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On periodic two-phase thermosyphons operating against gravity

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Abstract

Particular passive wickless two-phase loop devices are able to operate with or against gravity. In the past a lot of devices for miscellaneous applications have been proposed each of them with its characteristics. This paper aims to analyse different kinds of equipment as special versions of a generic thermal device that the author has named periodic two phase thermosyphon (PTPT).

An original classification criterion for these devices is proposed, and the more than 50 PTPTs, that have been described in the literature, have been classified. The paper also deals with the characterisation of the global thermal behaviour of a generic PTPT by experimentally analysing the results of the temperature evolutions of the loop in a single heat transfer cycle as a periodic stable regime is reached. A mathematical model, which had been developed in previous research by the author, is presented here in improved form; it allows us to determine using the experimental temperature and pressure evolutions, changes over time in several parameters, such as the velocity of the liquid in different parts of the loop, the specific volume of vapour, and so on. The operations that are involved in the complex mechanisms of the heat and mass transfer against gravity are explained in this way.

Keywords: Two-phase thermosyphon; Anti-gravity; Periodic operation; Thermal control; Solar applications

1. Introduction

In many terrestrial applications the fluid circulation is gravity-assisted. Although many natural heat transfer processes in terrestrial applications place the hot source below the cold sink, some cases could exist where this condition is not suitable: solar applications (source above sink), aerospace devices (micro-gravity) or microelectronic cooling applications (very small buoyancy forces).

The two-phase devices operating against gravity which have been studied in the past are mostly capillary driven; these comprise conventional heat pipes [1–3], loop heat pipes [4,5] and capillary pumped loops. The latter two types are characterised by good performance and are only slightly influenced by gravity [6], but need high technology to realise the capillary structures, which are expensive in case of micro-sized or high performance devices.

In the last few years particular wickless devices able to operate against gravity have been proposed and studied: Oscillating or Pulsating Heat Pipes (OHPs or PHPs) [7–9]. These devices are thermally and capillary driven because they consist of a long pipe with an inner diameter approximately equal to the bubble diameter, so that a slug flow is induced and a natural circulation is observed [9]. In the evaporator section there is, therefore, an oscillation of temperature with high frequency, which depends on the random motion of the vapour slugs and liquid plugs [8,9].

In addition in the recent past other different wickless devices have been developed for miscellaneous applications. They transfer heat with low frequency oscillations of temperature and are able to operate even against gravity. In spite of their use in different applications these devices are characterised by similar operating modes. They can be, there-

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Nomenclature

c_p	specific heat of the evaporator	$J\cdot kg^{-1}\cdot K^{-1}$
D	diameter	m
h_{fg}	latent heat of vaporization	$J\cdot kg^{-1}$
h'_{fg}	modified latent heat of vaporization	$J\cdot kg^{-1}$
H	level difference accumulator-condenser	m
H_1	level difference evaporator-condenser	m
H_2	level difference accumulator-evaporator	m
i	specific enthalpy	$J\cdot kg^{-1}$
l_f	specific friction work	$J\cdot kg^{-1}$
L	length	m
k	thermal conductivity	$W\cdot m^{-1}\cdot K$
$\dot{m}$	mass flow rate	$kg\cdot s^{-1}$
M	mass	kg
Nu	Nusselt number	
p	pressure	Pa
Pr	Prandtl number	
q	heat flux	$W\cdot m^{-2}$
Q	heat rate	J
$\dot{Q}$	input power	W
Re	Reynolds number	
S	heat transfer surface	m^2
t	time	s
T	temperature	K
u	velocity	$m\cdot s^{-1}$
U	heat transfer coefficient	$W\cdot m^{-2}\cdot K^{-1}$
v	specific volume	$m^3\cdot kg^{-1}$
V	working fluid volume	m^3

Greek symbols

ϑ	temperature head (see Eq. (2))	K
ρ	density	$kg\cdot m^{-3}$
η	thermal properties ratio (h_{fg}/c_p)	K
μ	kinematic viscosity	$m^2\cdot s^{-1}$
ζ	heat transfer efficiency	
ω_A	external heat losses parameter $(Q_A/M_T h_{fg})$	
ω_p	internal heat losses parameter $(Q_P/M_T h_{fg})$	
ξ	density ratio ρ_v/ρ_l	

Subscripts

1–7	measurement points
I–VII	control volumes
A	accumulator
C	condenser
con	condensing section
E	evaporator
f	heat sink of condenser
l	liquid
LL	liquid line
m	heat sink of accumulator
op	period with valves open
p	parasitic energy
RL	return line
S	saturation condition
sub	sub-cooling section
T	transferred every cycle
v	vapour
VL	vapour line
w	water

fore, studied as a special type of heat transfer loop, which is here generically named periodic two phase thermosyphon (PTPT).

This paper proposes a classification criterion of the PTPT devices in the literature, considering a large number of applications and aiming to provide an organic view of their operating modes. The complex mechanisms which are involved in the heat and mass transport against gravity in PTPT devices have been explained in this paper by using a mathematical code developed in the past at the Department of Energetic of University of Pisa in collaboration with a research team of the Moscow Power Engineering Institute [12]. This code allows us to determine several thermofluid-dynamic parameters as a function of time in different parts of the loop. In this paper an improved version of this code is used to calculate several parameters in time by using experimental temperatures and pressures. These data have been collected with a generic PTPT device which is operating at the University of Pisa [10–12], and which allows an anti-gravity operation with heat source 2 m above heat sink.

2. PTPT definitions, operating modes and classification

Natural circulation against gravity has always interested researchers in the past, so that several studies, papers, patents and other documents can be found in the literature and several two phase devices have been proposed for miscellaneous applications such as solar heating, terrestrial and space thermal control, geothermal energy exploitation, warm water storage and solar refrigeration.

In spite of various different applications of these devices, some of their technical characteristics and operating modes are the same. In the past several names have been put forward to identify devices of this kind: “down-pumping heat pipes” [13], “reverse thermosyphon” [14], “passive vapour transport systems” [15], “spontaneous downward heat transport systems” [16] and so on [17–52]. In my opinion, the best name for identifying these particular devices is periodic two-phase thermosyphons (PTPT), because they can operate with the evaporator located either above or below the condenser, they can be seen as a special version of a heat pipe or, more precisely, a two-phase thermosyphon and, lastly, they all operate in a periodic heat transfer regime.

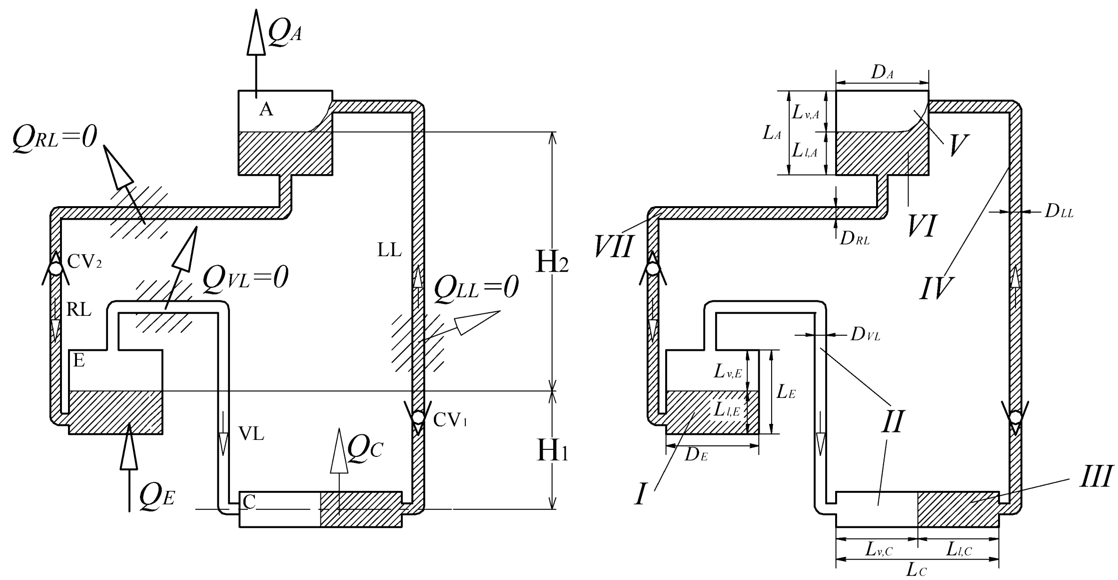

Fig. 1. Generic PTPT device diagram with definition of control volumes in right-hand side.

