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# Experimental analysis of gravitational heat pipe using R507 as a working fluid

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ABSTRACT

The paper presents possibility of using R507 as a working fluid in gravitational heat pipe. The outcome of experimental analysis are compared to the results obtained with mathematical model. The simulations allow to calculate fluid and wall temperatures in the thermosyphon. The model equations contain the formulas for boiling and condensing heat transfer coefficients. Proposed new model is in good agreement with real operation of the device, especially at the steady state. There are discrepancies during the start-up process, but the differences between the results obtained by the model and experiments do not exceed 5% at the steady state. The thermal efficiency of the thermosyphon is also determined as 0.63.

**KEYWORDS:** *thermosyphon, mathematical model, heat transfer*

## 1. INTRODUCTION

Thermosyphon consists of three basic components: the evaporator, adiabatic section and the condenser. The heat supplied to the evaporator causes evaporation of the liquid collected at the bottom of the device. The vapor flows through the adiabatic section to the condenser where phase change takes place. Condensed liquid comes back to the evaporator as a consequence of gravitational forces. There is no moving parts what increase the reliability of the installation. Due to high heat transfer coefficient of both processes-boiling and condensation, heat pipes can transfer heat very efficiently. Heat conduction coefficient for 1 m long heat pipe with the heat flux supplied to the evaporator in a range from 3 to 12 $\mathrm{kW/m^2}$ can be as high as 700–1500 W/m·K depending on the type of the working fluid. The impact of the refrigerant is important, so the selection should be made carefully. To provide proper operation of the device, heat pipe must work continuously. To investigate heat pipe thermal performance, its maximum heat transport capacity (the maximum heat applied to the evaporator which does not cause the dry-out characterized by intense increase in wall temperature) and temperature distribution in heat pipe during the whole operation of the device should be determined. It can be done empirically, but it is very often too labor/time consuming and troublesome. To obtain experimental data for just one heat pipe, 1.5–2 h are needed [1]. That is why creation of suitable mathematical model can be better solution.

## 2. Mathematical model

The processes occurring in the heat pipes were studied empirically and theoretically by many researchers. The first model describing operation of this kind of devices was proposed among other things by [2], [3]. There was no available data, so this model was not validated empirically. Tsai et al. [1] proposed model based on the energy equations which divide the device into three control sectors: evaporator wall with wick structure, condenser wall with wick structure and the working fluid. It was assumed that the refrigerant is saturated and there is no heat conduction between particular sections. Heat transfer coefficient were determined empirically. Hamidreza et al. [4] proposed two-dimensional numerical model to simulate thermosyphon operation during the start-up depending on the filling ratio. Balance equations of energy, mass and momentum were solved using the finite volume method. The validation of the model was based on the experimental data available in the literature.

The dynamic, differential model presented by Farsi et al. [5] was adopted to examine the working parameters of the gravitational heat pipe filled with different refrigerants. The model divide the device into two parts: the working fluid and the wall of evaporator. To determine wall temperature in the condenser section third equation was added. Figure 1 presents the schema of the pipe used for both, the model and the experiment. The following set of equation was used in the simulation:

$$C_w \frac{dT_{w_e}}{dt} = Q_e - h_e A_e (T_{w_e} - T_f) \tag{1}$$

$$C_f \frac{dT_f}{dt} = h_e A_e (T_{w_e} - T_f) - h_c A_c (T_f - T_{wat}) \tag{2}$$

$$C_w \frac{dT_{w_c}}{dt} = h_c A_c (T_f - T_{w_c}) - Q_c \tag{3}$$

where:

- $C$ – heat capacity of the wall and the fluid, J/K;
- $T$ – temperature, K;
- $Q_e$ – heat transfer to the evaporator, W;
- $h$ – heat transfer coefficient, W/m$^2$K;
- $A$ – heat transfer area, m$^2$.

Index:

- $w$ – wall of the evaporator;
- $f$ – working fluid;
- $e$ – evaporation section;
- $c$ – condenser section;
- $wat$ – cooling water.

