Method of determining operation parameters of stand tests of vehicle suspension elements under conditions of ultra-low temperatures

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Abstract. This article presents the description of the method for estimation of heat flows characteristic during research of operability of suspension elements under the conditions of ultralow temperature. The mentioned research is performed at the test bench created in VSTU. Solid carbon dioxide (also known as artificial ice) is used as the coolant. The method provides evaluation of required quantity of artificial ice and cooling dynamics of the test unit in the insulated low-temperature chamber with structurally specified parameters of heat insulation. Also the method allows computation of heat intake into the chamber volume.

1. Introduction
The test bench that provides researching of operation processes in hydraulic, pneumatic, hydropneumatic and other types of suspension elements of wheel and tracked vehicles, performing a comparative analysis of its operational characteristics and testing the operability of suspension elements during operation at air temperature below -60 °C was developed and created in VSTU. This article presents the main statements of the method of estimation of testing modes parameters for conditions of ultra-low temperatures.

2. Estimation of parameters of ultra-low temperatures tests
For estimation of the necessary mass of artificial ice [3, 4] and evaluation of dynamics of the spring cooling in the chamber with design specified parameters of heat insulation calculation of heat intake to the chamber volume (fig. 1) was made.
Figure 1. Scheme of low temperature chamber with HPS: 1 - HPS, 2 - HPS rod, 3 - lower rod extender, 4 - upper rod extender, 5 - oil pressure sensor (OPS), 6 - air pressure sensor (APS), 7 - thermocouple introduced in body of HPS case for measuring HPS temperature, 8 - heat insulator of thermocouple, 9 - thermocouple for measuring temperature in the chamber volume, 10 - thermocouples wires, 11 - case of the low-temperature chamber, 12 - top cover, 13 - door for coolant loading, 14 - bottom cover, 15 - heat insulation, 16 - coolant volume, 17 - seal in connection between case and covers, 18 - rod seal

Total heat intake is the sum of intake through the side wall and end covers and also intake through the metal of upper and lower rods. It is assumed that thermal resistivity of insulated side walls of the chamber is constant in the whole height, and the end covers transfer the heat flow through the surface which is equal to their external surface. The calculation is performed for the steady-state mode with temperature -60 °C.

Heat flow from outside through the walls to the chamber volume is calculated using the known formula for heat transfer through a flat wall:

\[ Q = \frac{(t_u - t_f)}{\Sigma R_t}, \]  

(1)

where \( \Sigma R_t \) – total overall thermal resistivity of chamber walls.

Total overall thermal resistivity of chamber walls is:

\[ \Sigma R_t = \frac{I}{\alpha_{ex}F_{ex}} + \sum_{i=1}^{n} \frac{\delta_{si}}{\lambda_sF_{si}} + \frac{I}{\alpha_{in}F_{in}}, \]  

(2)

where \( \alpha_{ex}, \alpha_{in} \) – average coefficients of heat transfer on external and internal surfaces of a chamber wall;

\( F_{ex} \) – area of external surfaces of a chamber;
\( F_{in} \) – area of internal surfaces of a chamber;
\( F_{si} \) – areas of wall parts with different thickness of heat insulation;
\( \delta_{si} \) – thickness of heat insulation of chamber walls;
\( \lambda_s \) – thermal conductivity of heat insulation material;

Values of heat emission coefficient \( \alpha_{ex} \) on external surfaces of the chamber are specified in accordance with the recommendations about selection of this coefficient values for heat losses in living accommodations and workrooms [2,3]. At this larger value, this coefficient was taken: \( \alpha_{ex} = 15 \ W/(m^2 \cdot K) \).

It is hard to define precisely the averaged value of heat emission coefficient \( \alpha_{in} \) on internal surfaces of the chamber as the coldest area in the chamber volume is located in the bottom part of the chamber. Taking into account that the significant part of the internal surface of the chamber wall contacts with artificial ice, it is possible to think that surface temperature is close to temperature of artificial ice. Averaged temperature on the internal surface of the chamber wall is assumed to be –60 \(^\circ\)C. Heat flow through the walls into the chamber in this case would be calculated from specifying of border conditions of the first kind on the internal surface:

\[
Q = \frac{(t_a - t_{w2})}{\frac{1}{\alpha_{ex} F_{ex}} + \sum_{l}^{n} \frac{\delta_{si}}{\lambda_s F_{si}}}, \tag{3}
\]

where \( t_{w2} \) – temperature on internal surface of a chamber wall.