2.1. PTPT operating modes

A generic PTPT (Fig. 1) consists of an evaporator E, a condenser C and a tank separated from the evaporator, called accumulator A. These elements are interconnected and constitute a loop. The connecting lines are thermally insulated and they can be named vapour line *VL* (evaporator-condenser), liquid line *LL* (condenser-accumulator) and return line *RL* (accumulator-evaporator). Two check valves are inserted in the loop, the first CV_1 in the liquid line and the second CV_2 in the return line.

A single cycle of periodic heat and mass transfer can be divided into two main parts: a transfer time, where the liquid is transferred from the evaporator to the accumulator through the condenser and a return time, where the liquid collected in the accumulator comes back to the evaporator. By considering that the fluid inside the evaporator and accumulator is motionless, the kinetic terms can be considered negligible and the heat and mass transfer operation is given by the equation:

$$\big(p_{\mathrm{E}}(t) - p_{\mathrm{A}}(t)\big) - \big[\rho_{\mathrm{C,l}}(t) g \big[H_1(t) + H_2(t)\big] - \rho_{\mathrm{E,v}}(t) \times g H_1(t)\big] = \int_{\mathrm{E}}^{C} \rho_{\mathrm{E,v}}(t)\, \mathrm{d}l_{\mathrm{f}} + \int_{C}^{A} \rho_{\mathrm{C,l}}(t)\, \mathrm{d}l_{\mathrm{f}} \tag{1}$$

where H_1 and H_2 depend on time because the levels inside all the three vessels vary slightly. The right-hand term of Eq. (1) represents the friction work. The transfer of liquid from the evaporator to the accumulator starts as the left-hand side of Eq. (1) becomes higher than right-hand side. As soon as a volume of liquid is transferred from the evaporator, all the liquid collected in the accumulator must return to the evaporator and the starting conditions must be restored. In a two-phase thermosyphon the two operations occur simultaneously and a steady state regime is reached, while in a PTPT device these operations occur during two consecutive time periods so that a periodic regime is reached and an anti-gravity heat and mass transfer becomes possible. In order to restore the starting condition of a single heat transfer cycle either the pressure in the accumulator must increase or the pressure in the evaporator must decrease. In terrestrial applications there is an additional opportunity: by opening a connecting line between the two vapour zones of the evaporator and accumulator, the pressure in the two vessels is equalised and the return of the liquid is gravity-assisted, as long as the accumulator is above the evaporator. This additional line inserted in the loop is called equalisation line.

2.2. PTPT classification

According to these different modes for achieving the return of the liquid into the evaporator, a classification of all the devices in the literature is proposed. The PTPT devices in the literature are, therefore, divided into three different groups: group A, where the pressure in the evaporator falls, group B, where the pressure in the accumulator rises and group C, where there is no pressure difference and an equalisation line is employed.

Several researchers in different parts of the world have proposed passive devices to lift liquid against gravity. Some of these devices are only technical sketches, notes or other documents without further development. By starting from heat pipes, some interesting versions capable of operating against gravity have been reported in [17–20], but an initial attempt to describe the historical development of "Passive Pressure-Pumped Thermosyphons" was presented by Feldman [21] in 1987, where he focused his attention exclusively on solar devices. Another partial review of these particular heat transfer systems is available in [22]. In [23] several schemes of Russian two-phase thermosyphons are reported, few of them were operating in anti-gravity mode.

This paper supplements the previous reviews, even if it does not claim to be a complete review of the PTPT state of the art, but only an attempt to give an organic view of this kind of device, on the basis of the classification criterion mentioned above. Lastly, the following overview of PTPT devices is an extended abstract of the complete document presented in [24]. The following discussion presents a wide variety of technical details referring to many devices, while some of the experimental results reported in the literature have been omitted.

2.2.1. Group A—decrease in pressure in the evaporator

In this kind of device the condensed liquid collected in the accumulator returns to the evaporator because the pressure inside the evaporator decreases. It has been obtained in two different ways: by periodically interrupting the supply to the evaporator or by emptying the liquid inside the evaporator.

The first is typical of solar applications, because the solar heat flux is naturally discontinuous and periodic. The first device was patented by Bienert et al. [13] in 1977 and it was a "down pumping heat pipe for the purpose of maintaining an area as a roadway free of ice and snow". This device operated as a thermosyphon in winter and as "down pumping heat pipe" in summer. A sketch of this device is shown in Fig. 2(a). It consists of two chambers, evaporator and condenser, with ammonia as working fluid. As the solar heat flux is supplied to the evaporator, the ammonia vaporises and goes down to the condenser due to the ammonia vapour buffer in the condenser. The pressure inside the condenser increases; so, as the solar heat flux stops or decreases, the liquid collected in the condenser is lifted to the evaporator. An oscillating flow of vapour (downwards) or liquid (upwards) is present in the connecting pipe. In order to increase the pressure in the condenser an inert gas as argon can be inserted.

Other devices of similar type with discontinuous heating of the evaporator, were patented in the 1980s [25–27] for solar applications.

The second way to implement a PTPT with a decrease in evaporator pressure makes possible a continuous supply of heat to the evaporator. The first device was patented in 1977 by Tamburini [28]. Tamburini noted that a natural decrease in pressure inside the evaporator occurs once all the liquid has evaporated and the evaporator is empty. The pressure falls until it reaches the saturation pressure relative to the temperature of the cold sink, which is lower than the pressure in the accumulator. But this kind of device presents a problem. As the first droplets of cold condensed liquid return to the empty evaporator, they fall directly on the overheated surface and immediately evaporate. The pressure inside the evaporator suddenly increases and the liquid collected in the accumulator cannot return any longer to the evaporator. One way to resolve this problem is to insert an intermediate vessel (delaying tank) between the accumulator and the evaporator, connecting it with the evaporator through an equalisation line and a hydrosyphon, as shown in Fig. 2(b). This solution allows the return of the liquid to the overheated surface to be delayed till a volume of liquid capable of quickly cooling the evaporator wall temperature has been transferred from the accumulator. Tamburini presented this device for electronic equipment cooling applications, but no experiments were reported.

In 1980 Nasonov et al. [14] presented a "reverse thermosyphon (RTS)" for solar heating applications; this was based on the Tamburini's device, and experimentally investigated with various working fluids: water, ethanol and carbon tetrachloride. They inserted an auxiliary condenser connected with the evaporator as shown in Fig. 2(c). By getting the idea from Nasonov's work, Sasin developed a large number of devices for different applications in solar heating, geothermal energy exploitation and refrigeration [29–34]. He developed and experimented PTPTs with two condensers, with an open delaying tank, with a delaying tank inserted inside the evaporator and many other configurations.

He was the first to set up a mathematical model to predict the PTPT thermal behaviour over time. This model, based on a lumped parameter method, was built by observing the experimental operating modes of the device [29]. Moreover, Sasin has tried to combine a PTPT device with a vapour ejector heating [33] or a refrigeration cycle [34] (Fig. 2(d)).

Recently Buz et al. [35] proposed a PTPT operating with little accumulation of liquid and a high functioning frequency (0.25–2 Hz). In this case the hydrosyphon is made possible by a special conformation of the connecting pipes. In 2003 the present author and co-workers proposed a group A-device for electronic equipment cooling with FC72 as working fluid [36]. The device has shown high operating temperatures, large volumes, which are difficult to shrink, and a low flexibility in the operating modes so that, a group A-device will be hardly applied to the miniaturised electronic cooling in future.

