Experimental research on cooling the condensation area to flat micro heat pipe using the inverted Seebeck effect

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ABSTRACT

This paper presents the experimental results obtained by cooling the condensation zone of a flat micro heat pipe (FMHP) using the inverted Seebeck effect (Peltier mode). In all cases of heat pipes, the working liquid must flow when there is a temperature difference between the evaporation zone and the condensation zone. The condensing area is cooled by blowing air through a radiator, which uses a Peltier module with a power of 50W as a cooler. At low temperatures, the vapor pressure of the liquid in the evaporator is very low. As the condenser pressure cannot be less than zero, the difference in vapor pressure is insufficient to exceed the viscous and gravitational forces, thus preventing the operation of FMHP at the heat transfer limit. The condenser is cooled after it reaches a thermal equilibrium. It is desired to highlight the temperature variations on the condensation area at the time of cooling air, as well as the temperature variation on the radiator of the Peltier module. The analysis starts from the theoretical component of cooling the condensing zone and compares the results obtained from the experimental research with those obtained by calculations. The research shows the temperatures recorded in the cooled area of the condenser, using forced cooling with air blowing using a Peltier module, powered by a direct current source. It also highlights the experimental contribution to the capture of the phenomenon of forced convection, possibly used for cases where FMHP should work at temperatures close to the occurrence of thermal blockage. The temperature values were taken by means of thermocouples mounted in thermal contact on the FMHP surface and are recorded by a data-logger in order to draw the graphs of temperature variation. This highlights the possibility of transporting high heat fluxes through FMHP, other than if it is conventionally cooled. For the calculations, the Mathcad programming environment will be used to plot the graphs of temperature variation on the condensation zone at forced cooling.

Keywords: flat micro heat pipe, biphasic transformation, inverted seebeck effect

1. THE HEAT TRANSFER IN THE CONDENSATION AREA AT FMHP

The study of heat transfer in the condensation zone highlights how the working liquid inside the FMHP passes from the vapor phase to the liquid phase. If we refer to the surface of the working liquid inside the FMHP, during evaporation liquid molecules leave the surface in a continuous flow. If the liquid is in thermodynamic equilibrium with the vapors above its surface, then the flow of molecules will return to the liquid with no loss or gain of mass. However, when a surface of liquid loses mass by evaporation clearly the pressure and temperature of the liquid's vapors must be less than the equilibrium value. The same principle applies to the condensation zone. In order to achieve a total condensation of liquid vapors in the condenser, the vapor pressure and temperature must be higher than the equilibrium value. Condensation occurs when vapors resulting from vaporization in the vaporization zone meet the cooled surfaces of the condenser. The heat transported by the vapor to the condenser is thus transferred, and through it is dissipated in the ambient environment. The greater the cooling of the condensation zone, the more heat flux can be transported through the FMHP. The value of the temperature drop can be estimated as follows: first the vapor parameters near the liquid-vapor interface are considered.

The average speed $\dot{v}_{va}$, of the vapors at temperature T_{va} and M molecular mass of liquid vapor, according to the kinetic theory, the average speed of the vapors can be written as [1],[2],[3],[4]:

$$\dot{v}_{va} = \sqrt{\frac{8k_B T_{va}}{\pi M}}, \tag{1.1}$$

It was noted with k_B – Boltzmann's constant. The average flow of liquid vapor molecules $\dot{\varphi}_{va}$ in every direction is given by:

$$\dot{\varphi}_{va} = \frac{n\dot{v}_{va}}{4} \cdot S^{-1}. \tag{1.2}$$

In this way, the heat flux corresponding to the average vapor flux can be expressed, respectively:

$$\dot{Q} = \frac{M\lambda_{va} n\dot{v}_{va}}{4} \cdot S^{-1}, \tag{1.3}$$

where n is the number of molecules of liquid vapor per unit volume and λ_{va} is the latent heat of vaporization. If it is considered that the liquid vapor inside the FMHP behaves like a perfect gas, then the vapor pressure can be written as:

