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A CONTRIBUTION TO THE STUDY OF THERMAL LOAD OF TRUCKS POWERTRAIN SUSPENSION SYSTEM DUE TO VIBRATIONS

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vibration parameters,
degradation,
thermal load powertrain mounting system

Abstract

Dynamic simulation, based on modelling, has a significant role during to the process of vehicle development. It is especially important in the first design stages, when relevant parameters are to be defined. Powertrain mounting system is exposed to thermal loads which can lead to its damage and degradation of characteristics. Therefore, this paper attempts to analyse a conversion of mechanical work into heat energy by use of a method of dynamic simulation. Bearing in mind the most frequent application of classic - mechanical and hydraulic powertrain suspension in modern trucks, this paper will present analyses of thermal load of a vehicle manufactured by the domestic manufacturer FAP. The issue of heat dissipation from the powertrain mounting system has not been taken into consideration. The experimental verification of the results will be done in next investigations.

1. INTRODUCTION

As it is known, the vibration and noise from the powertrain of the vehicle are transmitted to the vehicle structure. In order to reduce this effect to a satisfactory measure, the powertrain is to be elastically linked to the vehicle structure. In the opposite direction, the same system absorbs vibration excitations coming from the road, and passing through the vehicle suspension system to the powertrain. Kinetic energy of the powertrain, which is a result of vibration, causes a mechanical work in powertrain mounts which is transformed into thermal energy [1].

In practice, in the stage of vehicle development, powertrain mounting system are chosen from the condition of damping of vehicle vibrations, but in order to avoid the negative impact on the function, thermal loads should be taken into consideration [2, 3]. The goal is to convert, as much as possible, the mechanical work received from the ground and powertrain into thermal energy which will be transferred to the environment and thus provide the cooling of powertrain mounting system. Wrong selection of mount characteristics, from the standpoint of thermal loads, can cause rapid degradation of its properties during service life. Excessive amount of heat energy, eventually kept "inside" the mounts, would cause rapid deterioration of sealing elements and loss of function of the damping element.

Tests have shown that the mechanical work is partly converted into the heat which is transferred to the mounts, and the remaining amount of heat delivery is transferred to the environment, thus cooling the powertrain mounting system. Mathematically, it can be displayed by the formula [3-7]:

$$A = Q_t + Q_f + Q_e \qquad (1)$$

where:

– A - mechanical work (equal to the quantity of heat),

- Q_t - quantity of heat transferred to the mount body,
- Q_f - quantity of heat transferred to the working fluid (in the case that it exists), and
- Q_e - quantity of heat transferred to the environment.

The work of the force in the mount is of relevance because it enables the analysis of its transformation into heat energy. The work of force in the mount is experimentally measurable, but it is hard to measure the amount of heat released from the mount [1, 2]. This phenomenon is complex and difficult to measure in test conditions because it is known that a part of the generated energy is distributed to the mount elements, working fluid (in case it exists), etc. In addition, the nature of heat transfer from the mount to the environment is very complex. Heat transfer is carried out by convection, as dominant, also by conduction and radiation [2].

From the point of maximal cooling, proper selection of mounts requires a comprehensive analysis of the transformation of mechanical energy into heat. Method of transformation of mechanical energy into heat is largely determined by the mount design. It is not possible to influence directly on the conduction of heat and radiation from the mount. It is necessary to increase the influence of the heat transfer by convection from the mounts to the surrounding environment, as dominant appearance. The idea is to utilize convection flow of air around the mount with the least complexity of the structure. In practice, this solution is rarely used, but can be applied. Making some kind of air deflectors on the elements of the body should increase the effect of convective heat transfer to the environment.

Note that the objective of this study was not to analyse the cooling of the powertrain mounts, but only thermal load to which it is exposed. Therefore, it was deemed expedient to analyse the heat which is obtained by converting mechanical work into heat energy per time unit. Mechanical work in powertrain mounting system was calculated by use of mechanical powertrain model, which will be discussed below.

This research does not take into account thermal load caused by the engine combustion process, which is transmitted to the mounts, because this load does not arise from vibration. To be more specific, only thermal stresses that result from mechanical work of elasto-damping forces in mounts were analysed.

As it is known, conventional mounts consist of rubber-metal elements, with the damping effect coming from the tire hysteresis, and further having hydraulic damping of the oil flow.

It is well known that vibrations of the power train are investigated in detail [8,14]. However, there are very rare cases of research of the power train mounts thermal loads. Classic power train mounts consist of metal-rubber elements, with damping coming from rubber hysteresis and additional hydraulic damping from the oil streaming through them. Considering the presence of classic – mechanical and hydraulic power train mounting systems in modern trucks, analysis of power train mounting thermal loads of a FAP 1213 vehicle [8] was conducted. Bearing in mind that cooling of the powertrain mounts is very complicated, the experimental verification of the thermal powertrain mounts will be done in next investigation.

