Loop heat pipe dynamics modeling and analysis

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Abstract. The report presents mathematical model and computational analysis of dynamic processes in the loop heat pipe, as well as some ways of regulating its work. This is considered a one-dimensional unsteady model of the loop. For each element of loop the model contains the energy equation for flow of fluid and to the walls, and the equation of conservation of momentum of the working fluid. The joint solution of a system of such equations for all elements of the loop allows to calculate the temperature distribution and pressure contour at any point in time. The calculations showed that under certain conditions in the loop heat pipe can be auto-oscillations of different nature. Analysed methods of eliminating these auto-oscillations. One way to regulate the loop heat pipe is the use of a control valve on the additional bypass line. The analysis showed that this method can significantly improve the quality of temperature control and reduce energy consumption by regulation or do without energy consumption. Presents a model of the dynamics of loop heat pipe gives physically reasonable results of calculations and is consistent with the results of the tests.

1. Introduction

One of the most effective devices for diverting heat from both ground and on-board equipment is loop heat pipes (LHP) [1]. In some cases, it is necessary to ensure temperature stability when the working conditions of the equipment change. To do this, additional measures should be taken to regulate the contoured heat pipe. One such measure may be the use of a bypass line with a valve-regulator, which ensures the transfer of part of the vapor from the vapor line to the compensation chamber. This leads to the displacement of the liquid from the compensatory cavity into the capacitor, the reduction of the condensation zone and the increase in the temperature of the evaporator. By regulating the consumption of vapor in the bypass line with the help of a valve-regulator can regulate the temperature of the vaporizer. In practice, this method has shown its performance and reliability [2].

For successful design of LHP-based thermo-stabilization systems, calculation LHP models with bypass and valve-regulator in its composition are needed. The development of the LHP mathematical model has been carried out by different authors for a long time [3-11], but even in a stationary form the model has problems due to complex and interconnected heat-hydraulic processes in LHP.

The dynamics model of LHP without the bypass line, is also of independent importance. It allows you to calculate the launch of LHP, transients, the occurrence of auto-oscillations, and identify possible problems of stable LHP operation. This is important because LHP is an autonomous self-regulating device.

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**Figure 1.** Schematics of the simplified circuit LHP: 1 - porous wick; 2 - vapor line; 3 - condenser; 4 - condenser line; 5 - segment condenser line inside the reservoir; 6 - reservoir; 7 - pump module head; 8 - bypass line; 9 - bellows; 11 - valve.

2. **Modeling of the simplest LHP**

2.1. **Mathematical modeling**

An important feature of loop heat pipes is the self-regulation of their work. It is determined by a complex of interrelated heat and hydraulic processes. Numerous internal feedbacks, the physical nature of which is not always clear, make it difficult to use traditional universal methods of calculation.

As a first step we will consider model of LHP dynamics having simplified circuit (Figure 1). The loop has been divided into j elements shown in Figure 1 and for each j-th element of the loop we will write down the equations of energy for the working fluid flow and for a wall, and also the conservation of momentum equation of the working fluid:

\[
\frac{\partial (\rho h_f)}{\partial \tau} + \frac{\partial (\rho u_f h_f)}{\partial z} = \lambda_f \frac{\partial^2 t_f}{\partial z^2} + q_f
\]  
(1)

\[
\rho_w c_w \frac{\partial t_w}{\partial \tau} = \lambda_w \frac{\partial^2 t_w}{\partial z^2} + q_w
\]  
(2)

\[
\frac{\partial p_f}{\partial z} = \frac{\xi \rho u_f^2}{2} + g \frac{\partial u_f}{\partial z}
\]  
(3)
Figure 2. Schematics of the simplified circuit LHP with a valve-regulator: 1 - porous wick; 2 - vapour line; 3 - condenser; 4 - condenser line; 5 - segment condenser line inside the reservoir; 6 - reservoir; 7 - pump module head; 8 - bypass line; 9 - bellows; 11 - valve.

