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HEAT PIPES FOR HEAT RECOVERY SYSTEMS

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Abstract: *The paper presents the results of experimental research on two heat pipes, one wicked and the other wickless (thermosiphon). The conventional heat pipe is more efficient for small average temperature differences, but it blocks rather quickly. A calculus based on the pressure balance with the one dimensional heat pipe model leads to a good correlation with the experimental results. In contrast, the thermosiphon achieves significantly higher functional performances at higher temperature differences. So, the conventional heat pipes can be recommended to be used in low average temperature difference applications as air conditioning and thermosiphons are usable in heat recovery from industrial waste sources (air preheaters in boilers or ovens).*

Key words: *heat pipe, thermosiphon, waste heat recovery system.*

1. Introduction

The heat pipe is defined as a device that achieves an efficient heat transfer by combining in a closed loop the next phenomena: vaporization, vapour transport, condensation and condense return of a working fluid.

For space applications, the action of the liquid pumping, in order to be returned in the condenser is accomplished by the surface tension forces that are created in a special component of the heat pipe, called wick. The wick can be made of sintered porous material, grooves incised in the shell of the heat pipe, or complex structure using a combination of these materials and technologies. But in terrestrial applications, the wick, even if very efficient, can achieve only a very modest pumping of the liquid against the gravitational mass forces.

Thus, in case of terrestrial applications, gravitational mass forces are a very simple and effective means for returning condense to the vaporization zone, and the wickless heat pipes must operate only with the condensation zone above the evaporation zone [5]. They are known as having the names of thermosiphons although their operation does not fully correspond to the basic definition of the thermosiphon: “a system in which a coolant is circulated by convection caused by a difference in density between the hot and cold portions of the liquid” [6], [3].

The wickless heat pipes are also known as having somehow the inappropriate names

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of gravitational biphasic thermosiphons to differentiate them from the anti-gravitational thermosiphons (inverse thermal syphon, [4]) that use a gas-lift process to return the condense, where the condensation zone is below the evaporation zone. It is experimentally well demonstrated that the gravity-assisted wickless heat pipes have a pulsating operation in the field of low and medium temperatures, the frequency of pulsations being function of the angle of inclination and the average temperature difference between the heating carrier and the heated one.

The functional performance curve is the graphical representation of the maximum heat flux axially transferred by the heat pipe (Q [W]) as a function of the average temperature difference (Δt [K]). But, in the gravitational mass forces field the heat flux also depends on the angle of inclination. A preliminary determination of the optimum tilt angle is therefore required, positioning the heat pipe at this tilt angle, and then the functional performance curve can be accomplished.

In the study reported below, a comparison between a conventional heat pipe and a thermosiphon is made from the point of view of the heat transfer performances. Both have the same external geometric characteristics, but one of the two has a wick, and the other has a smooth inner wall. The heat transfer devices were subjected to the same experimental conditions of heat transfer between the heat carrier and the outer surface of the device, both in the evaporator zone and in the condenser zone.

2. Materials Methods and Equipment

2.1. The experimental equipment

The heat pipe is installed on a supporting frame which can be rotated around the horizontal axis between the horizontal position ($\alpha = 0°$) and the vertical position ($\alpha = 90°$) and which supports the two water heat exchangers, one for the evaporator zone and the other for the condenser zone, to which the tested heat pipe is connected (Figure 1). The two heat exchangers are very similar in size, being provided with water inlet/outlet connections and with spirals inserted between the heat pipe and their inner surface in order to increase the length of the water path and increase the turbulence.

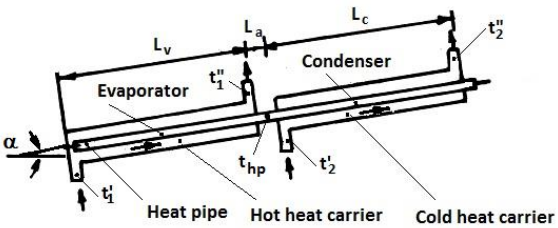

Fig. 1. H*eat pipe performance test rig*

The temperature measurement of the heat carriers was performed with thermometers having the precision of 0.1 [K], and that of the heat pipe (t_{hp} [°C]) was performed with a T-type thermocouple having the precision of 0.1 [K].

The heat source consisted of a 200 L thermally insulated container in which the water was electrically heated up to the temperature of about 95 °C, and the circulation of the hot heat carrier was ensured by a centrifugal pump. The cooling of the heat pipe was carried out by using water from the residential water network. In order to avoid pressure fluctuations in the network, which can affect the accuracy of the mass flow measurement and create the impossibility of establishing a steady state operating regime, a constant-level reservoir has been inserted in the circuit. The measurement of the mass flow rate of the cold water was performed by assessing the time interval in which a well-determined volume is filled.