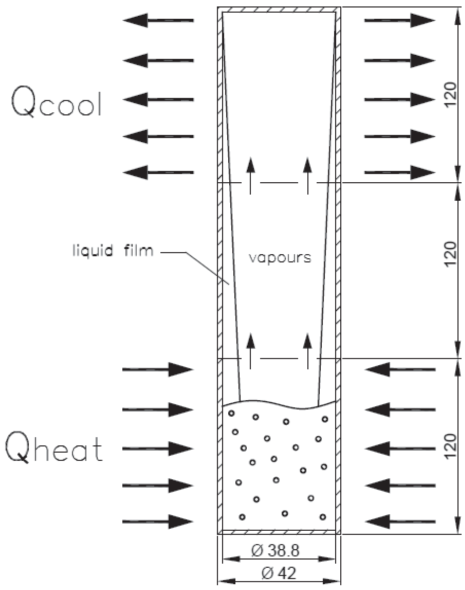


**Fig. 1:** The schema of the thermosyphon used for the simulation

Farsi et al. [5] assumed constant values of heat transfer coefficients what allow to use the based model only for given condition and one, particular refrigerant (in this case – pentane). That is why additional equations allowing to calculate heat transfer coefficients were added to the model. The evaporator in the industrial application is heated by medium with varying temperature, so the amount of heat applied to the device is also changing. New model takes into account the differing amount of heat in the evaporator based on the initial temperature of the heating medium. In the model proposed by Farsi et al. [5] the heat was supplied by the electrical wire, so this value was fixed.

Correlations for calculation of heat transfer coefficient during phase change are determined empirically for certain conditions. Boiling heat transfer coefficient was calculated using Cooper correlation [6], because it shows good agreement with experimental results for wide range of comercially available refrigerants [7]:

$$\alpha_{Cooper} = 55 p_r^{0.12} (-\log(p_r))^{-0.55} M^{-0.5} q^{0.67} \tag{4}$$

Heat transfer coefficient in the condenser was calculated as the arithmetic average of coefficient obtained by Nusselt theory with and without correction factor designated experimentally [8]:

$$\alpha_{c_1} = 0.943 \frac{\lambda_{3l}^3 \rho_{3l} (\rho_{3l} - \rho_{3v}) g L_3}{\eta_{3l} (T_1 - T_3) H_3} \tag{5}$$

$$\alpha_{c_2} = 1.13 \frac{\lambda_{3l}^3 \rho_{3l} (\rho_{3l} - \rho_{3v}) g L_3}{\eta_{3l} (T_1 - T_3) H_3} \tag{6}$$

where:

$p_r$ – reduced pressure;
$M$ – molar mass, g/mol;
$q$ – heat flux, W/m$^2$;
$\lambda$ – thermal conductivity, W/m;
$\rho$ – density, kg/m$^3$;
$g$ – acceleration gravity, m/s$^2$;
$L$ – length, m;
$\eta$ – dynamic viscosity, Pa·s;
$T$ – temprature, K;
$H$ – heat of evaporation;

Index:
1 – evaporator section;
3 – condenser section;
$l$ – liquid;
$v$ – vapour.

Outlet temperature of the coolant in the evaporator and the condenser sections were calculated by the following formula:

$$T_{heat_2} = T_e + (T_{heat_1} - T_e)\exp\left(\frac{-k_e A_e}{m_{heat} c_{p_{heat}}}\right) \tag{7}$$

$$T_{cool_2} = T_c + (T_{cool_1} - T_c)\exp\left(\frac{-k_c A_c}{m_{cool} c_{p_{cool}}}\right) \tag{8}$$

where:

$T$ – temperature, K;
$k$ – heat trasfer coefficient, W/m$^2$K;
$A$ – heat transfer area, m$^2$;
$m$ – mass flow, kg/s;
$c_p$ – heat capacity, J/kg·K.

Index:
$cool$ – cooling water;
$heat$ – heating water;
$e$ – evaporator;
$c$ – condenser;
1 – inlet;
2 – outlet.

In order to check if additional equations gives fine agreement with the model proposed by Farsi et al. [5] the simulations for both versions of the model were carried out. To make the comparison more reliable, geometry of the pipe and the type of refrigerant (pentane) were the same as in the simulation conducted by Fasi et. al. [5].

Figure 2 shows how the evaporator temperature is changing with time for both versions of the model. After including of heat transfer coefficients, the temperature of the wall drooped approximately by 2.8%, while the temperature of the fluid increase approximately by 2.7%. Difference between the wall and refrigerant temperatures decreased by about 2°C. Additionally, thermosyphon quicker reaches the steady state – for the model with the corrections this time is approximately 2 times shorter. It is probably caused by different values of heat fluxes applied to the evaporator - in the proposed version of the model this parameter is changing with time. In the steady state heat transfer coefficients reach values similar to that assumed by Farsi et al. [5] – for the evaporation process 707 $\mathrm{W/m^2}$K was obtained, while for condensation process – 321 $\mathrm{W/m^2}$K (for the base model it was 720 $\mathrm{W/m^2}$K and 180 $\mathrm{W/m^2}$K, respectively). Taking into account compatibility of the correlations (4)–(6) with different refrigerants, the model can be used to analyze the thermosyphon operation based on the type of the used substances.