2.2.2. Group B—increase in pressure in the accumulator

In this kind of PTPT device the pressure oscillation occurs inside the accumulator instead of the evaporator. These devices show good thermal performances fully appropriate to electronic equipment cooling applications, because the temperature of the evaporator, to which the chip must be connected, remains constant throughout. However, in order to generate a pressure oscillation in the accumulator, a periodic supply of heat to this vessel is needed (pumping by heating). The only device found in the literature and classifiable in this group, is the PTPT tested by Ogushi et al. in 1986 [37], which was proposed for electronic equipment cooling in space applications. In the experiments Ogushi observed that the time related to the heat pumping is approximately 36% of the whole heat transfer period and the thermal power required to make the working fluid circulate was 7–10% of

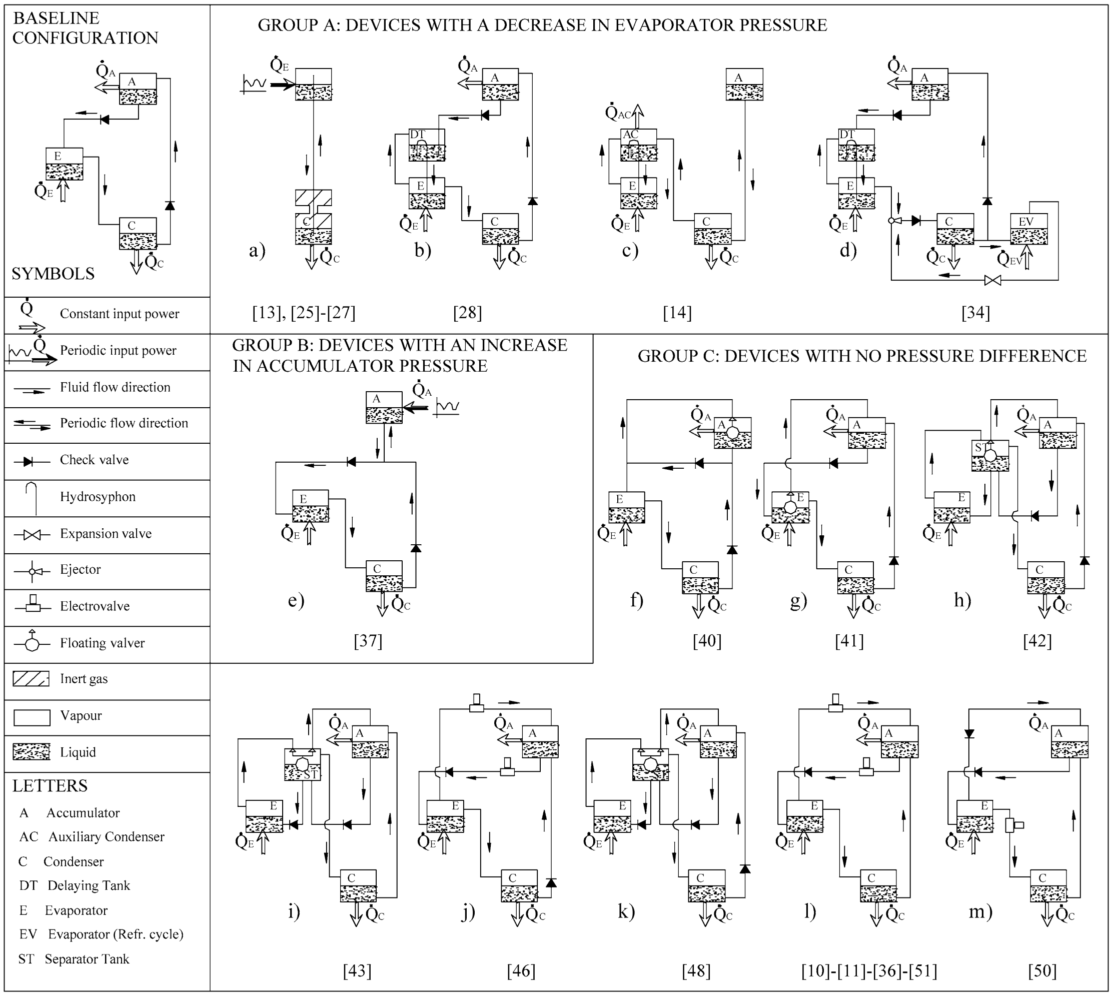

Fig. 2. Different versions of PTPT devices and their main technical topics.

the power supplied to the evaporator. A scheme of the apparatus is shown in Fig. 2(e).

2.2.3. Group C—no pressure difference

The PTPT devices in this group are gravity-assisted, because the accumulator is placed over the evaporator. The pressure and temperature oscillations inside the evaporator are lower than those with group A devices, so that the thermal resistances are lower too. Most group C devices were invented in the 1980s for solar heating and water storage applications, but they have also been proposed for electronic equipment cooling [10–12].

The main feature of these devices is that the evaporator is connected with the accumulator by means of an equalisation line, which is periodically opened and closed. The opening and closing of the equalisation line is achieved with valves, which can be electrically or mechanically (floating valves) activated.

The first PTPT device in this group was patented by Anderson in 1940 [38]. It was a passive device used as an air-conditioning system with the evaporator located in the room and the condenser located outdoors at a lower level. In this device a floating valve was used to allow periodic connection of the evaporator with the condenser, but no check valves were included in the loop.

In 1977 Feldman [39] made a solar PTPT for house heating applications, with a floating valve inserted in the accumulator. In this device the inlet of the working fluid was on the top of the evaporator (Fig. 2(f)).

In 1977 Bohanon patented a device "applicable to solar heating installations" similar to Feldman's device [40]. In this device a trap for the vapour is inserted in the loop after the condenser, and two floating valves are inserted in the accumulator. In 1981 De Beni et al. [41] presented a solar device with a floating valve inside the evaporator (Fig. 2(g)). In 1982 Neeper et al. [42] presented a solar device with the floating valve inserted in a separate vessel called separation tank (Fig. 2(h)). Then De Beni et al. [43] improved the system proposed in [42] by including the floating valve in the separate vessel modification, as shown in Fig. 2(i); their experimental results were obtained using methanol as working fluid. In [16] and [44] De Beni et al. strengthened their original system by making other improvements, while in [45] an example of their system was used to produce warm water by solar heating in a mountain refuge.

In 1986 Hedstrom and Neeper [46] modified their device previously proposed in [42] by inserting two electrovalves in the loop, as shown in Fig. 2(j). They noted that the use of an electrical valve improves the heat transfer and the efficiency of the device, however the apparatus is not completely passive. They also noted that a good thermal insulation of the accumulator slightly improves heat transfer efficiency, but sharply increases the operating temperatures of the devices. In 1988 Neeper presented a simplified analytical model [22] that is useful in predicting the overall behaviour, the transferred energy rates and the efficiency of a solar PTPT device, but does not taken into account the operations occurring in a single cycle that influence the heat transfer performance.

In 1986 Feldman [47] reported another version of his device presented in [21], inserting an additional accumulator, which is at the same temperature and level as the condenser.

In 1997 Kawabata et al. [48] proposed a PTPT to operate an electronic equipment thermal control based on a previous paper [49]. In this case both the opening and the closing of the equalisation line are controlled by one floating valve, as shown in Fig. 2(k), while an additional check valve is inserted after the condenser. They tested the device under these conditions: R141b and R134 as working fluids, 2.2 m and 10 m as H, and 1000 up to 3500 W as evaporator input powers.

Recently, Kadoguchi et al. [50] have proposed a PTPT with a motor-operated valve located after the evaporator, Fig. 2(m). The pressure inside the evaporator increases when the motor-operated valve is closed. They have observed that a two-phase flow in the liquid line is induced in a way dependent on the pressure inside the evaporator, as soon as the motor-operated valve is opened.

The present author together with his co-workers have studied a couple of PTPT devices since 1998 [10,12,24] (Fig. 2(l)). In particular, a large device has been set up in order to study the complex heat transfer mechanism involved in anti-gravity mass transfer. More recently, the feasibility and the performance of a miniature PTPT as thermal control system for microelectronic equipment cooling have been studied [36,51,52].

3. Mathematical background

A simplified but effective analysis of the thermal behaviour of a PTPT can be carried out by considering only the average temperatures relative to a single heat transfer cycle as a stable periodic regime is reached. In [22] Neeper proposed an analysis of the thermal performances by using an approximate model and considering the whole system as operating in only two equilibrium states, which are defined by two different temperatures: $\overline{T}_E$, which represents the average temperature of the working fluid in the evaporator and $\overline{T}_C$, which represents the average temperature in the condenser. In order to operate against gravity, according to Eq. (1), the pressure difference between accumulator and evaporator must be higher than the hydrostatic liquid head, so that the particular quantity $\overline{T}_E - \overline{T}_A$ which pushes up the liquid may be regarded as a "temperature head" ϑ [22] and can be expressed by:

$$\vartheta = \frac{\rho_{C,l} g H}{(\frac{\Delta p}{\Delta T})_S} \tag{2}$$

where the term $(\Delta p/\Delta T)_S$ is the average rate of change of saturated vapour pressure with respect to temperature. As reported in [22], if the energy and mass balances are expressed, the following relationship can be obtained:

$$\frac{\overline{T}_E - \overline{T}_C}{\eta} = \xi + (1-\xi)\frac{\vartheta}{\eta} - (\omega_A - \omega_p) \tag{3}$$

where η is the ratio of the latent heat of vaporization to the specific heat at constant pressure ($\eta = h_{fg}/c_p$), ξ is the vapour–liquid density ratio, ω_A is the ratio of the external heat loss rate Q_a—hypothesising heat loss as being concentrated in the accumulator—to the latent energy of the liquid mass transferred during each cycle ($M_T h_{fg}$) and, lastly, ω_p is the ratio of parasitic to latent energy ($Q_P/M_T h_{fg}$).