Boundary conditions for these equations (1)-(3) we will write down as:

At $z=0$

$$\begin{align*}
(h_f)_j &= (h_f)_{j-1} \\
(t_w)_j &= (t_w)_{j-1} \\
(p_f)_j &= (p_f)_{j-1}
\end{align*}$$

(4)

At $z=l_j$

$$\begin{align*}
\left(\frac{\partial h_f}{\partial z}\right)_j &= \left(\frac{\partial h_f}{\partial z}\right)_{j-1} \\
\left(\frac{\partial t_w}{\partial z}\right)_j &= \left(\frac{\partial t_w}{\partial z}\right)_{j-1}
\end{align*}$$

(5)

For the vapour line at $z=0$ it is accepted

$$(p_f)_1 = (p_f)_4 + \Delta p$$

(6)

Where $\Delta p \leq \frac{2\sigma}{R}$ is the general pressure losses in a LHP. For the porous cartridge at $z=l_j$ it is accepted that:

$$\left(\frac{\partial t_w}{\partial z}\right)_4 = \left(\frac{\partial t_w}{\partial z}\right)_1 + \frac{Q}{\lambda_w J^2}$$

(7)

The correlation linking the enthalpy and the temperature of the working fluid was determined with the help of the following rationale:
\[
\begin{cases}
  \text{if } h' = h'_s & t = t_S + \frac{h'_{s} - h}{c_r} \quad x = 0 \\
  \text{if } h'_s < h < h'' & t = t_S \quad x = \frac{h - h'_s}{r} \\
  \text{if } h' > h'_s & t = t_S + \frac{h - h'_{s}}{c_r} \quad x = 1
\end{cases}
\] (8)

Where \( x \) is the vapour quality. Saturation temperatures of working fluids in each section were determined as function of pressure on the equations describing relation of pressure and temperature on the saturation line. The homogeneous model according to which the density was determined on the equation was applied to the two-phase working fluid.

Source components in the equations (1) and (2) were determined as follows:

\[
q_f = \frac{a_{in}(t_w - t_f)b_{in}}{\omega_f} \quad (9)
\]

\[
q_w = \frac{[a_{ext}(t_{ext} - t_w)b_{ext} - a_{in}(t_w - t_f)b_{in} + q_l]}{\omega_w} \quad (10)
\]

With help of the combined equation of balance of heat and weights for the reservoir

\[
\rho \left( \frac{dm_f}{dt} - G_4 x_4 \right) = k_R(t_4 - t_R) + (\alpha_{ext})_R(t_R - t_4)S_4 - C_R \frac{dt_R}{dt} - G c'_4 (t_R - t_4) \quad (11)
\]

it is possible to determine weight of the vapour of working fluid in the reservoir at any moment of time. The working fluid’s vapour density in the reservoir was determined on the ratio

\[
\frac{\rho_f}{\rho} = \frac{m'_{fR}}{V'_R} \quad (12)
\]

Where volume of a vapour phase in the reservoir,

\[
V'_R = \frac{v_{Rz} \rho' - (m_{f})_0 - (m_{f})_x}{\rho' - \rho} \quad (13)
\]

and \((m_f)_x\) is the weight of the working fluid in the loop expressed by:

\[
(m_f)_x = \Sigma \int_0^l (\rho_f \omega_f \partial z)_l + (m')_R + (m'')_R \quad (14)
\]

Where the temperature \( t_R \) of working fluid in the reservoir was determined during solution of the written down system of equations. Density of the vapour calculated on equation (12) is equal to density of saturated vapour at the temperature \( t_R \).

The temperature \( t_7 \) of temperature-stabilized item at any moment could be determined from the equation of thermal balance:

\[
Q_7 - C_7 \frac{dt_7}{dt} - \alpha_{ext}S(t_7 - t_{ext}) = k_7(t_7 - t_1) \quad (15)
\]
2.2. Modeling of LHP temperature regimes when changing heat load
The results of loading step change calculations (5 W and 15 W) are presented in Figure 3. As it can be seen, this change at absence of any measures on regulation corresponds to change 4.5 °C of the temperature-stabilized item’s temperature.

![Graph showing temperature changes](image1)

**Figure 3.** Behavior in time of the temperature in various points of the LHP at step change of the power. (1 – pump module head, 2 - evaporator wall, 3 - saturation in the evaporator, 4 - saturation in the reservoir, 7 - output from the condenser).

The temperature difference between the cooling object and the environment varies from 7 to 2.5 degrees. In time there is a redistribution of the working fluid rate. At that the some periodic filling and draining of the condenser are occurred.