2.2. The determination of the heat flux

By measuring the mass flow rate of the cold water and its temperatures at the inlet and outlet of the corresponding heat exchanger, the heat flux transferred within the heat pipe can be determined in all kinds of operating conditions:

$$Q = m_2 \cdot c \cdot (t_2'' - t_2') \text{ [W]}. \tag{1}$$

In calculations, as in graphical representations, the following notations were used: t_1'[°C] - the inlet temperature of the hot heat carrier; t_1''[°C] - the exit temperature of the hot heat carrier; t_2'[°C] - the inlet temperature of the cold heat carrier; t_2''[°C] - the outlet temperature of the cold heat carrier; $t_1 = (t_1' + t_1'')/2$ [°C] - the average temperature of the hot heat carrier; $t_2 = (t_2' + t_2'')/2$ [°C] - the average temperature of the cold heat carrier; $\Delta t = t_1 - t_2$ [K] - the average temperature difference; L_t [m] - the total length of the heat pipe; L_v [m] - the length of the vaporization zone of the heat pipe; L_c [m] - the length of the condensation zone of the heat pipe; L_a [m] - the length of the adiabatic (or transport) zone of the heat pipe; m_1 [kg/s] - the mass flow rate of the hot heat carrier; m_2 [kg/s] - the mass flow rate of the cold heat carrier; c [J/(kg·K] - the specific heat of water.

2.3. Heat pipes

For testing, two heat pipes with the same geometric characteristics: diameter, $\Phi 32 \times 2.5$ mm, L_t = 1.5 m; L_v = 0.7 m; L_c = 0.660 m; L_a = 0.105 m were manufactured and the extreme upper end of the heat pipe, on a length of 35 mm, was permanently in contact with air for periodic verification of the heat pipe operation. It is known that if this area is cold, it is obvious that the heat pipe is blocked. The material from which the heat pipes were built was stainless steel, and as working fluid acetone was used. This combination of materials is known to be compatible. One of the heat pipes is wickless, the filling ratio (the volume of the liquid working fluid relative to the heat pipe's total volume) was $U_t = 10.5\%$; thus a volume of 90 cm^3 of acetone was introduced in the heat pipe.

The heat pipe had on the inner wall a wick composed of a stainless steel layer of 50 mesh and the wire thickness of 0.15 mm and a stainless steel layer of 17 mesh and a thickness of the wire of 0.38 mm, having the role of maintaining the first layer in contact with the smooth inner wall of the heat pipe. As working fluid, acetone of the same quality, as in the case of the first heat pipe, was used, but with a volume of 140 cm^3, sufficient to saturate the wick, whose volume was 67 cm^3.

2.4. Test procedure

The flow rates and temperatures of the heat carriers are adjusted in order to obtain a steady state operating regime and afterwards, the temperatures of the heat carriers, the heat pipe wall temperature in the adiabatic section and the cold water mass flow rate are measured. Thus, a certain operating regime is established, which is characterized by the following quantities, heat flux, Q [W], temperature difference, Δt [K] and the inclination angle with respect to the horizontal plane, α [°].

Previous research [1] done on gravitational wickless heat pipes has shown that there is a strong influence of the degree of filling and the angle of inclination on the heat flux transferred axially by the heat pipe. Therefore, the gravity assisted wickless heat pipe measurements were accomplished at an inclination angle of 40°, for a temperature range from 60°C to 90°C of the hot heat carrier, values that were obtained from previous research [1], [2]. When the temperature of $t_1' = 78.5\,°C$ was reached, the functional performance of this heat pipe was obtained, if the inclination angle ranged from 0° (horizontal position) to 90° (vertical position).

The gravity assisted wicked heat pipe was tested at an inclination angle of 40° relative to the horizontal plane, with the evaporator section below the condenser section, so that the gravitational mass forces helped the condense to return to the evaporation section.

It is important to mention that the mass flow rates of the hot and cold water were kept almost constant, so that the fluids' velocities were constant, ensuring a similar thermal transfer regime at the interfaces between the heat pipe and the heat carriers.

3. Results and Discussion

The results of the experiments are graphically presented in Figures 2, 3 and 4.

Figure 2 shows the variation of the axial heat flux Q [kW] with respect to the inclination angle, relative to the horizontal plane α [°] for the thermosiphon at a hot heat carrier average temperature of around $t_1 = 76.0\,°C$.

It can be observed that the axial heat flux is very small for the horizontal position of the thermosiphon, but it increases immediately up to the maximum value, even from an inclination angle relative to the horizontal plane of $\alpha = 5°$. Also, it can be observed that the axial heat transfer rate of thermosiphon is around the maximum for angles between $\alpha = 5°$ and $\alpha = 90°$.