Parasitic energy is typical of group C devices and represents a sort of internal heat losses. It is the energy that is collected in the accumulator during the opening of the valves, because a portion of the evaporator hot vapour rises to the accumulator and mixes itself with the cold vapour which is there. The accumulator vapour is, therefore, at a higher temperature than the accumulator liquid: the product of this difference of temperature and the thermal capacity of the vapour represents the parasitic energy. The parasitic energy may be zero if the valves are activated efficiently and the time during which they are open is small with respect to the rest of the single cycle time.

Neeper concluded that if the accumulator is thermally insulated ($\omega_A = 0$) and the parasitic energy is zero ($\omega_p = 0$), a difference of temperature $\overline{T}_E - \overline{T}_C$ is required even with a negligible difference in level between accumulator and condenser ("temperature head" equal to zero): this difference of temperature is equal to $\eta\xi$. Some values of $\eta\xi$ are reported in Table 1 for various working fluids in a temperature range typical of solar and thermal control applications. In the cases reported there, $\overline{T}_E - \overline{T}_C$ remains below 2 K with exception of ammonia and HCFC141b. Ammonia presents $\overline{T}_E - \overline{T}_C$ up to 10 K. The $\overline{T}_E - \overline{T}_C$ is therefore the minimum temperature difference in a PTPT device required to make the working fluid circulate.

Moreover, as noted above, in a PTPT the heat and mass flow depends on time. An accurate analysis requires a numerical solution of the time-dependent differential equations which describe the complex heat and mass transfer mechanisms involved.

In the literature there are several mathematical models that resolve the time-dependent differential equations for OHPs or PHPs; some of these are reported in [8,53,54],

Table 1
Product $\varepsilon \cdot \eta$ for some working fluids with the boiling temperature shown in brackets, at various operating temperatures

	T [K]	h_{fg} [J·kg^{-1}]	c_p [J·K^{-1}·kg^{-1}]	ρ_v [kg·m^{-3}]	ρ_l [kg·m^{-3}]	$\xi\eta$ [K]
Ammonia	293	1186300	4758.3	6.723	610.5	2.74
(239.3 K)	323	1050400	5067.8	15.824	563.0	5.82
	353	874000	5759.9	33.992	505.7	10.20
HCFC141b	293	233026.7	879.5	3.157	1248.4	0.67
(306 K)	323	221772.1	879.5	8.186	1194.6	1.73
	353	220209.1	879.5	18.519	1126.8	4.11
FC72	293	94368.5	1041.89	3.441	1698.5	0.18
(328 K)	323	86403.9	1088.22	10.702	1631.1	0.52
	353	77491.5	1134.55	26.810	1571.4	1.16
Methanol	293	1210000	2460	0.000	0.791	0.00
(337.7)	323	1147150	2520	0.00005	0.764	0.03
	353	1084400	2520	0.002	0.7	1.17
Water	293	2453800	4182.2	0.022	998.4	0.01
(373 K)	323	2382800	4181.2	0.080	987.9	0.05
	353	2309200	4197.4	0.290	972.0	0.16

while the models to simulate the PTPT behaviour are extremely rare. In general, the models developed for OHPs and PHP propose relations in estimating the bubble-plug flow, which is the pressure oscillation promoter, while a semi-empirical model has been proposed by Groll et al. in [55]. On the other hand in PTPT devices no slug flow is observed in the evaporator and the mathematical model must be different.

The first time-dependent mathematical model developed for PTPT devices has been proposed by Sasin in [29] for group A devices. It is based on the observation of the experimental thermal behaviour of a generic group A device. He had divided the single heat transfer cycle into four parts according to the different operations observed during the experiments. The model has been rearranged, tested and experimentally validated for a group C device in [12]. The original model has been improved by the present author for group C devices and is described here. In this case a single transfer period is divided into two parts: a transfer operation with connecting valves closed and a return operation with connecting valves open. In both parts, seven types of one-dimensional control volumes are defined, as shown in Fig. 1, right-hand side. Control volume I is the liquid inside the evaporator, control volume II is the vapour entrapped in the evaporator, in the vapour line and in the section of the condenser occupied by the vapour, control volume III is the liquid inside the condenser, control volume IV is the liquid in the liquid line, control volume V is the vapour volume in the accumulator, while the control volumes VI and VII are the liquid in the accumulator and the return line, respectively.

This mathematical model uses a lumped capacitance method to describe the parameters of each control volume, and is based on the following two hypotheses: non-condensable gases are not present in the loop, and evaporator and accumulator are constantly in a saturation condition ($T_{v,E} = T_{l,E} = T_E$, $p_{v,E} = p_{l,E} = p_E$ and $T_{v,A} = T_{l,A} = T_A$, $p_{v,A} = p_{l,A} = p_A$).

In this way the rate of the change of energy in control volumes I and II over time can be expressed by:

$$Q_E = -h_{fg}(T_E)\frac{dM_I}{dt} + M_I\frac{di_I}{dt} + M_{II}\frac{di_{II}}{dt} \tag{4}$$

while the rate of the change of mass can be expressed over time by

$$\frac{dM_I}{dt} = -\frac{dM_{II}}{dt} - \frac{dM_{con}}{dt} \tag{5}$$

where (dM_{con}/dt) represents the mass of vapour that condenses in control volume II. This last term depends on the condensing heat transfer regime and on the fluid-dynamic conditions of the vapour inside the condenser. In the case studied the exchanger transferring heat to the cold sink is a tube–tube type cooled by water at a constant temperature T_f. This exchanger, generically termed as condenser consists of two sections: the first is occupied by the vapour ($L_{v,C}$, Fig. 1) and really works as a condenser, and the second ($L_{l,C}$, Fig. 1) works as sub-cooler.

The liquid velocity inside the condenser depends on the different operations occurring during the heat transfer that have been experimentally observed; it is therefore expressed by the relationships:

$$\begin{cases} u_{l,C} = 0, \quad p_E \leqslant p_A + \rho_{l,C} g H + \Delta p_{EA} \\ u_{l,C} = \sqrt{\frac{2}{\rho_{l,C}}\left(p_E - p_A - (\rho_{l,C} g H + \Delta p_{EA})\right)} \\ \quad p_E > p_A + \rho_{l,C} g H + \Delta p_{EA} \end{cases} \tag{6}$$

where Δp_{EA} is the pressure drop relative to the mass transfer from the evaporator to the accumulator, as experimentally determined and reported, together with a pressure characterisation of the experimental loop, in [11].

The change of rate of heat and mass over time in the condenser is given by the following relationships for the condensing section,

$$h_{fg}(T_E)\frac{dM_{con}}{dt} = Q_{con} = U_{con}\pi D_C L_{v,C}(T_E - T_f) \tag{7}$$

$$\frac{M_{\mathrm{I}}}{\rho_{\mathrm{l,E}}}+\frac{M_{\mathrm{II}}}{\rho_{\mathrm{v,E}}}=V_{\mathrm{E}}+V_{\mathrm{LL}}+\frac{\pi D_{\mathrm{C}}^{2}}{4}L_{\mathrm{v,C}} \tag{8}$$

where Q_{con} is the condensing heat rate, $\rho_{\mathrm{l,E}}$ and $\rho_{\mathrm{v,E}}$ are the densities of liquid and vapour at the temperature T_{E}, V_{E} and V_{LL} are the volumes of the evaporator and liquid line and, lastly, U_{con} is the condensing heat transfer coefficient expressed by Chato's correlation [56] in the case of horizontal tubes with an angle subtended from the centre of the tube to the chord forming the liquid level equal to 60°:

$$U_{\mathrm{con}}=0.55\sqrt[4]{\frac{g\rho_{\mathrm{l,E}}(\rho_{\mathrm{l,E}}-\rho_{\mathrm{v,E}})k_{\mathrm{l,E}}^{3}h'_{\mathrm{fg}}}{\mu_{\mathrm{l}}(T_{\mathrm{E}}-T_{\mathrm{f}})D_{\mathrm{C}}}} \tag{9}$$

In Eq. (9) h'_{fg} is a modified heat of vaporization according to $h'_{\mathrm{fg}}=h_{\mathrm{fg}}+\frac{3}{8}c_{\mathrm{pl,E}}(T_{\mathrm{E}}-T_{\mathrm{f}})$.