The results of numeric analysis of the LHP startup and transition from one mode to another. It is shown that these processes are not always successful (curves 1-3 in Fig. 3). The key parameters that determine their success are the mass of the working fluid charged and a heat capacity of compensation chamber.

![Graph showing time dependencies](image2)

**Figure 4.** The LHP evaporator time dependencies when control is absent and heat removal zone effective temperature is suddenly changed from -45°C to -37°C when different values of compensation chamber total heat capacity (mc, J/K): 1 - 0; 2 - 10; 3 - 20; 4 - 50; 5 - 100 and when d4 = 1 mm, mc = 20 J/K (curve 6). Shade line - are stationary regimes temperatures ( -45°C and -37°C).
2.3. Auto-oscillation at LHP start-up

Results of calculations LHP characteristics with length of a vapor line and liquid line 0.2 m, the condenser is 0.5 m, diameter of a vapor line and the condenser is 1 mm, liquid line is 0.6 mm are presented on Figure 5.

**Figure 5.** Start-up of LHP when a heat transfer factor between the evaporator and reservoir negligible: 1 – temperature of the evaporator; 2 – saturation temperature; 3 – temperature on an outlet of condenser.

The working fluid is ammonia. Evaluation of the evaporator over the condenser is absent. Cooling of the condenser is convective. A heat load of evaporator is 30 W.

Calculation results is shown, that at small value of a heat transfer factor between evaporator and reservoir, there can be significant self-oscillations of the liquid, accompanied moving of vapor from the condenser to reservoir and return liquid flow from reservoir. At increase in a heat transfer between the evaporator and reservoir auto-oscillations decrease and become insignificant.

2.4. Influence of the variable evaporator heat load on the quality of thermal stabilization

An example of the calculation of LHP for a stepwise change in the heat load on the evaporator is considered. Figure 6 shows the graph of temperature variation in the loop without a regulator. If the load varies from 50 to 200 W, the temperature of the evaporator (line 1) can be changing by 15 degrees. Moreover, thermal-hydraulic self-oscillations of temperature are observed because of the "breakthrough" of steam from the condenser to the reservoir.

**Figure 6.** The calculated time variation of temperature in different points of the LHP without the regulators when changing the heat load of the evaporator. 1 – evaporator temperature, 2 – reservoir temperature, 5 – outlet temperature from the condenser.
Figure 7 shows the results of the LHP tests for different values of the heat load on the evaporator. As can be seen, fluctuations in the temperature of the evaporator are observed, the amplitude of which, as in the calculations, is of the order of 5 degrees.

An important parameter of the temperature control system is the quality of thermo-stabilization. In LHP to improve the quality of temperature stabilization uses additional units - device of heat action, hydraulic regulators with bypass lines or without them. Conducted design analysis of the LHP work with such devices (Figure 8). Conditions of quality temperature stabilization is defined.

Figure 7. LHP test results. Working fluid - ammonia, condenser cooling temperature - 150°C, evaporator heat power - 0-120 W.

Figure 8. The calculated time variation of temperature in different points of the LHP with regulators when changing the heat load of the evaporator. 1 – evaporator temperature, 2 – reservoir temperature, 5 – outlet temperature from the condenser.

3. Modelling of the LHP with bypass regulation
Circuit of the LHP with bypass and a valve - regulator is presented on Figure 2. As a first approximation by processes of heat exchange in bypass lines it is neglected. We suppose that hydraulic resistance bypass is not enough pipeline in comparison with resistance of the valve.
The working fluid rate through bypass line $G_8$ is determined, on the one hand, by difference of pressure between vapour line 2 and the reservoir 6, on the other hand - factor of hydraulic resistance of the valve 11. If we assume that inertia forces have negligible role, the rate could be found from following equation

$$p_2 - p_6 = \zeta_{V_2} \frac{G_8^2}{2 \rho \omega^2} - \frac{G_8^2}{2 \rho \omega^2}$$ (16)

In case of LHP with bypass line the design model presented in section 1.1, will contain the following features:

1) The working fluid rate through vapour line and further - the condenser and condenser line should be reduced on the value;
2) Pressure losses in vapour line will increases on value of valve–regulator’s local losses that are approximately equal to:

$$\Delta p = \zeta_{V_2} \frac{G_2^2}{2 \rho \omega^2}$$ (17)

