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AN EXPERIMENTAL INVESTIGATION OF A CO_2 PULSATING HEAT PIPE

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ABSTRACT

Since their invention in the mid-nineties, pulsating heat pipes (PHPs) have been typically suited for micro electronics cooling. However, behavior and efficacy of these devices under low temperature (less than room temperature) is a new and promising challenge. This paper attempts to present preliminary experimental results of pulsating heat pipes operating with an evaporator average temperature ranging from -20°C to 5°C and having carbon dioxide (CO_2) as the working fluid. The results show the effects of input heat flux, inclination angle and volumetric filling ratio on the PHP thermal performance. The present results enables one to conclude that CO_2 can be used as a working fluid to efficiently transfer heat at low temperature.

KEY WORDS: pulsating heat pipes, carbon dioxide, heat exchangers.

1. INTRODUCTION

Pulsating heat pipes are a relatively new type of heat transfer devices which can be classified as a special category of heat pipes. The basic structure of a typical pulsating heat pipe consists of capillary tubes having no internal wick structure, working as an evaporator and condenser coil. Behind its constructional simplicity lies an intriguingly complex thermo-hydrodynamic operational characteristic: a two-phase, bubble-liquid slug system formed inside the coil due to the dominance of surface tension. The coil receives heat at one end and is cooled at the other. Temperature gradients give rise to temporal and spatial pressure instabilities and phase change phenomena. Bubbles are generated and grow in the evaporator and simultaneously collapse in the condenser. This growing and collapsing processes act as pumping elements, transporting the liquid slugs in a complex oscillating-translating behavior. This phenomenon is a direct result of thermo-hydrodynamic coupling of pressure/temperature fluctuations with the void fraction distribution and is responsible for the heat transfer, essentially as a combination of sensible and latent heat portions (Charoensawan et al., 2003 and Khandekar et al., 2003).

Nowadays the use of PHPs for thermal control of electronic devices is already well-established, allowing not only the cooling of high power levels but also operation in applications with elevated flux density (Karimi and Culhan, 2004). However, the behavior of PHPs under temperatures lower than those used for cooling of electronic devices was not well studied so far. As a result, this work intends to experimentally investigate the performance of a Closed Loop Pulsating Heat Pipe (CLPHP), working between 5ºC to -20ºC, with carbon dioxide (CO_2) as the working fluid. This temperature level is appropriate for some refrigeration applications, where a PHP could be applied as a passive heat exchanger with low thermal resistance. Carbon dioxide was chosen as the working fluid for being environmentally friendly, offering benefits such as non toxicity, non-flammability, easy availability, low price, no need of recycling, and compactness of components. It has favorable transport properties (low viscosity and high thermal conductivity), which combine to improve the heat transfer characteristics (Sawant et al., 2003).

2. BACKGROUND

Previous works have studied PHPs as general purpose passive heat exchangers. Katoh and Xu (2004) tested two heat sink technologies: all-metal heat sink and pulsating heat pipe heat sink. The pulsating heat pipe heat sink thermal resistance was 40% less than the thermal resistance of the all-metal heat sink.

To meet the thermal requirements of next generation CPUs with a low profile heat sink, Vogel and Xu (2005) tested four heat sink technologies (i. e., all-metal heat sink, embebbed heat pipe heat sink, vapor chamber heat sink and pulsating heat pipe heat sink) and their associated prototypes. Measured test results verified that all prototypes, which utilized liquid-to-vapor phase change technologies, met the heat sink design requirements, including the critical 0.18°C/W sink-to-air thermal resistance. However, the thermal resistance of the all-metal heat sink was 44% greater (i. e., 0.26°C/W) than the measured thermal resistance of the tested prototypes.

According to Khandekar and Groll (2006), the following thermal-mechanic parameters have major influence over a CLPHP behavior:

- input heat flux;
- volumetric filling ratio of the working fluid;
- device orientation with respect to gravity;
- total number of turns;
- internal diameter of the CLPHP tube;
- thermo-physical properties of the working fluid.

In this paper we focus in the first tree parameters. However, CLPHP performance is also linked with internal flow patterns, which in turn, depends on the complex combination of above listed parameters. Various flow patterns other than capillary slug flow, developing/semi-annular and fully developed annular flow have also been reported, having a significant effect on the thermal performance (Khandekar and Groll, 2004).