The changes in heat and mass over time for the sub-cooler section are given by:

$$\begin{aligned}&U_{\mathrm{sub}}\pi D_{\mathrm{C}}L_{\mathrm{l,C}}(T_{\mathrm{l,C}}-T_{\mathrm{f}})\\&\quad=Q_{\mathrm{sub}}\\&\quad=\frac{\mathrm{d}M_{\mathrm{con}}}{\mathrm{d}t}c_{\mathrm{pl,C}}(T_{\mathrm{E}}-T_{\mathrm{l,C}})+M_{\mathrm{III}}c_{\mathrm{pl,C}}\frac{\mathrm{d}T_{\mathrm{l,C}}}{\mathrm{d}t}\end{aligned} \tag{10}$$

$$\frac{\mathrm{d}M_{\mathrm{III}}}{\mathrm{d}t}=\frac{\mathrm{d}M_{\mathrm{con}}}{\mathrm{d}t}-\rho_{\mathrm{l,C}}u_{\mathrm{l,C}}\frac{\pi D_{\mathrm{C}}^{2}}{4} \tag{11}$$

where U_{sub} can be calculated from the following Nusselt numbers in case of laminar flow (Eq. (12)) and turbulent flow (Eq. (13)) [57]

$$Nu=3.66,\qquad Re<10^{4} \tag{12}$$

$$Nu=0.023\,Re^{4/5}Pr^{1/3},\qquad Re\geqslant10^{4} \tag{13}$$

The rate changes for heat and mass over time in the accumulator are expressed by Eqs. (14) and (15), while Eq. (16) represents the rate of change of the mass inside all control volumes within the loop [57]:

$$\frac{\mathrm{d}M_{\mathrm{VI}}}{\mathrm{d}t}=\rho_{\mathrm{l,C}}u_{\mathrm{l,C}}\frac{\pi D_{\mathrm{C}}^{2}}{4}-\frac{\mathrm{d}M_{\mathrm{V}}}{\mathrm{d}t} \tag{14}$$

$$\begin{aligned}&U_{\mathrm{A}}S_{\mathrm{A}}(T_{\mathrm{m}}-T_{\mathrm{A}})+\rho_{\mathrm{l,C}}u_{\mathrm{l,C}}\frac{\pi D_{\mathrm{C}}^{2}}{4}c_{\mathrm{pl,C}}(T_{\mathrm{l,C}}-T_{\mathrm{A}})\\&\quad=M_{\mathrm{VI}}\frac{\mathrm{d}i_{\mathrm{VI}}}{\mathrm{d}t}+M_{\mathrm{V}}\frac{\mathrm{d}i_{\mathrm{V}}}{\mathrm{d}t}+r(T_{\mathrm{A}})\frac{\mathrm{d}M_{\mathrm{V}}}{\mathrm{d}t}\end{aligned} \tag{15}$$

where S_{A} is the external surface of the accumulator, U_{A} is the heat transfer coefficient (natural convection in this case with $U_{\mathrm{A}}=5\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$) and T_{m} represents the temperature of the heat sink, which is in contact with the external surface of the accumulator ($T_{\mathrm{m}}=293$ K).

$$\frac{\mathrm{d}}{\mathrm{d}t}(M_{\mathrm{I}}+M_{\mathrm{II}}+M_{\mathrm{III}}+M_{\mathrm{V}}+M_{\mathrm{VI}})=0 \tag{16}$$

It is interesting to note that $\mathrm{d}M_{\mathrm{IV}}/\mathrm{d}t=\mathrm{d}M_{\mathrm{VII}}/\mathrm{d}t=0$. An explicit Euler numerical method is used to solve all the 10 linear differential equations.

The second part of a single transfer period is linked to the return of the collected liquid from the accumulator to the evaporator. The equations which describe the rate changes for mass and heat over time are the same as those for the transfer operation that are written above. A further hypothesis is, however, assumed; the parasitic energy and the velocity of the liquid in the condenser are equal to zero.

In this way the hydrodynamic condition which allows the fluid collected in the accumulator to come back to the evaporator, as soon as the valves are opened, is expressed by the relationship:

$$p_{\mathrm{v,A}}+\rho_{\mathrm{l,A}}gH_{2}>p_{\mathrm{v,E}}+\Delta p_{\mathrm{AE}} \tag{17}$$

where the level difference H_2 is defined in Fig. 1, while Δp_{AE} represents the pressure drop related to the mass transfer between accumulator and evaporator, which has been experimentally determined [11].

The velocity of the cold liquid front level in the accumulator is expressed in time by:

$$\begin{aligned}u_{\mathrm{l,A}}(t)=\Biggl[2\biggl(\frac{p_{\mathrm{v,A}}(t)-p_{\mathrm{v,E}}(t)-\Delta p_{\mathrm{AE}}(t)}{\rho_{\mathrm{l,A}}}+g\cdot H_{2}(t)\biggr)\\\times\biggl(\frac{\rho_{\mathrm{l,E}}}{\rho_{\mathrm{l,A}}}\frac{D_{\mathrm{E}}^{2}}{D_{\mathrm{A}}^{2}}-1\biggr)^{-2}\Biggr]^{1/2}\end{aligned} \tag{18}$$

This model tested on experimental data has shown prediction errors lower than 2% for the average temperature of the accumulator, evaporator and condenser measured with an experimental error of ±0.5 K. In 2003 Buz [35] proposed a model for a group A-PTPT with a small diameter evaporator by analysing its behaviour over time. The differential equation written for the two consecutive liquid plugs with a single big vapour bubble inside can be solved by applying a Runge–Kutta method. However in this case the capillary force cannot be neglected.

4. Experimental facility

The experimental facility which has been arranged in our laboratory at the University of Pisa consists of a generic group C apparatus and its control and measurement devices. In Fig. 3 a scheme of the apparatus is shown. It comprises an evaporator (internal volume of $11.2\times10^{-3}\ \mathrm{m^3}$), a copper tube-tube condenser comprising of three internally microfinned tubes ($D_{\mathrm{C}}=9.52$ mm, $L_{\mathrm{C}}=0.7$ m) and an accumulator (internal volume of $19\times10^{-3}\ \mathrm{m^3}$). A water flow rate of $0\text{–}3\times10^{-4}\ \mathrm{m^3{\cdot}s^{-1}}$ at the inlet temperature of 293 K cools the condenser, which is 1 m below the evaporator, while the accumulator is 1 m above the evaporator; in this way the height of the anti-gravity liquid column H is 2 m.

The apparatus has the connection lines opened by two special valves (K, Fig. 3), which are electrically activated by two floating level switches located in the accumulator. All the connection lines ($D_{\mathrm{VL}}=D_{\mathrm{LL}}=D_{\mathrm{RL}}=10.4$ mm) are made of copper and are thermally insulated. The liquid line is made of glass to allow visualization of the two-phase or

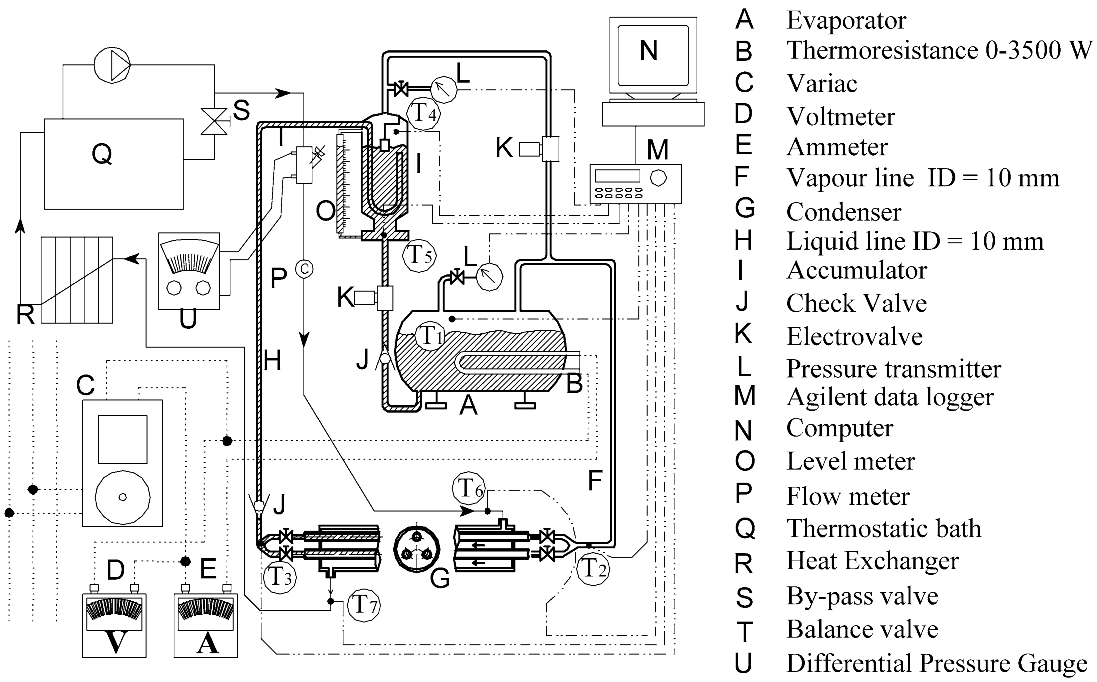

Fig. 3. Scheme of the experimental apparatus with the temperature measurement points (1–7).

single phase upward flow (anti-gravity flow). The evaporator and condenser are thermally insulated too, while the accumulator is cooled by the surrounding air at a temperature of 293–295 K. The working fluid is the fluorinate HCFC141b.