3) The equation of balance of heat and weights of the reservoir (13) should take into account additional working fluid inflow $G_8$. We will write down this equation as:

$$r \left( \frac{d m}{d \tau} - G_4 x_4 - G_B \right) = k_R (t_1 - t_R) + (\alpha_{ext})_R (t_{ext} - t_R) S_R + (\alpha_{in})_R (t_4 - t_R) S_4 + \cdots$$

$$\cdots - C_{R_2} \frac{d x_2}{d \tau} + \cdots - G_4 c'(t_4 - t_4)$$ (18)

The basic problem of calculation is related with determination of the factors $\zeta_{V_2}$ and $\zeta_{V_8}$ of hydraulic resistance for equations (16)-(17). Values of these factors could be changed and they are depending on the valve position.

Let’s assume that $y$ is the axis along which the valve moves. The $y_1$ and $y_2$ are extreme positions corresponding the completely closed and completely open bypass valve. Then range of the change of the valve is $y_1 \leq y \leq y_2$. We will assume that factor of the valve local resistance is inversely to degree of the valve opening, i.e. $\zeta_8 = 1/A_1 (y - y_1)$, where $A_1$ is a numerical factor. In view of it from the equation (16) we will receive:

$$G_B = \left( A_1 (y - y_1) [ (p_2 - p_0) 2 \rho \omega^2 ] + G_2^2 b 2 \omega^2 \omega^2 \right)^{0.5}$$ (19)

Let us assume that gas pressure in bellows is proportional to the absolute temperature, and the value of valve movement depends linearly on pressure difference between the bellows and loop. Then

$$\frac{A_2}{y} (t_9 + 273) - p_S = A_3 (y - y_0)$$ (20)

Where $y_0$ is the coordinate of the valve position at equality of pressure between the bellows and loop. In case of ideal gas in bellows $A_2 = m R / S$, where $m$ is gas weight in bellows, $R$ is the gas constant, $S$ is the area of cross-section section bellows and lastly $A_3$ is the factor of bellows stiffness. From last equation it follows that
\[ y = \frac{1}{2} \left( \frac{pS_3}{A_3} - y_0 \right) + \left( \frac{1}{4} \left( \frac{pS_3}{A_3} - y_0 \right)^2 + \frac{A_2 S}{A_3} (t_9 + 273) \right)^{0.5} \] (21)

By solving the equations (18), (19) and (21) together with the equations system (1)-(16) it is possible to determine the valve position, working fluid rate in bypass line and the all LHP characteristics. Operation of the valve - regulator is determined by the following numerical characteristics: \( A_1, A_2, A_3, y_0, y_1, y_2 \). At the using inert gas in bellows and absence of heat supply to the bellows these factors are constants. In case of use of two-phase agent and heat supply some of them, and first of all could be changed. It is resulted in changing parameters of regulation.

Figure 9 shows the calculation results of LHP parameters change at the cooling temperature 25°C and at heat generation change in the pump module head. Behaviour of power change is presented in Figure 9. Initial value of the power is 15 W. Gradual decrease of the power by 2 W was observed in every 20 seconds. After reaching 5 W power was increased up to 15 W. In a minute the process was repeated. In accordance with the results pressure regulator provides high quality of saturation temperature stabilization in LHP (curve 4).

![Figure 9. Change in time of LHP parameters when changing the evaporator's heat load](image)

Stabilization of evaporator body temperature (curve 3) is quite reasonable too. However, the pump module head temperature (curve 2) is changed within two degrees due to the thermal contact and heat capacity and that is above the required normal twice. Working fluid rate at the evaporator outlet (curve 6) and working fluid rate at the compensation chamber inlet (curve 7) are equal only in the set mode and may be significantly different when the power is changed that leads to filling condenser with liquid or working fluid removal from condenser (curve 8).

Changing of the clearance \( y \) that characterizes the bypass valve opening is presented by the curve 11 in the bottom diagram. Decreasing the power the valve clearance is decreased too, and at the moment of the power increase the complete closing of the bypass valve during about 30 seconds is occurred.

