s temperatures below 120°C at more than 140 W. Using FC72, $R_h$ values are well superimposed and both PHPs have a dried-out evaporator condition in horizontal position. The results with ethanol are meaningful because if, on one side, the presence of external grooves enhances flow instabilities, on the other hand they seem to limit the heat transport capability of the FP-PHP, probably because of the less homogeneous heat flux distribution around each channel with higher peaks in the grooved plate outer side, where the heat power is applied. This could probably cause the sudden formation of a dry spot starting from low input heat powers that prevent liquid from rewetting the evaporator de-priming flow oscillations. When oriented on the edge, the situation improves for the GP-PHP as long as flow instabilities are also sustained by the hydrostatic pressure head.

This trend appears to be surprising because, in horizontal orientation, an increase of instabilities between channels, provided by the presence of the external grooves, should have improved performances. It appears that it was not the case during these experiments and this specific point deserves to be further studied.

![Figure 11. Effects of external grooves on the thermal performance in horizontal and on the edge orientations (ethanol).](image)

4. Conclusions
Two copper Flat Plate, Closed Loop Pulsating Heat Pipes with a 1.6x1.7 mm$^2$ channels cross section have been tested on ground under various operating conditions using ethanol and FC-72 as working fluids with an equivalent filling ratio of 50%. External insulating grooves have been machined on one PHP (GP-PHP) whilst the other is a smooth plate (SP-PHP). According to Mameli et al. [9], three different zones have been distinguished on the thermal resistance curves over the input heat power: Start-up, Normal operating and Medium-high thermal inputs.

- The thermal performance is strongly affected by gravity and vertical orientation always provides the lowest thermal resistances.
- The secondary fluid temperature also has a strong impact due to the temperature-dependence of the thermophysical properties of the primary working fluid.
• It has been found that increasing the coolant temperature provides lower thermal resistances due to the decrease of the liquid dynamic viscosity of both ethanol and FC72.
• Moving condenser towards evaporator the thermal performance increases, the influence of gravity is reduced and evaporator dry out is postponed.
• Finally, the presence of insulating grooves on the external side of the PHP plate seems to not improve the performances. Nevertheless, temperature oscillations are increased up to two times, showing a good fluid flow activation during steady-state operation.

5. Nomenclature

| Symbol | Description |
|--------|-------------|
| \( \alpha \) | Inclination angle (deg) |
| \( \lambda \) | Thermal conductivity (W.m\(^{-1}\).K\(^{-1}\)) |
| \( \delta \) | Thickness (mm) |
| \( \sigma \) | Surface tension (N.m\(^{-1}\)) |
| \( \rho \) | Density (kg.m\(^{-3}\)) |
| \( d \) | Diameter (m) |
| \( e \) | Uncertainty (K.W\(^{-1}\)) – (W) – (K) |
| \( g \) | Gravity acceleration module (m.s\(^{-2}\)) |
| \( L \) | Length (m) |
| \( Q \) | Heat power (W) |
| \( R_{th} \) | Thermal resistance (K.W\(^{-1}\)) |
| \( R \) | Electrical resistance (\( \Omega \)) |
| \( T \) | Temperature (°C) |

Subscripts:
- adia: adiabatic
- c: condenser
- crit: critical
- e: evaporator
- in: input
- l: liquid
- loss: losses
- v: vapor
- II: secondary fluid

6. References

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