$$P_{va} = nK_B T_{va}. \tag{1.4}$$

Therefore, the density of heat flux at the surface of the liquid in the vapor layer is:

$$\dot{q}_{va} = P_{va}\lambda_{va}\sqrt{\frac{M}{2\pi k_B T_{va}}}, \tag{1.5}$$

respectively the heat flux density for the working liquid:

$$\dot{q}_l = P_l\lambda_{va}\sqrt{\frac{M}{2\pi k_B T_l}}. \tag{1.6}$$

The liquid vapors will condense in the condenser and thus drops of liquid will appear which, by decreasing the temperature, leads to a decrease of the thermal resistance. Condensation can occur in two forms: either condensing vapors that form a continuous liquid surface or forming a large number of droplets. The condensing capacity is seriously affected by the presence of non-condensable vapors in the condenser. The phenomenon can occur only in the case of large heat fluxes in short time intervals. However, in FMHP the production of vapors is produced by the applied external heat flux, thus producing a concentration of vapors on the end of the condenser. This part of the condenser will stop condensing and external intervention is required to excessively cool the area. If the vapor temperature is considered to be the same as that of the liquid, respectively $T_{va}=T_l$, the result is that the temperature on the entire surface will be the same (T_s), therefore, the density of the thermal flow will be:

$$\dot{q} = (P_l - P_{va})\lambda_{va}\sqrt{\frac{M}{2\pi k_B T_s}}. \tag{1.7}$$

The problem of calculating the vapor pressure drop by radial flow due to evaporation or condensation in the evaporation and condensation regions is complicated. It is convenient to further define the Reynolds number, the radial Reynolds number:

$$\mathrm{Re}_{rd} = \frac{\rho_{va} v_{rd} r_{va}}{\mu_{va}}. \tag{1.8}$$

The convention dictates that the radius of the vapor space r_{va} is used rather than the diameter of the vapor space that is commonly used in the definition of the radial Reynolds number. Re_{rd} is positive in the vaporization zone and negative in the condensation zone. In most micro heat pipes Re_{rd} number is between the interval 0.1÷100. If the heat flux density is to be related to the radial velocity of the liquid vapor, then: $\dot{q} = \rho_{va}\lambda_{va}v_{rd}$. The pressure difference can thus be calculated from the above equations P_l-P_{va} resulting:

$$(P_l - P_{va}) = \mathrm{Re}_{rd}\ \mu_{va}\left[r_{va}\left(\frac{M}{2\pi k_B T_s}\right)^{1/2}\right]^{-1}. \tag{1.9}$$

At high heat fluxes, the velocity of liquid vapor inside the FMHP increases. If this speed is high enough to drive the liquid that returns to the vaporizer, then the performance of the FMHP in the heat transfer capacity will be declining. The radial heat flow in the condenser is accompanied by a temperature difference that is relatively small. When the critical value of the heat flux on the evaporator is reached, the liquid vapors cover the entire surface of the evaporator resulting in an excessive temperature difference between the evaporator and the condenser. This problem can be overcome by constructive solutions by incorporating a short capillary layer contained in the vaporization zone and separating the flow of vapors reaching the condenser from the return of the liquid through the inner capillary layer. Another solution that does not aim at constructive modifications of the FMHP, is the forced cooling of the condensation zone by additional cooled air.

2. THE SHAPING OF THE HEAT TRANSFER IN THE CONDENSATION AREA

For FMHP to work properly, all limitations must be investigated given the resulting heat transfer capacities. If the analysis is done according to a limit, unexpected operating situations may occur and FMHP may not behave correctly under different conditions of use. This can cause syncope in the heat transport (e.g. computer CPU) because the transport capacity is not enough to cool the heat source. Different combinations of parameters can affect FMHP performance. But in order to see the individual effects of these factors, other parameters than those selected are kept constant. Investigating the effect of the internal vapor pressure of the FMHP liquid on the heat transport capacity, respectively of the temperature in the condensation area, other parameters such as: FMHP length, pipe radius, physio-chemical parameters of the internal working fluid and internal capillary structure by the degree of permeability, they are kept constant. FMHP used in the analysis have the physical properties presented in Table 1. For the working liquid we opted for demineralized water and for the calculations we used the physio-chemical properties of distilled water.