Thermal resistivity of heat emission on external surface of the chamber:

\[
R_{te_{ex}} = \frac{1}{\alpha_{ex} F_{ex}} = \frac{1}{15.2[(0.3 \cdot 0.59)+(0.4 \cdot 0.59)+(0.3 \cdot 0.4)]=0.0625 \ m^2 \cdot K/W.}
\]

The coefficient of thermal conductivity of foam propylene is taken \( \lambda = 0.03 \ W/(m \cdot K) \) in accordance with the reference data. Thermal resistivity of walls:

\[
R_{\lambda} = \sum_{l}^{n} \frac{\delta_{si}}{\lambda_s F_{si}} = \frac{0.1+0.09}{0.03 \cdot 0.3 \cdot 0.5} + \frac{0.05+0.05}{0.03 \cdot 0.1 \cdot 0.5} = 102.8 \ m^2 \cdot K/W.
\]

Thus, it is possible to neglect value of thermal resistivity of emission in comparison with value of thermal resistivity of insulation. So, the following value was obtained:

\[
Q = \frac{t_a - t_{w2}}{R_{\lambda}} = \frac{20 - (-60)}{102.8} = 0.778 \ W.
\]

Hydropneumatic spring (HPS) contacts with ambient air by the part of the surface of the upper rod and lower rod extenders which go through the bottom and top end cover accordingly, and connect the spring to the test stand (fig. 1).

For the described construction fairly accurate values of the heat flow from surroundings could be found only by means of detailed modeling of overall heat transfer processes. But approximate evaluation of these heat flows can be obtained from consideration of ribs as long rods with constant cross section. So it is the solution of the reverse problem - case of heat dissipation from surface of a rod to surroundings [3,4]. Heat dissipation from a rod with finite length is:

\[
Q_b = \partial \lambda_s d S_c \theta (\delta L), \tag{4}
\]

where \( \theta \) – excessive temperature on the rod end; \( \lambda_s \) – thermal conductivity of rod material; \( S_c \) – rod cross-section area; \( L \) – rod length; \( \phi \) – parameter used for taking into account rod geometry and relation between thermal resistivity of heat transfer and heat emission. Parameter
\[ \phi = \sqrt{\alpha_c \cdot \Pi / (\lambda_s \cdot S_c)}, \] 
where \( \Pi \) – perimeter of rod cross-section. Rods material (alloy steel) conductivity is taken [4] equal to 20 \( W/(m\cdot K) \).

For upper rod:
\[ \phi = \sqrt{\alpha_c \cdot \Pi / (\lambda_s \cdot S_c)} = \sqrt{15 \cdot 3.14 \cdot 0.03 / (20 \cdot 3.14 \cdot 0.03 \cdot 0.03 / 4)} = 10 \text{ m}^2 / \text{sec} \]
and
\[ Q_o = \partial \lambda_s \phi S_c \text{ th}(\phi L) = 80 \cdot 20 \cdot 3.14 \cdot 0.03^2 \frac{3}{4} \text{ th}(10 \cdot 0.12) = 9.49 \text{ W}. \]

For bottom rod
\[ \phi = \sqrt{\alpha_c \cdot \Pi / (\lambda_s \cdot S_c)} = \sqrt{15 \cdot 3.14 \cdot 0.048 / (20 \cdot 3.14 \cdot 0.048 \cdot 0.048 / 4)} = 7.9 \text{ m}^2 / \text{sec} \]
and
\[ Q_o = \partial \lambda_s \phi S_c \text{ th}(\phi L) = 80 \cdot 20 \cdot 7.9 \frac{3.14 \cdot 0.048^2}{4} \text{ th}(7.9 \cdot 0.05) = 13.65 \text{ W}. \]

Total heat intake into the chamber through its walls and through the spring rods is 24 \( W \).

To keep temperature -60 °C in the chamber it is necessary to compensate heat intake by its absorption during sublimation of artificial ice. As every gram of artificial ice absorbs 590 \( J \) of heat, to keep specified temperature it is necessary to consume 150 \( g \) of artificial ice per hour:

\[ m_{\text{CO}_2} = 2 \frac{4}{590} \cdot 3600 = 146 \text{ gph}. \]

To estimate consumption of artificial ice for the single test at low-temperature, it is necessary to calculate its mass for decreasing of air temperature and temperature of the spring from 20 °C down to -60 °C. Spring metal mass is about 12 kg, oil mass in the spring is about 0.65 kg. The thermal capacity of metal is taken equal to 0.5 \( kJ/(kg \cdot K) \). Oil thermal capacity \( c_l = 1.5 \text{ kJ/(kg \cdot K)} \).