## 3. Validation of the model with experimental data

Outcome of the experiment were compared to the results predicted by numerical simulation and the possibility of using refrigerant R507 as the working medium was

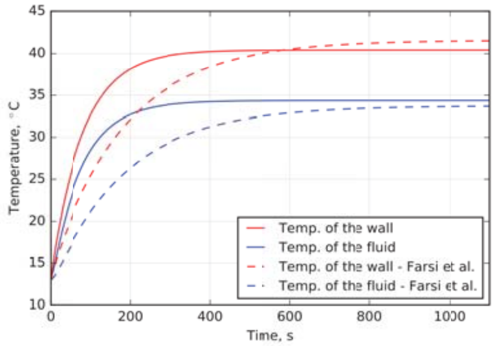


**Fig. 2:** Temperature evolution in the evaporation section for the model proposed by Farsi et al. [5] and for the model including proposed corrections

studied. Figure 3 presents the picture of experimental set-up which was used during the experiment. It consists of the thermosyphon made from stainless steel. The evaporator (condensing) section was heated (cooled) using cooper coil wrapped around the pipe. To allow the medium inside the coil to circulate, two circulation thermostats were used. It provided the temperature regulation at the outlet of the thermostat and this value could be changed according to the demands. To minimize heat losses (gaines) from the heating (cooling) water to the environment, the insulation was wrapped around the evaporator and condenser sections. The thermocouples measured temperatures at seven points along the length of the pipe (two at the evaporator and condenser sections, three at the adiabatic sections). To estimate how much heat was released (absorbed) by heating (cooling) water, the thermocouples were also placed at the outlet of the coils.

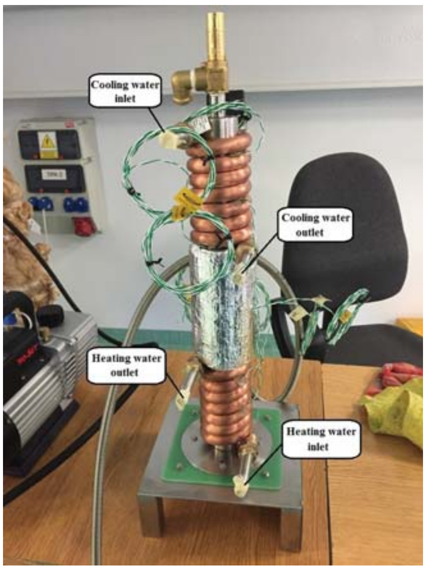


**Fig. 3:** Thermosyphon which was used to the experiment

### *3.1. Measurement method*

During the experiment the refrigerant R507 was used. The evaporation and condensing temperatures inside thermosyhon were assumed to be equal 25°C. The experiments were conducted for three different temperatures at the outlet of the thermostat (35°C, 40°C, 45°C) and two different mass flow rates (0.078 kg/s, 0.111 kg/s). The cold water was set for 15°C.

In fact, the temperature at the thermostat was not equal to that at the coil inlet, because of the temperature difference between the water inside the tubes and environment. The tube from the outlet of the thermostat to the inlet of the coil was not insulated, what increases heat losses. The temperature at the coil inlet cannot be constant as it is dependent on the ambient temperature. Heat losses to the environment were taken into account and the proper values of water inlet temperatures were calulated.

### *3.2. Results of the experiment*

The data obtained from the experiment were compared with proposed model. Figure 4 shows how the temperature in the evaporator changes with time according to the mathematical model and experiment for the temperatures at the circulating thermostats set as 40°C and 15°C. The mass flow rate was equal to 0.111 kg/s. The measurements were made at two points along the length of the evaporator section. One thermocouple showed higher values than predicted, the second one – lower. The average value gives almost excellent agreement, even during the transient state. One thermocouple was placed on the surface when the thermosyphon outer wall had no contact with the coil. The second one was installed between the pipes of the coil and even if there was no direct contact between the thermocouple and the pipe with hot water, this can affect the obtained results. Temperatures at the steady state obtained from two thermocouples was respectively 6.5% higher and 4.1% lower than the value results from the numerical model. In fact, the numerical model does not take into account the temperature difference between the upper and the bottom part of evaporator. That is why it is better to compare the simulation results with average value of the temperature arising from the experiment. The time needed for temperature stabilization is similar for both, experiment and simulation are equal to $\sim$200 s.