2. EXPERIMENTAL SETUP

A scheme of the experimental apparatus is shown in Figure 1. The PHP coil is divided in three sections: condenser, adiabatic section and evaporator. The condenser is placed inside a cooling manifold, with circulating ethanol kept at constant temperature by a thermostatic bath as the cooling fluid. The evaporator is inserted in an aluminum block, where a heating resistance is placed through a hole. Spaces between the PHP turns inside the evaporator block are filled with thermal paste. The heating resistance is connected to a DC power supply, providing the required power for each test. The adiabatic section is covered with a thick polyurethane shelf, as well as the cooling manifold and the aluminum block, with the whole assembly placed inside an insulated box. Maximum thermal gain of the assembly at the worst possible external environment condition was estimated at 2.7W. Further design characteristics are presented in Table 1.

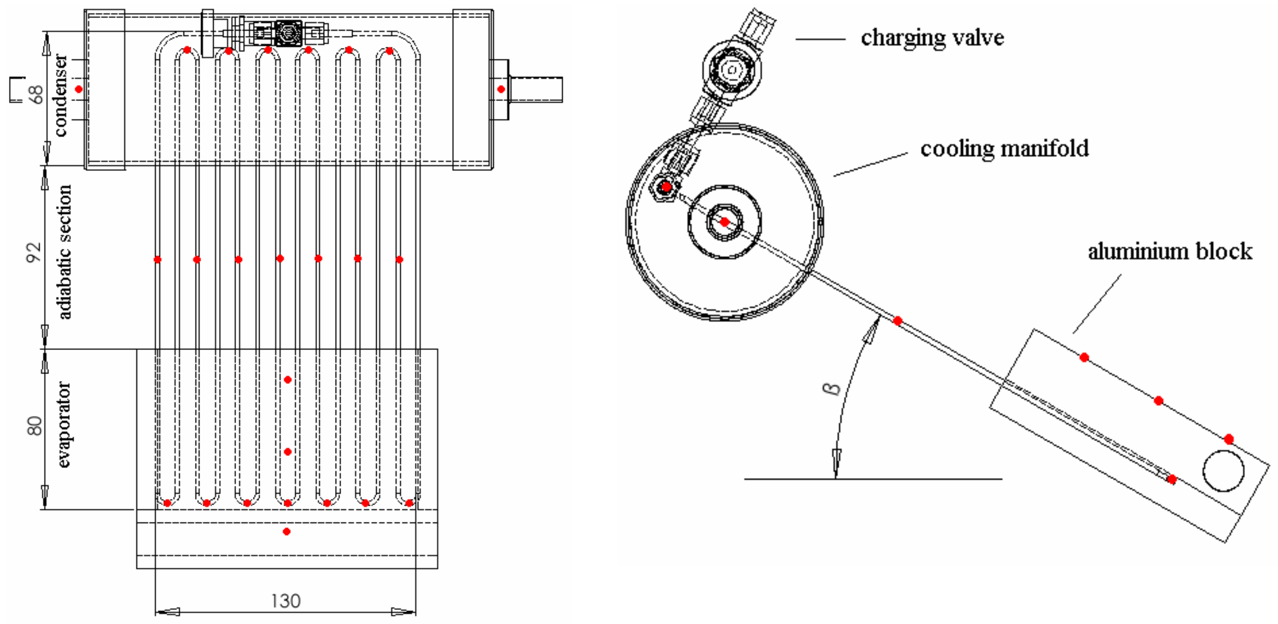

Figure 1. Experimental apparatus showing the thermocouple distribution (red dots).

Table 1. PHP Design Characteristics

number of turns	7
evaporator section lenght	80 mm
adiabatic section lenght	92 mm
condenser section lenght	68 mm
tube inner diameter	1.5 mm
tube external diameter	2.0 mm
PHP width	130 mm
total tube lenght	3.16 m
cooling bath fluid	ethanol
cooling temperature	-20°C

Temperatures were measured with copper-constantan thermocouples and a data acquisition system, with maximum error of 0.1ºC. The equipment list and variables tested are presented in Table 2 and Table 3.

Table 2. List of equipment

Equipment	Manufacturer	Model
data acquisition system	Agilent	34970A
DC power supply	Heinzinger	PTN 125-10
thermostatic bath	Lauda	RK 8 KP
needle valve	Swagelok	B-2JN

Table 3. Test parameters

Filling ratio (FR):	25%, 50%, 75%
Inclination angle (β):	-12.5º, 0º, 45º, 90º
Power level:	25W, 50W, 75W, 100W
Work fluid	CO_2

A proof body was used for the charging procedure because the complete experimental setup could not be weighted to the necessary accuracy of fractions of gram. The proof body was first evacuated and fully charged with CO_2. The charge was then released until it weighted exactly the amount necessary for each test filling ratio (FR) plus the calculated charge remain for the proof body total volume. The proof body was then connected to the PHP and the PHP was evacuated. The charge in the proof body was then transferred to the PHP, by cooling and gravity. The charging valve was closed and the mass was checked again.