The working fluid temperatures are monitored over time by 7 thermocouples located at various points along the loop for each time step; in particular, T_1 is the temperature of the liquid in the evaporator $T_{\mathrm{l,E}}$ ($\overline{T}_1 = \overline{T}_{\mathrm{E}}$), T_2 is the vapour temperature at the inlet of the condenser, T_3 is the temperature of the working fluid flow at the outlet of the condenser ($\overline{T}_3 = \overline{T}_{\mathrm{C}}$), T_4 and T_5 are the temperatures of vapour and liquid in the accumulator ($\overline{T}_4 = \overline{T}_{\mathrm{A}}$), while T_6 and T_7 are the temperatures of the cooling water at the inlet and outlet of the condenser, respectively. The pressures in the accumulator and evaporator are measured by two pressure transmitters (0–2.5 bar, accuracy 0.25% of full scale range). All the temperatures and pressures of the device are sampled by a data acquisition system and stored in a PC (scanning time step is 10 s). The volume V_{T} of working fluid transferred in every cycle has been measured by a level meter located inside the accumulator: its accuracy is about $\pm 0.76 \times 10^{-6}$ m^3.

The input power is directly supplied to the working fluid in the evaporator with an electric thermo-heater. The experimental procedure which has been followed during the tests is described here. At the starting time a vacuum is created inside the loop in order to insert 17.5 kg of working fluid at 293 K.

In particular, a liquid mass of 10.5 kg is inserted in the evaporator, while the other part is inserted in the accumulator and condenser. The evaporator filling ratio is, therefore, about 85%, while the accumulator filling ratio is about 50%. The working volume transferred every cycle is 2.2×10^{-3} m^3. We are assuming that a heat transfer cycle starts as soon as the valves are opened and ends as soon as the valves are opened again, after their closing.

All the tests have been carried out for 20 hours, but the apparatus has reached a periodic stable regime after only a few heat transfer cycles. The temperatures are periodically repeated at each opening and closing of the valves, with a difference lower than 0.15 K between two consecutive cycles.

5. Experimental results

5.1. Heat and mass transfer characterisation

The global thermal behaviour of a PTPT as a heat transfer device can be analysed by taking into account the mean temperatures of a single heat transfer period. A PTPT, in fact, can be seen as a thermal device which exchanges heat with a hot source (evaporator) and two heat sinks (condenser and accumulator) at two different temperatures (T_{f} and T_{m}), but no work is exchanged with external systems. In the case of this experimental analysis $T_{\mathrm{f}} = T_{\mathrm{m}}$.

The thermal efficiency of PTPT as a heat transfer device is defined as the ratio of the heat supplied to the evaporator to the heat dissipated by the condenser, and, according to the simplified model proposed by Neeper in [24], can be expressed by the relationship

$$\zeta = \frac{Q_{\mathrm{C}}}{Q_{\mathrm{E}}} = \frac{1 + (\overline{T}_{\mathrm{E}} - \overline{T}_{\mathrm{C}})/\eta}{(1+\xi)(1+\vartheta/\eta) + \frac{Q_{\mathrm{p}}}{M_{\mathrm{T}} h_{\mathrm{fg}}}} \tag{19}$$

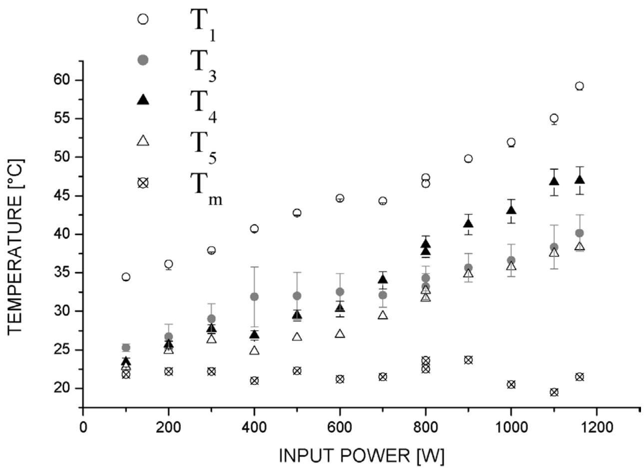

Fig. 4. Temperatures averaged over time at various measurement points along the loop against input powers.

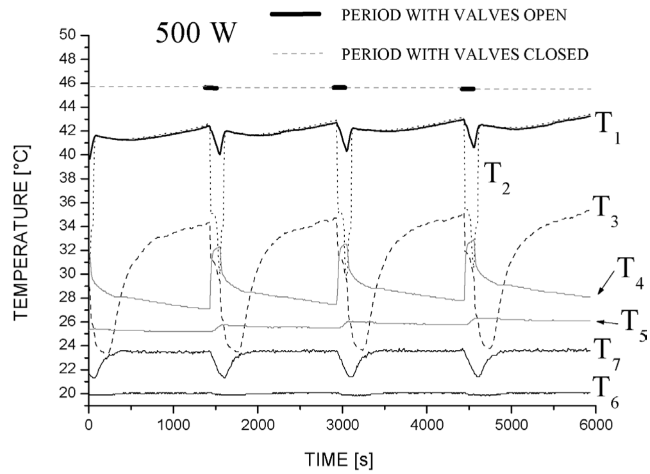

Fig. 5. Temperature evolutions for 4 consecutive heat transfer cycles for an input power of 500 W.

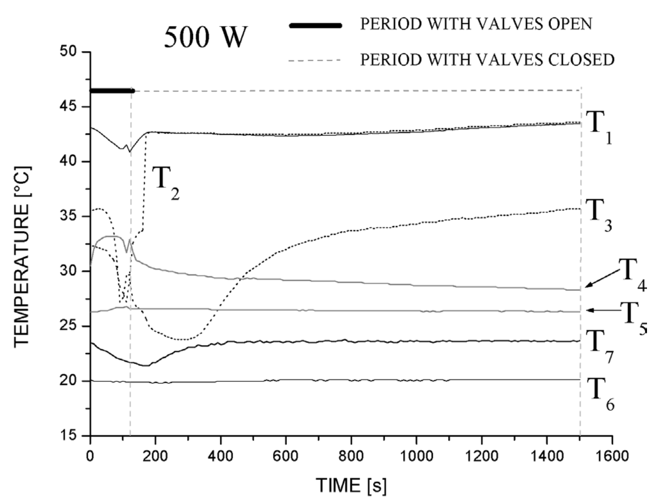

Fig. 6. Temperature evolutions during a single heat transfer cycle as a stable regime is reached for an input power of 500 W.

where Q_P is the parasitic energy calculated with the term $Q_P = c_{p,vA} M_{v,A}(T_4 - T_5)$.

Fig. 4 shows the experimental mean temperatures of the main elements of the loop as a periodic regime has been reached. For low input powers the parasitic energy, representing internal losses, is negligible, the temperatures of the vapour and liquid in the accumulator are approximately the same and the heat transfer efficiency is very high. On the other hand, as the input power increases, the temperature of the vapour and liquid in the accumulator diverges and the parasitic energy becomes important, so that the efficiency falls to 0.85 and T_4 becomes higher than T_3. From Fig. 4, another important topic must be mentioned; the temperature of the evaporator T_1 rises at the same ratio as the temperature in the condenser T_3 so that, in this case ($T_f = T_m$), the thermal resistance is approximately constant at various input powers.

Moreover, in order to describe the mechanisms that are involved during the transfer of heat and mass, the thermal behaviour of the apparatus over time must be considered. Typical working fluid temperature evolutions in various sections of the loop are shown in Fig. 5, where the temperatures for 4 consecutive cycles with an input power of 500 W are shown, while the same temperatures are shown for a single cycle in Fig. 6.