Table 1. Functional parameters of FMHP

Parameter	**Value**

FMHP radius	0.003 [m]
Length of the condensation area	0.05 [m]
FMHP wall thickness	0.0005 [m]
Initial internal pressure	$2.026x10^4$ [Pa]
Reynolds	1÷80

The FMHP operating configuration is formed using these properties. The temperature scale is chosen for demineralized water as a working liquid and is in the range T_{ini} = 22°C and T_{fin}=95°C, thus covering the operating range of the working liquid between ambient temperature and critical temperature (boiling temperature). However, this does not mean that the selected working fluid will show a similar behavior throughout this whole temperature range. Therefore, the useful operating range of the FMHP is generally less than the entire temperature range chosen for the working fluid. For distilled water it was considered that the saturation pressure at *0°C* is *611Pa* and for the critical one the temperature of *95°C* is *101.4KPa*. In Figure 1 is presented the graph of the variation of the pressure difference in the condenser between that of the inner liquid and that of the liquid vapors for Re_{rd} according to Table 1.

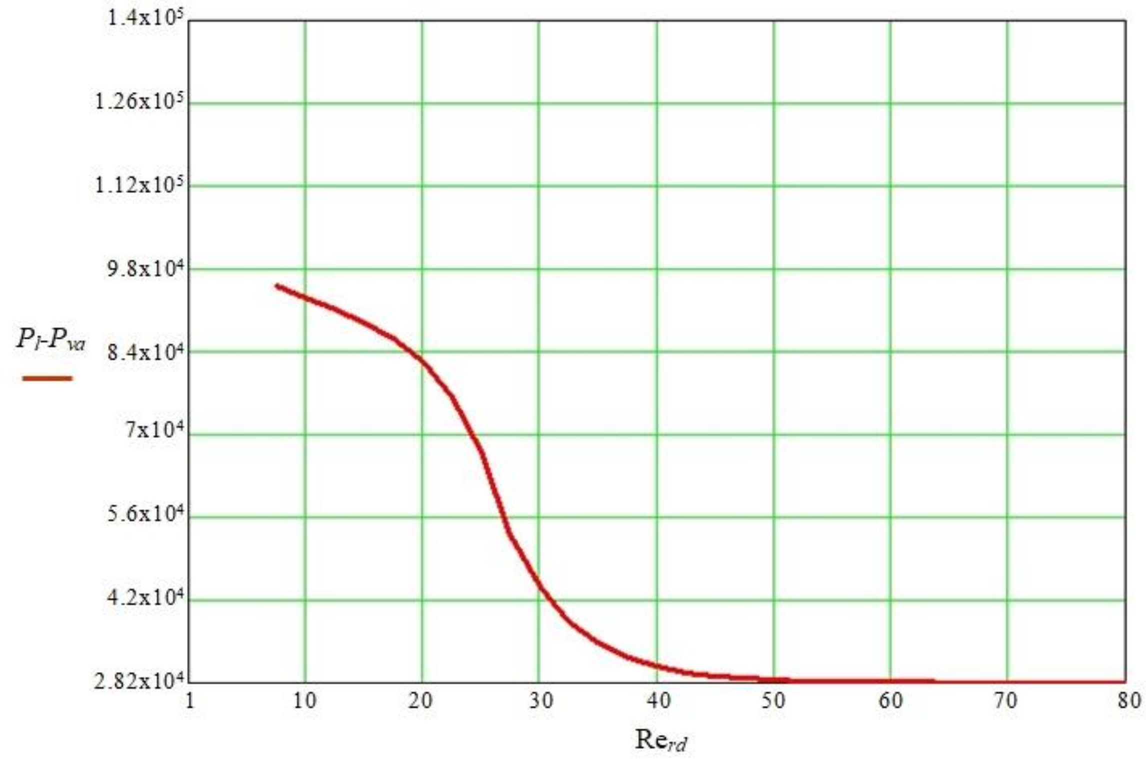

Figure 1. The variation graph of P_l-P_{va} according to Re_{rd} la T_l=45°C