Thermal-hydraulic auto-oscillations are possible during LHP operation. Fig. 9 shows the example of the steady LHP operation after single cooling temperature decrease (curve 7) from 0 to -5°C. Steady oscillation process with the oscillation period of 100 seconds is observed. It should be noted that there is no total closing of the bypass valve. Oscillations are caused by interaction of thermal and hydrodynamical processes. These oscillations lead to oscillations of the pump module head temperature. Its temperature is within the range of ±0.2°C.
Figure 10. Change in time of LHP parameters with a change in the temperature of the cooling medium (designation of the lines, see Figure 9).

Calculations showed that with a decrease in the cooling temperature by 1 degree every 10 seconds due to the regulator, the temperature of the plate does not practically change, while the cooling temperature decreased by more than 30 degrees. With a stepwise increase in temperature by 1 degree (Figure 10), reverse phenomena are observed which have features. At a cooling temperature of more than 34°C (and adjusting the regulator to maintain 42°C), a qualitative change in the occurring phenomena occurs. At low cooling temperatures, the LHP control was performed by changing the position of the regulator valve. At high temperatures, the valve is normally closed, and self-regulation is provided by varying the supercooling of the liquid at the outlet of the condenser. If at low temperatures a monotonic opening of the condenser occurs (line 8 in Fig. 9), then at a high temperatures rapid change in the length of the condensation section at the time of temperature change.

4. Conclusions
The presented model of the LHP dynamics gives physically grounded calculation results and is consistent with the test results.

The installation of the bypass line and the regulating valve in the LHP allows to significantly improve the quality of thermo-stabilization in comparison with the traditional methods, with no energy costs for regulation.

Presence of two free border of phases (in the condenser and in a compensatory camber) can cause moving of a liquid to a direct and return direction at insignificant additional influence. Calculation results is shown, that under certain conditions there can be auto-oscillations of the liquid, accompanied moving of vapor from the condenser to reservoir and return liquid flow from reservoir.

The cyclic heat input can provoke increase thermo-hydraulic auto-oscillations of the working fluid in a loop that can lead to growth of thermal resistance in 1.5 times and to essential rise in temperature of the evaporator. Thus, the LHP continues to work, but essential growth of temperature in comparison continuous heat load in tests can wrongly be treated, how output LHP out of operation.

During the settlement analysis the opportunity of failure LHP owing to arising auto-oscillations is not established at a cyclic supply of thermal loading. In all cases the LHP keeps working ability though integrated characteristics can essentially different.
5. **Nomenclature**

| Symbol | Description |
|--------|-------------|
| $A_3$ | bellows stiffness coefficient (kg/m) |
| $b$ | perimeter of the heat-exchanger surface (m) |
| $C$ | total heat capacity (J/K) |
| $c$ | specific heat capacity (J/(kg·K)) |
| $d$ | diameter (m) |
| $G$ | mass flow rate (kg/s) |
| $i$ | specific enthalpy (J/kg) |
| $k$ | heat transfer coefficient (W/m²K) |
| $m$ | mass (kg) |
| $p$ | pressure (Pa) |
| $R$ | gas constant; radius of pores (m) |
| $r$ | latent heat (J/kg) |
| $S$ | area (m²) |
| $Q$ | heat flow (W) |
| $q$ | specific heat flow (W/m³) |
| $q_i$ | linear density of heat flow (W/m) |
| $t$ | temperature ($^\circ$C) |
| $u$ | velocity (m/s) |
| $V$ | volume (m³) |
| $x$ | mass vapor content |
| $z$ | coordinate along the loop (m) |
| $\alpha$ | heat exchange coefficient (W/(m²K)) |
| $\lambda$ | heat conductivity (W/(m·K)) |
| $\rho$ | density (kg/m³) |
| $\tau$ | time (s) |
| $\xi$ | coefficient of hydraulic resistance |
| $\sigma$ | surface tension (N/m) |
| $\omega$ | cross-sectional area (m²) |

**Subscripts:**

| Subscript | Description |
|-----------|-------------|
| ext | external |
| $f$ | working fluid |
| $j$ | sequence number of the section (1 - evaporator, 2 - vapor line, 3 - condenser, 4 - liquid line); |
| in | internal |
| $w$ | wall |
| ′ | liquid parameters |
| ″ | vapor parameters |
| R | reservoir |
| s | saturation parameters |

6. **References**

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