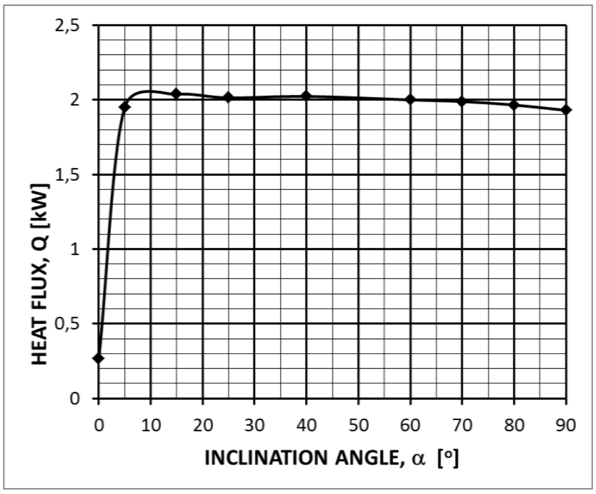

Fig. 2. *Heat flux versus the thermosiphon inclination angle*

It can be noticed that the maximum heat flow rate Q = 2.022 [kW] was obtained for $\Delta t = 61.8\ \text{K}$ and the optimum inclination angle, $\alpha = 40^\circ$. These results, which were expected to be obtained, conducted to the idea of raising the functional performance curves of these heat pipes at this optimum inclination angle.

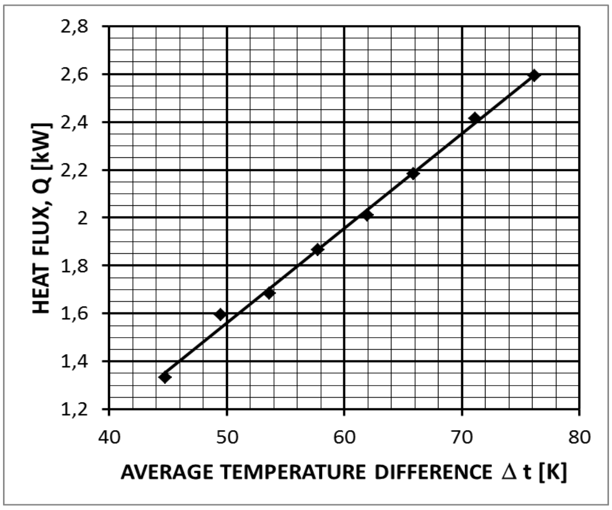

Fig. 3. *Functional performance curve of the gravity assisted wickless heat pipe (thermosiphon)*

Figure 3 shows the functional performance curve of the gravity assisted wickless heat pipe. The heat pipe worked according to the previous tests, the functional performance curve being a straight line. It did not block in the temperature range at which it was tested, the maximum heat flux being Q = 2.592 [kW] at $\Delta t = 76.16\ ^\circ\text{C}$ and $t_1 = 91.35^\circ\text{C}$.

Figure 4 shows, for comparison, the functional performance curves of the two heat pipes. The superiority of the wicked heat pipe at low temperature differences is obvious, even if for short term; instead a blockage occurs at a temperature difference Δt = 51.0 K, the average temperature of the hot heat carrier at the evaporator section being approx. $t_1 = 65.0°C$. The maximum axial heat flux obtained by this heat pipe is very small: Q = 1.80 [kW] at $\Delta t = 50.08\,K$.

The question is which is the cause of the blockage that has occurred in this heat pipe. First, the sonic limit is excluded because it could have been produced at the thermosiphon. The capillary limit is reached if the wick, that has a poor permeability, cannot carry the fluid flow required by the average temperature difference. But from the construction, the amount of liquid introduced into the heat pipe exceeded the volume of the wick, so the heat pipe could have worked without problems because the excess dosed liquid returns above the wick due to the field of gravitational mass forces.

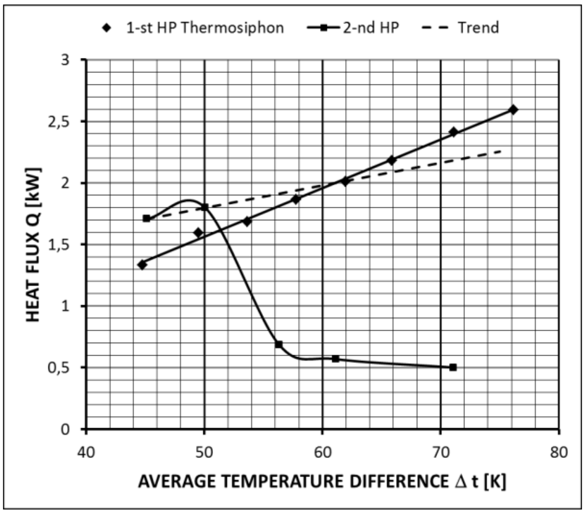

Fig. 4. *Functional performance curves of the two heat pipes*

The entrainment limit occurs when the vapor stream, having a high flow rate and therefore high velocity, exerts shear forces on the liquid in the wick flowing in the opposite direction. The shear forces slow or even block the flow of the liquid. Again, due to the extra volume of working fluid introduced into the experimental heat pipe this limit is unlikely to have occurred.