The filling ratio is defined as the liquid volume at the mean operation temperature divided by the total internal volume. The inclination angle (β) was measured in relation to the horizontal, as shown in Figure 1.

3. RESULTS

As described in the first section of this paper, the PHP behavior was studied in terms of variation of input power, filling ration and inclination angle. At a given filling ratio and inclination angle, the heat input was changed after quasi-steady state was achieved for each position. In the present experiment all tests were interrupted when the evaporator mean temperature reached 5ºC or the last stabilization level under 5ºC. The maximum Bond number (Bo=1.9) does not exceed the critical value (Bo=2; Akachi et al., 1996) for the entire range of the evaporator average temperature (-20ºC to 5ºC). The Bond number (or alternatively the Eötvös number) is the ratio of gravity forces and surface tension and is defined as follows,

$$Bo = \sqrt{E\ddot{o}} = d_i \cdot \sqrt{\frac{g \cdot (\rho_l - \rho_v)}{\sigma}} \qquad (1)$$

Figures 2 to 4 show the transient variation of mean evaporator, condenser and adiabatic section temperatures, in respect to changes in the inclination angle and power input, for a filling ratio of 50%. For horizontal operation (Figure 2), a temperature oscillation process of great amplitude started when the input power reached 50W. These intermittent fluctuations continued until the test interruption, when the evaporator temperature reached 5ºC. For β=45° (Figure 3), the prominent temperature oscillation also was observed, but only with a power input of 75W. In the vertical operation mode (Figure 4), no intermittent fluctuation was observed until the last stabilization level was reached and the test interrupted. In the upper heater operation mode (i. e., β= -12.5° or smaller) the PHP did not work as expected for any input heat power and filling ratio, showing a behavior that suggests an evaporator dry-out. This behavior was already expected for a negative inclination, since the tested PHP has a small number of turns, not fulfilling the minimum requirements for anti-gravity operation, as explained in Khandekar and Groll (2006).

It can be noticed from the three previously cited figures that the average condenser temperature changes with the inclination angle, for the same power input. Both the external cooling liquid temperature and the flow rate were the same for all tests, so it is unlikely that the external convection heat transfer coefficient has changed. Therefore it can be inferred that the condenser wall temperature distribution is not uniform along the condenser section length, and changes for different operation conditions. Because the condenser temperature is always measured at the extreme internal curve (Figure 1) and temperature distribution is not uniform in all tests, different local temperatures are measured. These changes in temperature distribution could be related to differences in the internal flow pattern for each condition, such as oscillation frequency and boiling regime, which affect the local thermal resistance and therefore the wall temperature distribution. Additional tests are necessary to determine the very nature of these internal flow changes.

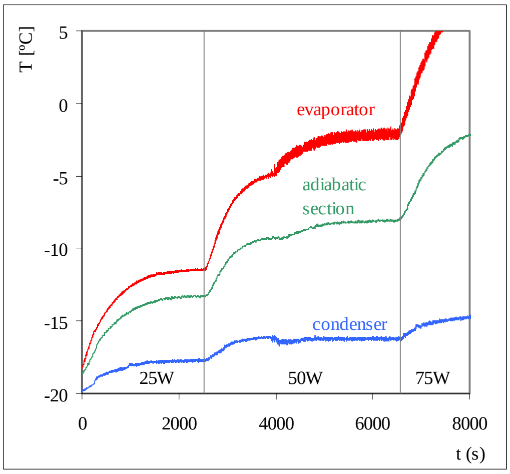

Figure 2. Transient variation of the average evaporator temperature for FR=50% and β=0°.

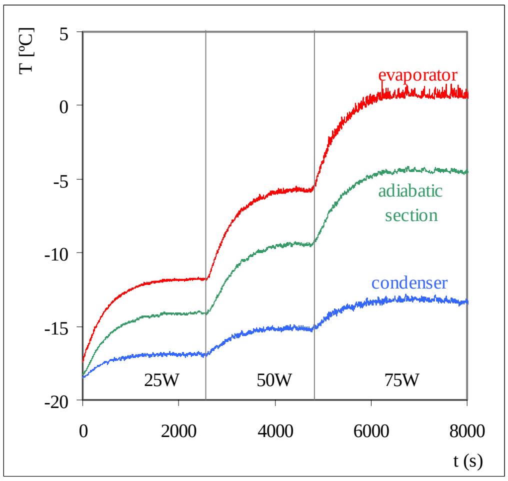

Figure 3. Transient variation of the average evaporator temperature for FR=50% and β=45°.

Table 4. Overall thermal resistance (K/W) as a function of the heat power, filling ratio and inclination angle.