The graph in Figure 1 shows the variation of the pressure difference P_l-P_{va} on the condensation area, depending on Re_{rd} variation, when the condensation zone reaches the temperature of *45°C*. The pressure difference has a decrease for Re_{rd} between the values 8÷50 after which the pressure does not decrease anymore, remaining constant at the value of $2.82 \cdot 10^4 Pa$. When the temperature on the condensation zone rises to *85°C*, the value of the pressure difference between liquid and vapors increases. The functional blocking limit on heat transport is reached. The graph in Figure 2 is drawn in the same coordinates as in Figure 1, respectively the variation graph P_l-P_{va}.

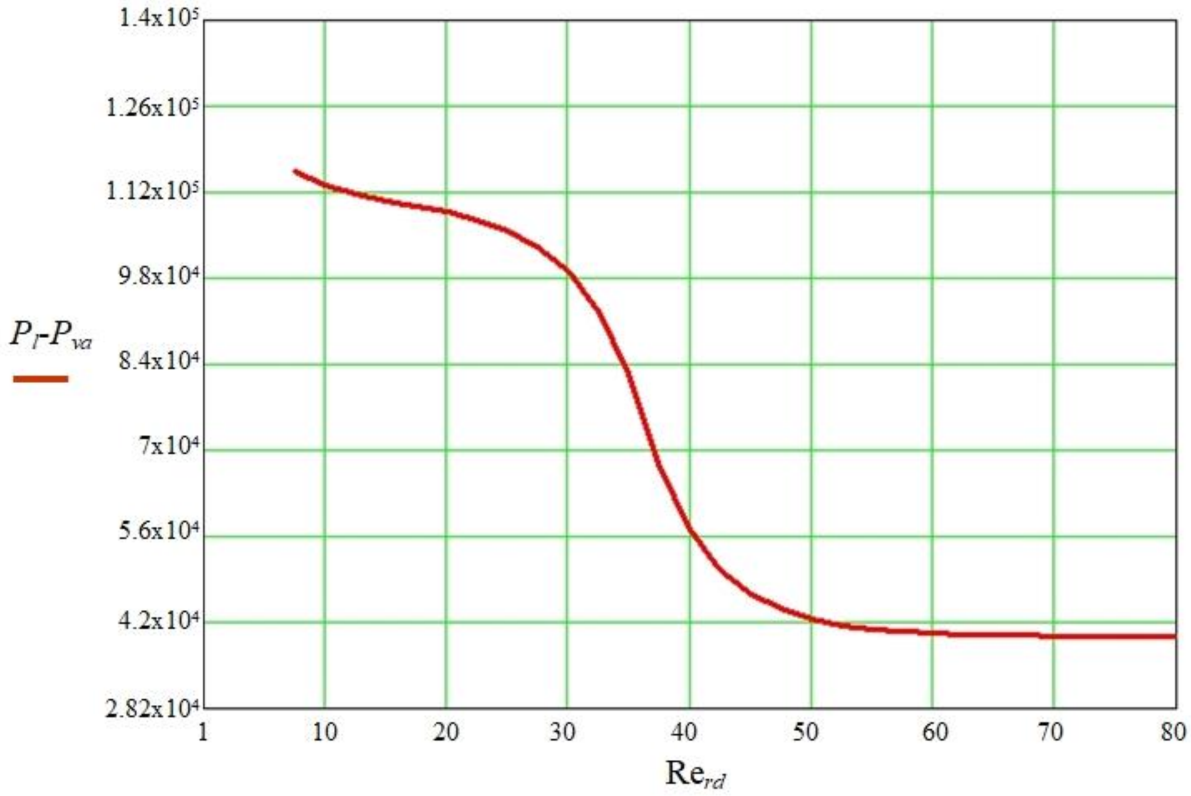

Figure 2. The variation graph of P_l-P_{va} depending on Re_{rd} la T_l=85°C

It can be seen that for Reynolds values between *8÷30* (Figure 2) pressure difference variation P_l-P_{va} is really small, this decreases from $1.2 \cdot 10^5 Pa$ to $9.8 \cdot 10^4 Pa$. With the addition of the excess cooling with the air cooled by the inverted seebeck effect there is a sudden decrease in the pressure difference inside the FMHP. This decreases on the interval $30 \div 70$ Re_{rd} from $9.8 \cdot 10^4 Pa$ to $3.81 \cdot 10^4 Pa$. This returns to internal pressure values that do not affect the return of the liquid through the inner capillary layer. The functioning of FMHP improves once the thermal transport capacity increases.