Heat quantity that has to be taken off:
\[ Q_r = (m_c c_s + m_l c_l)(t_0 - t_r) = (12 \cdot 0.5 + 0.65 \cdot 1.5) \cdot (20 - (-60)) = 558 \text{ kJ}. \]

To absorb heat quantity \( Q_r = 558 \text{ kJ} \) about 0.95 kg of artificial ice must be converted to the gas phase.

Evaluation of temperature field non-uniformity is performed for the stage of test unit heating after overcooling it to the temperature that is close to the equilibrium temperature of artificial ice (70 °C) and after extracting of artificial ice from the low-temperature chamber.

To evaluate rate of temperature field non-uniformity in the test unit metal value \( B_i \) was defined. At this, cylinder wall thickness was chosen as the main parameter. Value of the heat emission coefficient for external surface of the cylinder is defined using the similarity equation for the case of natural convection along a vertical wall.

Average air temperature in the chamber during the stage of spring heating to the surface temperature – 60 °C is assumed to be – 35°C. At this, temperature air kinematic viscosity is \( \nu_a = 10.42 \cdot 10^{-6} \text{ m}^2 / \text{sec} \) and thermal conductivity is \( \lambda_a = 0.0216 \text{ W/(m \cdot K)} \) [1, 5, 6]. The coefficient of thermal expansion of air is \( \beta_a = 0.00366 \text{ 1/K} \). Defining dimension - the spring body height is \( L_r = 460 \text{ mm} \). The cylinder wall thickness is \( \delta_s = 8 \text{ mm} \). Under these conditions product of Grashof number and Prandtl number was calculated:

\[ Gr \cdot Pr = \frac{g \cdot \beta_a \cdot \delta_s^3 \cdot \Delta T}{\nu_a^2} \cdot Pr = \frac{9.81 \cdot 0.00366 \cdot 0.460 \cdot (35 - (-60))}{(10.42 \cdot 10^{-6})^2} \cdot 0.725 \approx 2.33 \cdot 10^7. \]

At this value of complex \( Gr \cdot Pr \) similarity equation for defining of the coefficient of convective heat transfer at free convection along a vertical wall [3, 7] is:
\[
\text{Nu} = 0.6(\text{Gr} \cdot \text{Pr})^{0.25} \left( \frac{\text{Pr}_{\text{in}}}{\text{Pr}_{\text{r}}} \right)^{0.25}
\]  

(5)

For air, relation of Prandtl numbers [3, 4] specified for air temperature and wall temperature is almost equal to 1. Thus

\[
\text{Nu} = 0.6(\text{Gr} \cdot \text{Pr})^{0.25} = 0.6 \cdot (2.33 \cdot 107)^{0.25} = 41.7.
\]

Coefficient of the heat emission on the spring surface:

\[
\alpha_r = \frac{\text{Nu} \cdot \lambda_a}{L_r} = \frac{41.7 \cdot 0.0216}{0.460} \approx 2 \text{ W/(m}^2\text{K)}.
\]

Value of \(B_i\) is:

\[
\text{Bi} = \frac{\alpha_r \delta_r}{\lambda_s} = \frac{2 \cdot 0.09}{20} = 0.009.
\]

As far as obtained value \(B_i\) is less than 0.1 then it is clear that steel wall of the spring cylinder would heat uniformly on whole thickness.

Oil contained in the hydraulic volume of the test unit cylinder also would be heated relatively uniformly as in this volume heat would be transferred not only by the heat transfer but by oil circulation. Approximate scheme of circulation flows is shown in fig. 2. Intensity of circulation would be low since at first, this circulation takes place in limited volume and at second, at low temperature oil has high viscosity. Also, increasing the thermal conductivity of mineral oils during cooling leads to equalization of oil temperature in a volume.

![Figure 2. Approximate scheme of oil circulation](image)

The difference in temperatures between the cylinder wall and oil would be defined by the thermal resistivity of the heat transfer to oil. There is no reliable data on the value of the heat transfer coefficient in considered conditions and also data on physical characteristics of oil MGE-10A near the line of the phase transfer to solid state is not enough. But taking into account all conditions of oil heating, including the oil chamber geometry, it is possible to predict that difference between oil temperature and the spring body temperature wouldn’t be greater than 3°C.
3. Conclusions
For performing of stand researches of HPS operational processes under conditions of ultra-low temperatures method of computational estimation of heat flow parameters in the heat insulated low-temperature chamber for the test unit was developed.

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