The same comparison were made for different heating water temperatures. Figure 5 presents the results obtained for temperature at the heating thermostat 35°C. The cooling temperature and the mass flow rate was without any change. In this case experimental data do not fit the numerical simulation in such good way as before. The differences are higher during the start-up process, but at the steady state, the model can predict the wall temperature with high accuracy. For these parameters, the average temperature at the evaporator was approx. 2.3% higher for the experiment than for mathematical model. The time of temperature stabilization is greater for the experiment and equal approximately 500 s.

The measurement error was calculated using Mean Absolute Deviation (MAD) and Mean Relative Deviation (MRD). The MAD shows accuracy of the measurement, while MRD checks if the result was over-predicted or under-predicted [9]:

$$MAD = \frac{1}{N} \sum \left| \frac{T_{(i)predicted} - T_{(i)measured}}{T_{(i)measured}} \right| \tag{9}$$

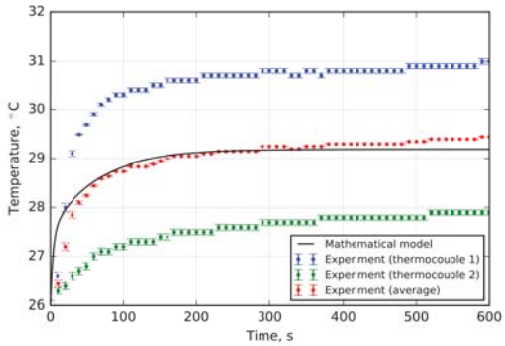


**Fig. 4:** Comparison of the temperatures in the evaporator section obtained by the experiment and numerical simulation ($T_{heat}$=40°C, $T_{cool}$=15°C, $m_{heat} = m_{cool}$ = 0.111 kg/s)

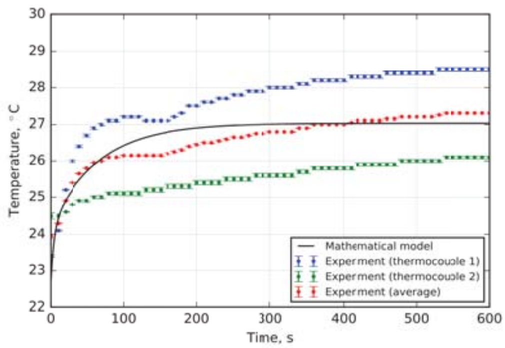


**Fig. 5:** Comparison of the temperatures in the evaporator section obtained by the experiment and numerical simulation ($T_{heat}$=35°C, $T_{cool}$=15°C, $m_{heat} = m_{cool}$ = 0.111 kg/s)

$$MRD = \frac{1}{N} \sum \frac{T_{(i)predicted} - T_{(i)measured}}{T_{(i)measured}} \tag{10}$$

Table 1 presents the values of temperatures at the steady state obtained for both, the experiment and numerical simulation for different initial parameters. Considered mass flow rate does not influence the temperature of the wall.

The experiments allowed to estimate the temperature along the entire length of thermosyphon. Figure 6 presents the average values of temperature in each section of heat pipe and the comparison with results arising from mathematical model.

The temperature of the adiabatic section was assumed to be constant during the entire process. The values predicted by numerical simulation closely follow the outcome of experiments. Condenser wall temperature stabilizes after 100 s, while the evaporator after about 200 s. At the beginning of the operation there is no liquid film. First the vapour must flow through the pipe to the upper part of the device, condense and then

**Table 1:** Average value of temperature of evaporation for different initial parameters including MAD and MRD

| | m = 0.071 kg/s | | | | m = 0.111 kg/s | | | |
|---|---|---|---|---|---|---|---|---|
| | Experiment | Model | MAD | MRD | Experiment | Model | MAD | MRD |
| | °C | °C | - | - | °C | °C | - | - |
| 35°C | 27.35 | 26.95 | 0.046 | 0.033 | 27.65 | 27.04 | 0.015 | −0.009 |
| 40°C | 29.46 | 29.07 | 0.032 | −0.012 | 29.55 | 29.19 | 0.009 | −0.002 |
| 45°C | 31.9 | 31.16 | 0.027 | 0.011 | 32.35 | 31.22 | 0.028 | −0.010 |