3. EXPERIMENTAL RESULTS

In order to carry out the experimental determinations regarding the cooling of the condensation zone at a flat micro heat pipe, an experimental stand will be made comprising the elements presented in Figure 3. A constant thermal flux is applied on the vaporization zone of FMHP at $t=1$. Heating the vaporization zone produces a vaporization of the liquid inside the FMHP. The movement of vapors through the FMHP will cause the condensation area to heat up [5]. Experimental determinations will highlight the change in temperature on the FMHP surface. The temperature variation will be monitored by mounting 9 thermocouples that will record temperature variations in the following areas: two thermocouples will be mounted on the FMHP vaporization zone, three thermocouples on the adiabatic zone, three thermocouples on the condensation zone and one thermocouple on the radiator mounted on the cold part of the peltier module [6].

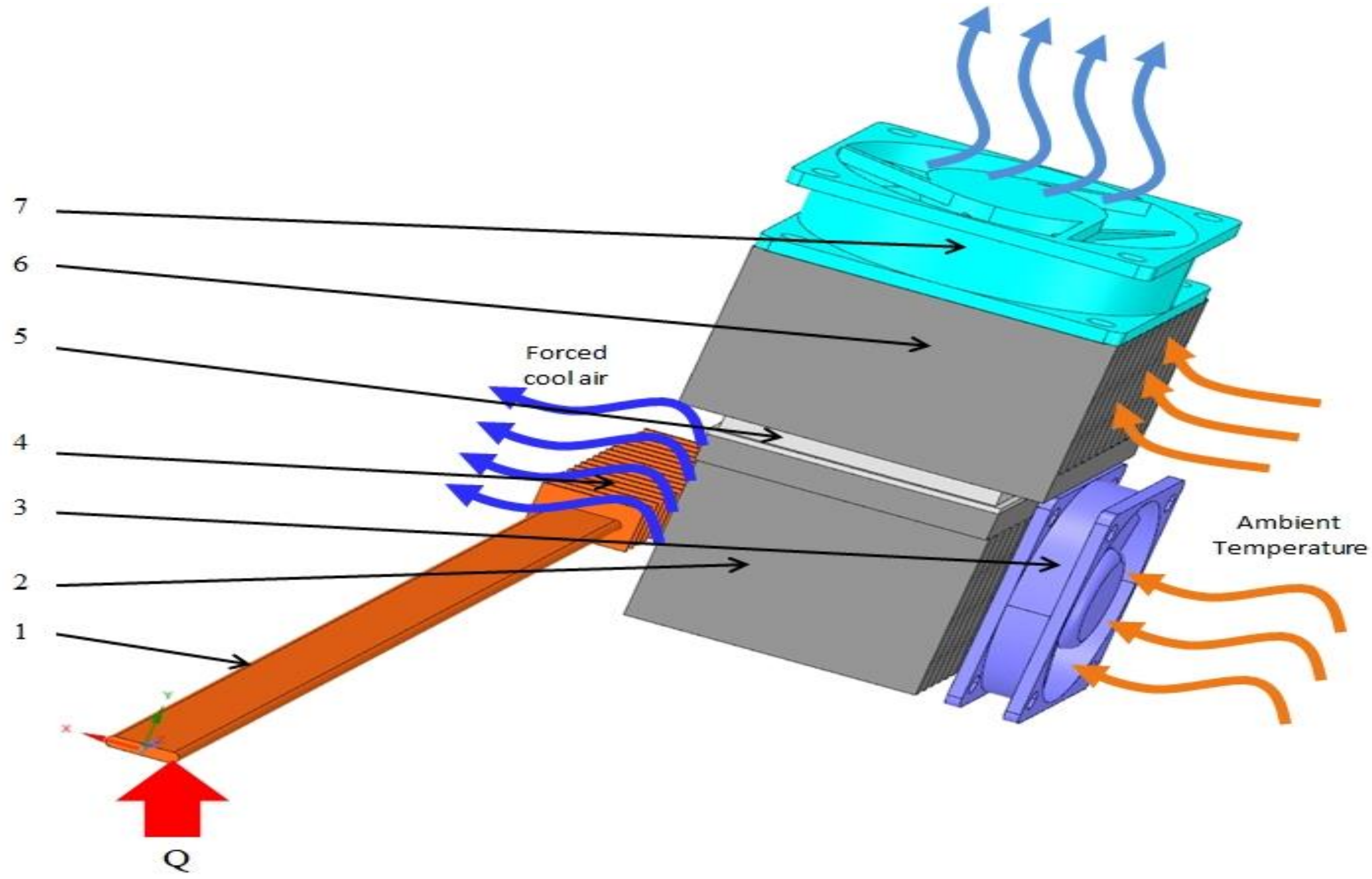

Figure 3. Cooling of the condensation zone by inverted Seebeck effect

FMHP, 2 – radiator mounted on the cold side of the peltier module, 3 – fan mounted on the radiator on the cold side of the peltier module, 4 – radiator mounted on the condensing area at the flat micro heat pipe, 5 – peltier module, 6 – radiator mounted on the hot side of the peltier module, 7 – fan for forced heat removal from the radiator mounted on the warm side of the peltier module.