The pressure evolutions of the fluid inside the evaporator and the accumulator are qualitatively similar to the temperature evolutions, because in the accumulator and the evaporator the working fluid is in a saturation condition. Some comments on the qualitative temperature evolutions must now be made.

As the transfer period starts, when the valves are opened, the temperature T_1, the temperature of the vapour T_2 at the inlet and of the liquid T_3 at the outlet of the condenser fall because a mass of cold liquid returns to the evaporator. At this time no mass flow rate in the condenser is observed. Moreover, the temperature of the vapour T_4 in the accumulator increases because hot vapour leaves the evaporator (parasitic energy). The difference between the temperatures of the cooling water at the inlet and the outlet of the condenser (T_6 and T_7) determines the heat rate dissipated by the condenser and the sub-cooler over time according to the relationship:

$$\dot{Q}_C = \dot{Q}_{con} + \dot{Q}_{sub} = \dot{m}_W c(T_7 - T_6) \tag{20}$$

where $\dot{m}_W$ is the mass flow rate, which is constant during the experiment, and c is the specific heat of the cooling water.

At the start of the transfer period the power rate $\dot{Q}_C$ is at a maximum; it decreases as soon as the valves are opened. On the other hand, as soon as the valves are closed, the temperature and the pressure inside the evaporator rise, no mass flow rate is still observed in the condenser, and the temperature of the vapour at the inlet of the condenser stays constant, while the temperature of the liquid at the outlet falls. This is due to the thermal inertia of the liquid inside.

As soon as the pressure in the evaporator can lift the liquid column after the condenser ($t = 200$ s in Fig. 6), the

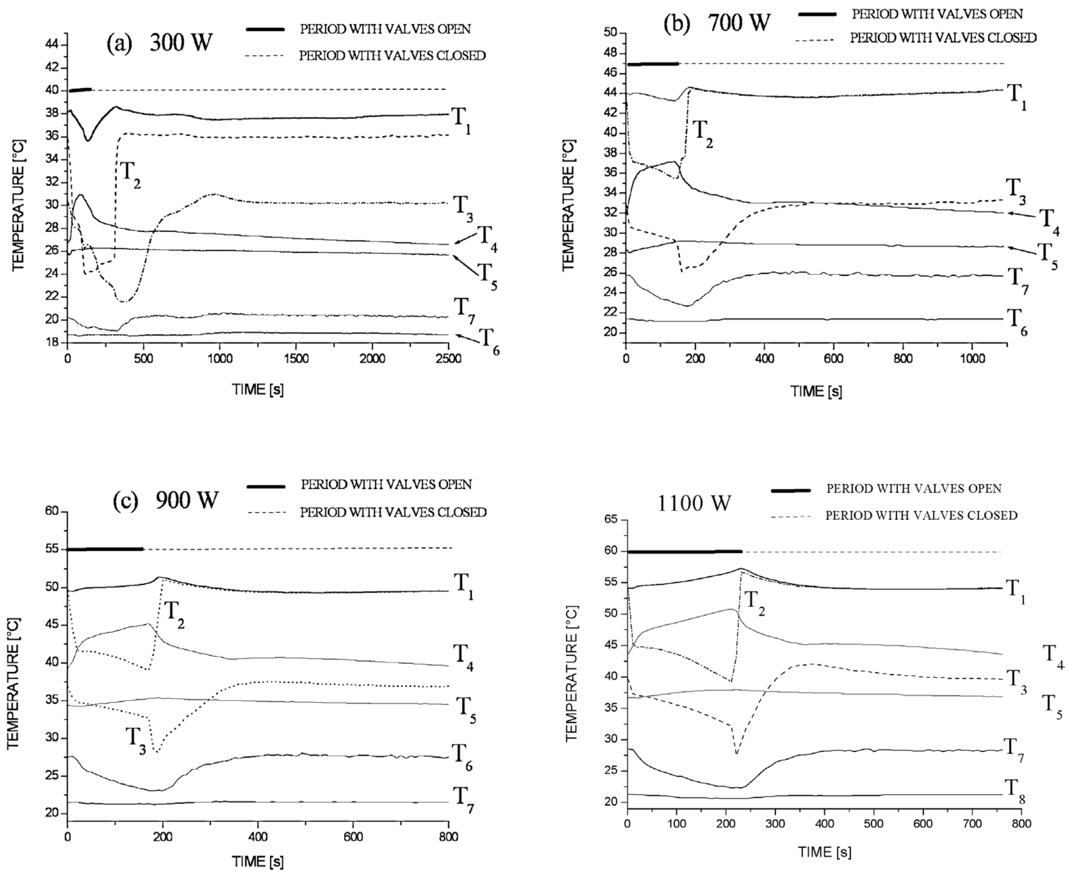

Fig. 7. (a)–(d) Temperature evolutions during a single cycle for an input power of 300 W (a), 700 W (b), 900 W (c), 1100 W (d).

liquid condensed and collected in the condenser starts to move up to the accumulator, the temperature T_1 in the evaporator decreases (expansion), as well as the temperatures of the accumulator T_4 and T_5. The temperature of the liquid T_3 decreases at first, as the cold liquid, which before was motionless inside the condenser, starts to move, but after a time ($t = 400$ s in Fig. 6) it increases again. From a certain time ($t = 600$ s in Fig. 6) all the temperatures stay approximately constant till the end of the heat transfer cycle, except for T_3 which rises till the end of the cycle. Figs. 7(a)–(d) show a series of similar temperature evolutions in the loop over time during a single heat transfer cycle, for input powers of 300, 700, 900 and 1100 W.

5.2. Single heat transfer cycle analysis

In order to understand how the single operation occurring in a single cycle can influence the heat and mass transfer, some qualitative comments on the temperature evolutions for various input powers must be made. By analysing the temperature evolutions in Fig. 7, it can be noted that all the temperatures in the loop except for the temperature in the evaporator T_1 have very similar qualitative shapes at various input powers. The difference of temperature T_1–T_4 is connected with the pressure difference evaporator-accumulator and, according to Eq. (1), with the anti-gravity circulation promoter in the loop.

While the temperature in the accumulator T_4 varies sharply at the opening and closing of the valves (return operation), T_1 varies also, however, in a different way. From Fig. 8 it can be seen that for low input powers (300 W, 500 W, 700 W), T_1 decreases during the period when the valves are open. When the valves are closed, T_1 increases sharply, after a while T_1 approaches a constant value. For high powers (900 W, 1100 W), T_1 increases during the period when the valves are open. When the valves are closed, T_1 drops sharply to approach a constant value after some time.

In order to understand how the main parameters of the device can influence the temperature in the evaporator, an experimental analysis on the single heat transfer cycle has been carried out for different input powers.

If Eqs. (4)–(18) are written in a discrete-time form, it is possible to calculate over time the mass and volume of each control volume (I–VII), the length of the liquid inside the condenser and the velocity of the liquid in the evaporator and accumulator, by assuming that the discrete time step is equal to the experimental one (10 s), by knowing the starting condition of the working fluid in the device and by measuring the temperatures and the pressures in time at various points of the loop (1–7). The volume and the mass of all

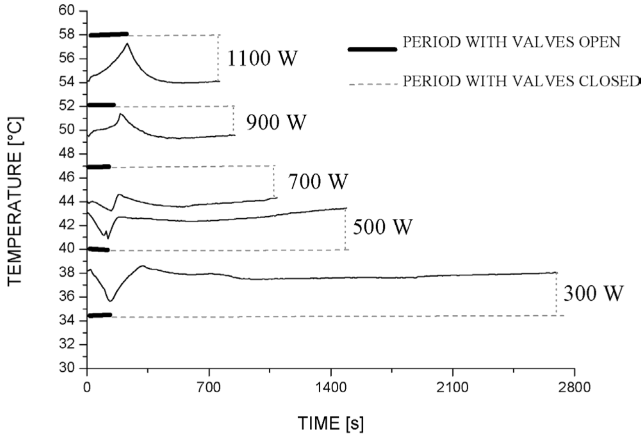

Fig. 8. Evolution of liquid temperature T_1 in the evaporator for a single cycle at various input powers.

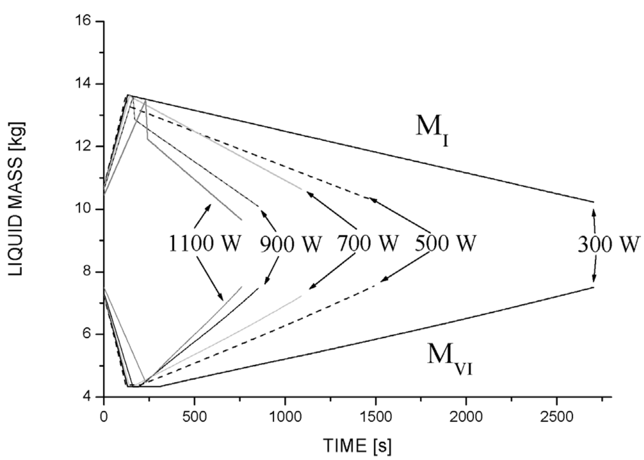

Fig. 9. Evolution of the liquid masses in control volumes I and VI for a single cycle at various input powers.