Most likely in the experimental heat pipe the limitation occurred by reaching the critical heat flux in the wick. By increasing the average temperature difference the flow of working fluid increases (liquid and vapor). Through the interface between the inner surface of the heat pipe and the liquid in the wick, the thermal flux density greatly increases and the nucleate boiling increases as the bubble quantity becomes so high that the liquid can't reach the inner surface of the heat pipe container in the evaporator section. Thus, in this zone of the evaporator section a continuous vapour film is formed in the wick. The boiling produced in the wick immediately leads to the overheating of the heat pipe wall in the evaporation zone and the phenomenon is known as burnout, boiling crisis or critical heat flux of the heat pipe.

Starting from the fact that from all the experiences performed all the functional performance curves have a linear shape, in Figure 3 the tendency by extrapolation of the functional performance curve of the wicked heat pipe was represented, if the blockage for the difference did not occur at an average temperature of about 51 ^{o}C. It was noticed an intersection with the other curve at an average temperature difference of 62 ^{o}C. Above this value, the performance of the heat pipe is below the performance of the thermosiphon. A more efficient wick would have had a functional performance curve above it. But the problem of blockage remains at higher average temperature differences.

4. Analytical Study

For a theoretical determination of the heat flow transferred by the wicked heat pipe, the simple relation of the pressure balance can be used [4]: the capillary pressure is equal to the sum of the pressure drop in the liquid, the pressure drop in the vapour and the hydrostatic pressure against the gravity mass forces. In this case gravity helps the flow, so the last term is of the opposite sign and the pressure drop when the vapor flow can be neglected:

$$\frac{2\sigma_l \cos\theta}{r_c} = \nu_l \cdot \frac{Q}{r} \cdot \frac{L_{ef}}{A_w \cdot K} - \rho_l g l_{ef} \sin\alpha \ \text{[Pa]}, \tag{2}$$

wherein: $\sigma_l\,[\mathrm{N/m}]$ - surface tension of the liquid; θ [rad]; [^{0}] - contact angle of the meniscus in the capillary pore; $r_c\,[\mathrm{m}]$ - capillary pore radius; $\nu_l\,[\mathrm{m^2/s}]$ - kinematic viscosity of the liquid; $A_w\,[\mathrm{m^2}]$ - wick cross sectional area; - $g\,[\mathrm{m/s^2}]$ gravitational acceleration; α [rad];[°] inclination angle relative to the horizontal plane ; $L_{ef}\,[\mathrm{m}]$ - total effective length for liquid flow in the heat pipe:

$$L_{ef} = \frac{L_v + L_c}{2} + L_a \ \text{[m]}; \tag{3}$$

K [m^2] - wick permeability obtained with the Blake - Koseny Equation [4]:

$$K = \frac{d_w^2 (1-\varepsilon)^3}{66.6 \cdot \varepsilon^2} \ [\mathrm{m^2}], \tag{4}$$

in which: d_w [m] is wire diameter and ε [–] - volume of the solid to the overall volume of the wick ratio.

It results the heat flux:

$$Q = \frac{K \cdot A_w \cdot r}{\nu_l \cdot l_{ef}} \left(\frac{2\sigma_l}{r_c} \cdot \cos\theta + \rho_l \cdot g \cdot l_{ef} \cdot \sin\alpha \right) \ \text{[W]}. \tag{5}$$

For the heat pipe there are obtained: $K = 2{,}568 \cdot 10^{-9}$ [m^2] and Q = 1813 [W], which accurately corresponds with the experimental result. It has been noticed that in the relation (5) the second term in parentheses is too large than the first one, demonstrating that the wick has a very low capillary height, however it has a high permeability which, together with the gravitational forces, contributed very much to achieve this performance.

5. Conclusions

The experimental study of two gravity assisted heat pipes one wicked and the other wickless (thermosiphon) showed that at small temperature differences the heat pipe is more efficient. But at large temperature differences, they are blocked; in contrast, the thermosiphon reach significantly higher functional performances. Thus, in the applications of heat recovery from residual sources using heat pipes [2] with working fluids suitable to be used at medium temperatures range (such as organic fluids), thermosiphons are recommended at high average temperature differences (boilers or hot water boilers, industrial ovens) and the conventional heat pipes are recommended at small temperature differences such as be in the field of HVAC.

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