As can be seen in Figure 3, the peltier module used for forced cooling of the vaporization zone, has mounted on the cold side a radiator and a fan. On the warm side of the module is also mounted a radiator and a fan that have the role of dissipating the produced heat into the environment. The heat dissipation produced by the hot part of the peltier module has an important role in lowering the temperature as much as possible on the cold side. The experimental results obtained highlight the temperature variations on the areas of the flat micro heat pipe as well as on the radiator mounted on the cold side of the peltier module. The data obtained were represented graphically in Figure 4. As it can be seen in the graph, the thermocouple mounted on the radiator of the cold part of the peltier module up to $t_2=131s$, records an average temperature of $T_1=26{,}5°C$. In the meantime, the peltier module is not charged, the temperature of the monitored radiator having the ambient temperature. The other 8 curves on the graph capture the temperature variation on the FMHP areas from $t_1=11s$ since the heating of the vaporization zone began. Temperatures in all monitored areas are constantly rising. At $t_2=131s$ the power supply to the peltier module starts, but the fan on the radiator mounted on the cold side remains off. The cooling of the condensing zone of the FMHP is carried out during this time by normal convection through its own radiator. At $t_3=290s$ on the radiator on the cold side of the module it reaches the temperature of $T_2=2{,}3°C$. Between $t_3=290s$ and $t_4=351s$ the thermal equilibrium of the radiator of the cold part is reached. Still at $t_4=351s$ the thermal balance on the FMHP areas, on the vaporization zone, is also reached $T_3=96{,}5°C$.