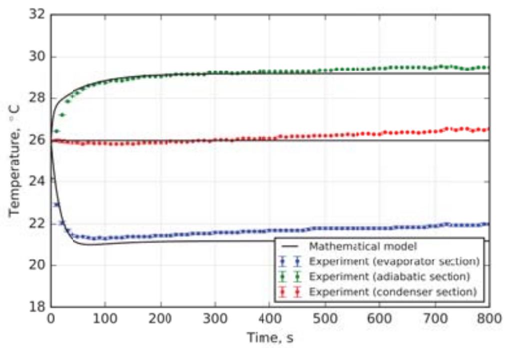


**Fig. 6:** Average temperature of the wall at each section of the thermosyphon as a function of time ($T_{heat}$=40°C, $T_{cool}$=15°C, $m_{heat}$ = $m_{cool}$ = 0.111 kg/s)

come back to the evaporator section. The time needed for that affect the temperature response of the evaporation section and slow the temperature stabilization at that part.

Figure 7 shows the temperature distribution along the pipe. Highest parts of evaporator posses lower temperature, because the inlet of the heating medium was placed at the bottom of the device. Some amount of heat was transferred to the evaporating refrigerant inside the pipe what decrease the temperature of the coolant and the wall with increasing length of thermosyphon. In this section the temperature rises with time and stabilize at 29°C in the steady state. In the condenser with the increasing length of the pipe, temperature drop is observed. The adiabatic section posses almost constant temperature during the whole operation of the device.

The thermal efficiency of thermosyphon is expressed by ratio of the output heat by condensation and input heat by evaporation:

$$\eta = \frac{Q_c}{Q_e} \tag{11}$$

The model assumed that at the steady state all the heat absorbed in the evaporator is transfer to the condenser and released to the cooling medium (efficiency equal 1). During the experiment the heat transferred to and from thermosyphon was calculated based on the temperature difference between the inlet and outlet of the coil with heating or cooling medium. The efficiency in this case has value 0.63. The efficiency drop is caused by the heat losses – not all the heat which is taken from the heating water is

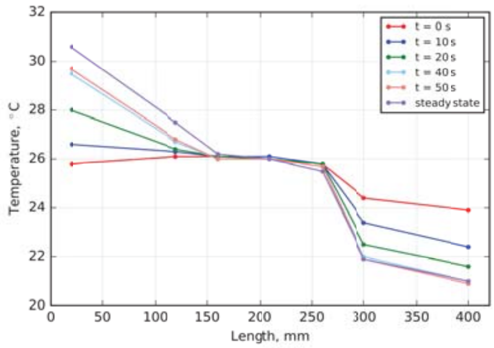


**Fig. 7:** Temperature distribution along the heat pipe ($T_{heat}$=40°C, $T_{cool}$=15°C, $m_{heat}$ = $m_{cool}$ = 0.111 kg/s)

transfer directly to the refrigerant. Some part is absorbed by the evaporator wall and some is rejected to the environment.

## 4. Conclusions

Possibility of using R507 as a working medium in gravitational heat pipe was analyzed. This refrigerant turned out to posses good properties under considered conditions. It allow the device to transfer heat efficiently. Wall temperatures stabilizes at fixed value, what means that the device is working continuously. No heat transfer limitations occur, so the amount of heat applied to the evaporator and rejected from the condenser was in a proper range according to the type of the used substance and the filling ratio.

The experimental results were compared with mathematical model. The analysis showed that the simulations were in good agreement with experimental data, especially at the steady state. The model did not take into account the varying temperature across the length of the pipe, so more suitable was to make comparison with average experimental values. The differences between the results collected from the experiment and predicted by the simulation did not exceed 5%. Especially, the average values of the measurements fits the simulation with high accuracy. The time of temperature stabilization from the model an experiments is equal to about 200 s.

The efficiency of the thermosyphon was calculated. The model assumed that the heat transfer occurs without any losses. During the experiment the efficiency was equal to 0.63. The temperature distribution along the pipe was also determined. The temperature has the highest value at the lower part of evaporator and then decreases with increasing length of the pipe. In the adiabatic section, temperature is almost constant, because there is no heat exchange between the fluid inside the pipe and the environment. The upper part of the device is the coldest one.

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