	Based on the condenser temperature									Based on the cooling bath temperature								
FR β	**25% 0º**	**25% 45º**	**25% 90º**	**50% 0º**	**50% 45º**	**50% 90º**	**75% 0º**	**75% 45º**	**75% 90º**	**25% 0º**	**25% 45º**	**25% 90º**	**50% 0º**	**50% 45º**	**50% 90º**	**75% 0º**	**75% 45º**	**75% 90º**
25 W	0.309	0.154	0.233	0.250	0.226	0.242	0.321	0.142	0.238	0.509	0.439	0.376	0.341	0.362	0.508	0.417	0.450	0.346
50 W	dry-out	0.202	0.239	0.282	0.189	0.226	dry-out	0.135	0.223	dry-out	0.448	0.354	0.357	0.285	0.457	dry-out	0.405	0.312
75 W	dry-out	dry-out	dry-out	dry-out	0.187	limit	dry-out	limit	0.218	dry-out	dry-out	dry-out	dry-out	0.278	limit	dry-out	limit	0.302
100 W	dry-out	dry-out	dry-out	dry-out	limit	limit	dry-out	limit	limit	dry-out	dry-out	dry-out	dry-out	limit	limit	dry-out	limit	limit

dry-out: evaporator dry-out.
limit: temperature reached 5°C or the last stabilization level under 5°C.

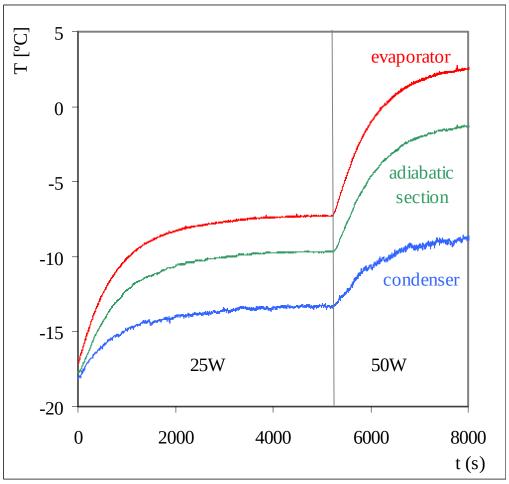

Figure 4. Transient variation of the average evaporator temperature for FR=50% and β=90°.

Table 4 shows the effect of input heat power, filling ratio and inclination angle on the overall thermal resistance of the PHP. The thermal resistance is calculated as the ratio between the temperature difference of the average evaporator temperature and the cooling manifold temperature (when based in the cooling bath temperature) or the average condenser temperature (when based in the average condenser wall temperature), and the input thermal power. The results show that the thermal resistance behavior based in the condenser wall temperature was different compared to the resistance based on the cooling bath temperature. This was expected according to the non-uniform condenser wall temperature distribution explained previously. In the first case the thermal resistance did not change much, despite variations in the inclination angle and input power. In the second case, the thermal resistance decreased with higher input power levels, for the filling ratios of 50% and 75%.

The CLPHP performed well for all tests with power input of 25W. However for power input of 50W, dry-out occurred in horizontal operation, for the filling ratios of 25% and 75%. The maximum heat flux achieved is 1.4W/cm^2 (based in the internal tube area in the evaporator section), corresponding to the power input of 75W. Analogous results are found in (Khandekar et al., 2003) for a PHP made of a copper tube of 2mm inner diameter and 5 turns, for water, ethanol and R123 as working fluids.

4. CONCLUSIONS

An experimental investigation of a CLPHP working between -20ºC and 5º C, with CO_2 as the working fluid was conducted. The performance was characterized in terms of overall thermal resistance based on the mean condenser wall temperature or the cooling bath temperature. The variables tested were the input evaporator power, filling ratio and the inclination angle in respect to the horizontal axis. It was verified that the CLPHP had adequate performance for all tests, up to the power level of 25W. However, for some inclination and filling ratio conditions dry-out eventually occurred for power inputs over this level.

Even though being promising, the obtained results show that a redesign of the PHP and additional tests are necessary. The number of turns should be increased in order to fulfill the requirements necessary for operation in anti-gravity mode (with negative inclination), in consequence also raising the overall heat transport capacity.

NOMENCLATURE

Bo	: Bond number
d	: tube diameter, m
Eö	: Eötvös number
G	: gravitational acceleration, m/s^2
T	: temperature (°C or K)
t	: time (s)

Greek Symbols

β	: inclination angle from horizontal axis (degrees)
ρ	: density, kg/m^3
σ	: surface tension, N/m

Subscripts

i	: inner
l	: liquid
v	: vapor

Abbreviations

CLPHP	: Closed Loop Pulsating Heat Pipe
FR	: Volumetric Filling Ratio (working fluid volume at mean operation temperature / total device internal volume)
PHP	: Pulsating Heat Pipe

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