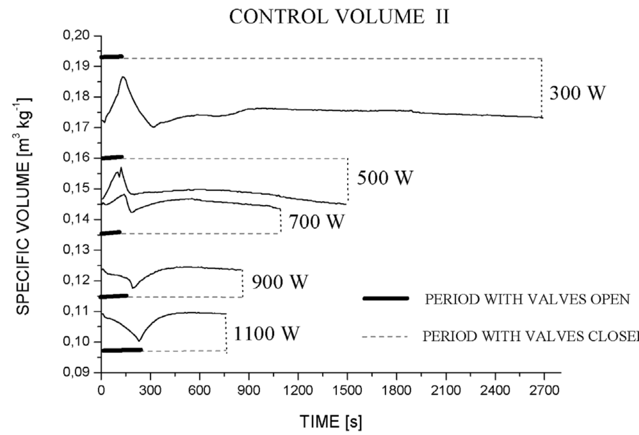

Fig. 10. Evolution of vapour specific volume in control volume II for a single cycle at various input powers.

the control volumes as a function of time have been, therefore, determined by using, as input data, the experimental pressures and temperatures in the mathematical model previously described. The mass and volume of the accumulator, which are the only data directly measured with time, has been compared with the experimental data and the errors are lower than 5%. A further control have been made on the mass of the working fluid calculated as the sum of the mass of each control volume (I–VII) calculated individually. For every time the maximum error is lower than 1%.

Fig. 9 shows the calculated masses of the liquid in the control volumes I (evaporator) and VI (accumulator) over time.

The data in Fig. 9 show that at high input powers (900 W, 1100 W) a sudden fall in the mass M_I, and hence in the volume, of the liquid in the evaporator is observed, whereas for low input power (300–700 W) it is not noted. This depends on the condition of the evaporator when the valves are opened. At this moment the condenser is full of motionless liquid and cannot function as a condenser. The power rate removed by the condenser is quite low and the input power in the evaporator induces a compression of the vapour and, therefore, an increase in pressure. As a result, as soon as the valves are closed, an expansion of the vapour in control volume II (see Fig. 1) is observed and, therefore, a sudden fall of the mass and volume of liquid in the evaporator. As soon as the expansion is finished, a constant mass flow rate of liquid is noticed.

On the other hand, if the input power is low (300–700 W), during the opening of the valves a cold liquid mass returns to the evaporator and the temperature inside increases, but the pressure necessary to lift the liquid against gravity is not reached. In this way, as soon as the valves are closed, the mass in the evaporator starts to evaporate, but the low pressure cannot lift the liquid against gravity and no mass in the accumulator is transferred.

The working fluid temperature T_1 in the evaporator therefore depends on the specific volume of the vapour within control volume II over time. If this specific volume decreases, a compression is produced and an increase in temperature is expected, while in the opposite case, a decrease in temperature is expected if the specific volume increases. The specific volume within the control volume II is shown over time in Fig. 10 for 5 input powers. The transient behaviour is quite analogous to the transient behaviour of T_1 in Fig. 8.

In the case of large working fluid volumes transferred in every cycle, the different behaviour of T_1 and the specific volume of the vapour in II at various input powers depends on the ratio expressed by

$$\frac{\dot{Q}_E \Delta t_{op}}{c_p \rho_{E,l} V_E} \geqslant \vartheta \qquad (21)$$

where V_E is the volume of the liquid in the evaporator and Δt_{op} is the time with the valves open. If the right-side hand of Eq. (21) is much lower than the “temperature head” ϑ a decrease in the evaporator temperature T_1 is expected as the valves are opened, while in the opposite case an increase in T_1 is expected. A more rigorous approach should at least take into account the thermal inertia of the evaporator walls,

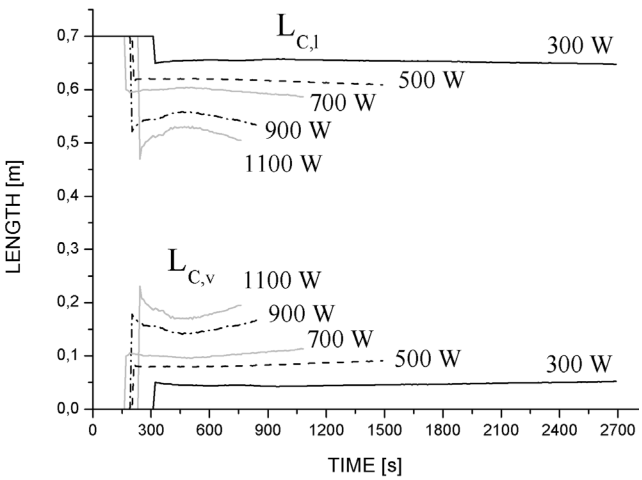

Fig. 11. Length of the vapour section and liquid section in the condenser for a single cycle at various input powers.

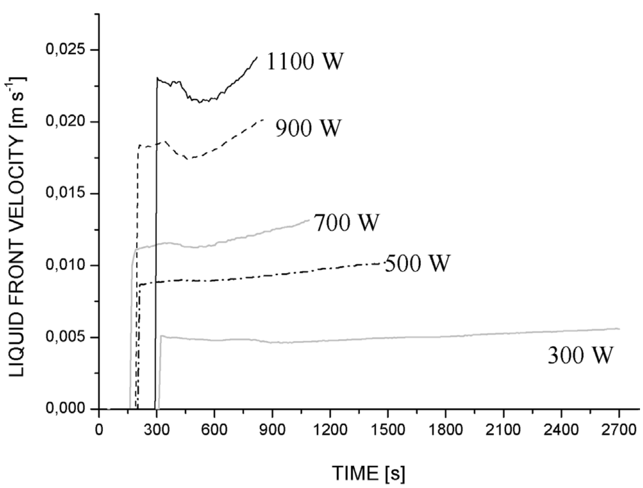

Fig. 12. Liquid velocity in the condenser for a single cycle at various input powers.

but this simple criterion, which is confirmed by other experiments carried out on a differently shaped and sized mini evaporator [36–51], can guide the design of the evaporator.

By using experimental pressure and temperature as input data of the mathematical model other parameters can be calculated. Fig. 11 shows the length of the heat exchanger occupied by the liquid (sub-cooler) and by the vapour (condenser) as a function of time. For low input powers the heat exchanger remains almost full of liquid, while for 1100 W about 25% is occupied by vapour. If the condenser is small, it may be that during the sudden expansion the whole condenser will be occupied by the vapour and a two-phase flow can be expected in the anti-gravity line.

In conclusion, the velocity of the liquid flow rate in the condenser is quite low, as shown in Fig. 12. It is estimated as being equal to 0.005 $m \cdot s^{-1}$ ($Re = 600$) for an input power of 300 W, while it is 5 times more for 1100 W. These values confirm that no large mass flow rate can be observed in these devices.

6. Conclusions

Periodic two-phase thermosyphons (PTPTs) are a special kind of unsteady state heat transfer devices capable of operating with or without gravity support, even against gravity. In the literature a large number of these devices is presented for miscellaneous applications, using several different names. This paper has tried to show how many of the heat transfer devices, which have been proposed in the literature can be seen as different versions of the same generic type of heat transfer device. The classification of a large number of these devices reported in the literature is presented together with the mathematical background developed so far.

An experimental activity has been carried out to find out how the main parameters can influence the thermal behaviour of such a device as the input power increases from 300 to 1100 W.

A mathematical code, able to predict the thermal behaviour of the apparatus with a low error margins, has been used to calculate, from the experimental data, the trend of the main parameters of the heat transfer cycle over time. The evolution of the temperature in the evaporator, which characterises the heat and mass transfer of the device, has been measured and explained, as has the thermal behaviour of the whole loop.

In conclusion this paper has attempted to provide a starting point for practical applications of these two-phase devices.

Acknowledgement

This paper is a summary of a long research project; several papers have been written on it over the last few years. I wish to express my thanks to all my co-authors of the papers reported in the references and, in particular, to professors V.Ja. Sasin and E. Latrofa, who have given me important scientific advices throughout.

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