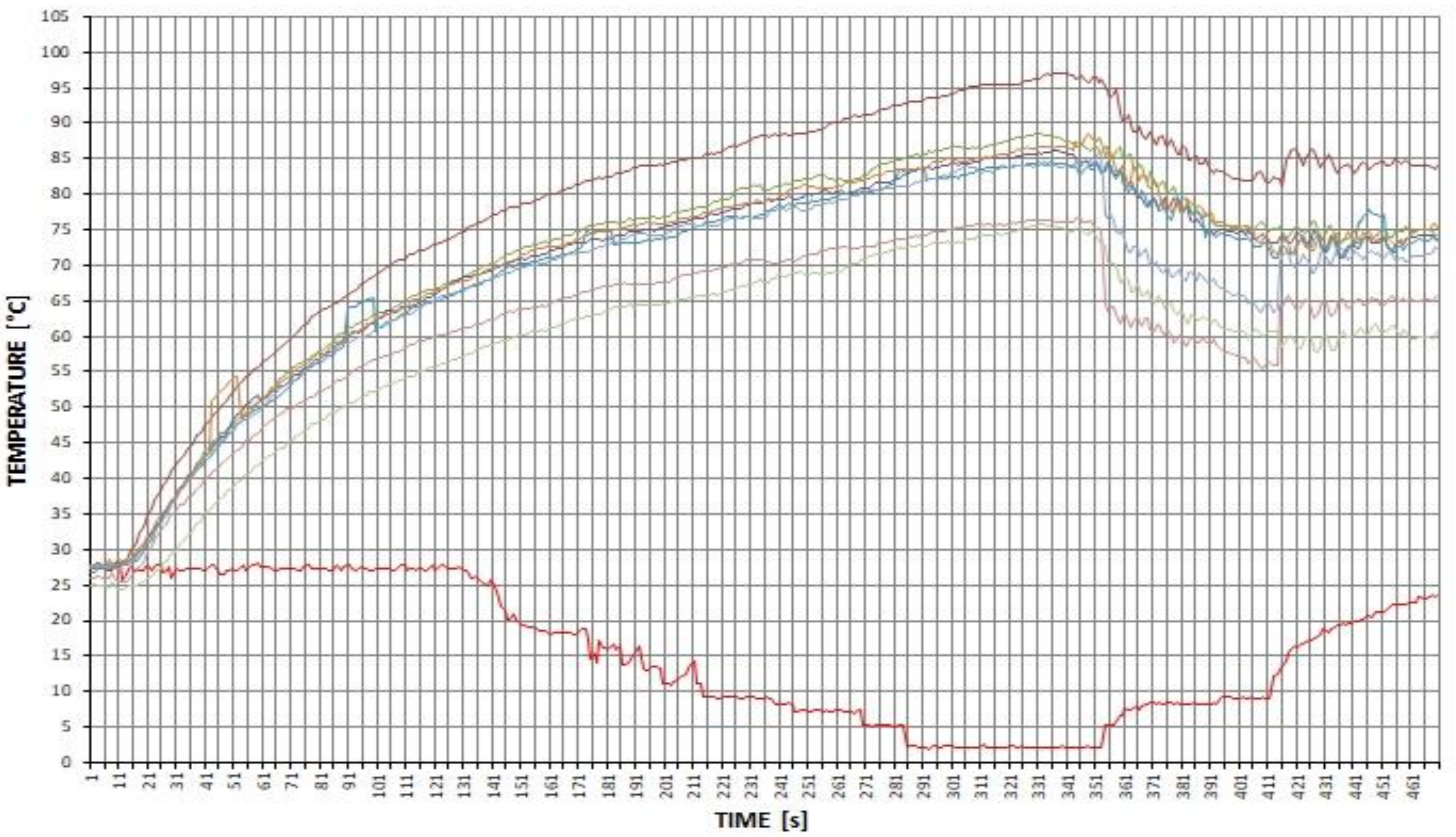

Figure 4. The temperature variation on the three areas of the FMHP and on the radiator cooled through the inverted seebeck effect.

After reaching the thermal equilibrium, start the fan on the radiator mounted on the cold side of the peltier module at $t_4=351s$. From the graph shown in Figure 4, there is a slight increase in temperature on the cold side of the radiator, and up to t_5=410s, thermal equilibrium is reached again at $T_4=8,45°C$. A sudden drop in temperature from the FMHP condensation zone occurs from $T_5=75,5°C$ to $T_6=64,8°C$. After stopping powering the peltier module at $t_5=410s$ it is observed that the temperature of the radiator of the cold part as well as on the condensation zone of FMHP increase.

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"ACKNOWLEDGMENT This paper has been financially supported within the project entitled „DECIDE - Development through entrepreneurial education and innovative doctoral and postdoctoral research, project code POCU / 380/6/13/125031, project co-financed from the European Social Fund through the 2014 – 2020 Operational Program Human Capital"