2. MODEL OF POWERTRAIN

In the literature, depending on the task to be solved, we can find different models of mechanical powertrains. From [9,10-15] it is well known that during the analysis of the problem of transferring dynamic loads from the powertrain to the freight vehicle frame, vibration of cab and cargo box may be neglected. To be more precise, the analysis shall include only the vibration motion of the vehicle powertrain and related
excitations of the vehicle frame For the sake of illustration, Fig. 1. shows scheme of powertrain suspension of a FAP freight vehicle [8].

Powertrain, as a rigid body in space, has six degrees of freedom (three translations and three rotations) [9,10, 11-13, 15-20]. In order to describe its spatial movement, a Cartesian coordinate system with the origin in the CG of the powertrain will be adopted, initially in steady state. One of axis is parallel to the axis of the engine crankshaft, while the other two are perpendicular to the first one [9,21-24]. The adopted coordinate axes will be called CG geometric axes, and they are, at the same time coinciding with the axes which are often used by powertrain manufacturers [8,9]. It should be pointed out that the use of geometric CG axes leads to the application of centrifugal moments of inertia, but in order to simplify the analysis, it was found appropriate to introduce the assumption that they are, at the same time, the major axes of inertia. Powertrain performs spatial vibrations as a result of the excitations from the frame (originally coming from the road unevenness and frame vibration as an elastic system), as well as uneven engine running, inertial forces and torques of rotating masses, etc.

For describing the vibrational motion of the powertrain, two coordinate systems will be adopted [9, 14], see Fig. 2:

- stationary, and
- movable, fixed to the powertrain.

The motion of the powertrain in the space is defined with three coordinates X, Y, Z, and rotation of the powertrain around CG (as a fixed point) is defined by three angles: φ-roll, θ-pitch and ψ-yaw.

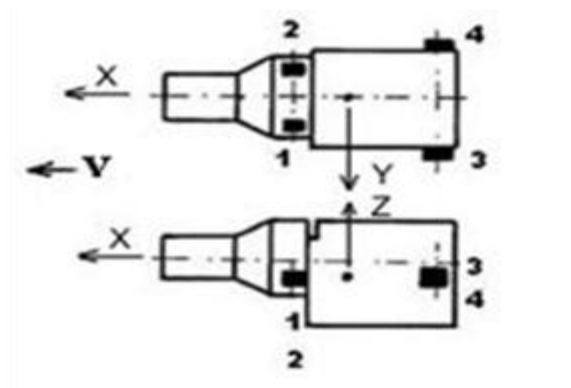

Fig. 1 Scheme of powertrain suspension of the observed freight vehicle

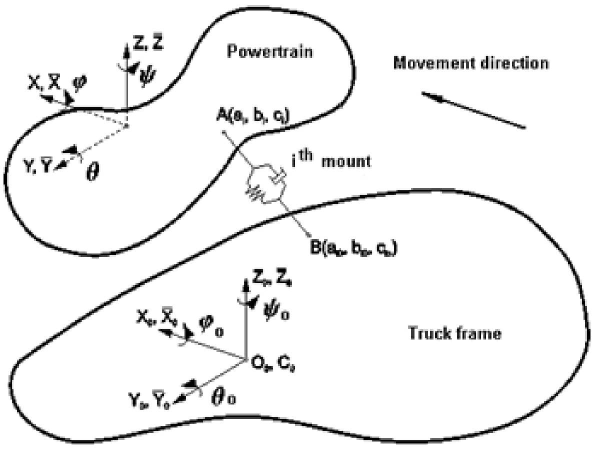

Fig. 2 Coordinate systems introduced to describe motion of the powertrain

Newton-Euler equations are applied to describe spatial motion of the powertrain [11].

$$M\ddot{X}_C = \sum X_i \quad (2)$$

$$M\ddot{Y}_C = \sum Y_i \quad (3)$$

$$M\ddot{Z}_C = \sum Z_i \quad (4)$$

$$I_u\dot{\omega}_u - I_{uv}\dot{\omega}_v - I_{uw}\dot{\omega}_w + (I_w\omega_w - I_{uw}\omega_u - I_{vw}\omega_v)\omega_v - (I_v\omega_v - I_{uv}\omega_u - I_{vw}\omega_w)\omega_w = \sum M_u^{F_i} \quad (5)$$

$$I_v\dot{\omega}_v - I_{uv}\dot{\omega}_u - I_{uw}\dot{\omega}_w + (I_u\omega_u - I_{uv}\omega_v - I_{uw}\omega_v)\omega_w - (I_w\omega_w - I_{uv}\omega_u - I_{vw}\omega_v)\omega_u = \sum M_v^{F_i} \quad (6)$$

$$I_w\dot{\omega}_w - I_{uv}\dot{\omega}_u - I_{vw}\dot{\omega}_v + (I_v\omega_v - I_{uv}\omega_u - I_{vw}\omega_w)\omega_u - (I_u\omega_u - I_{uv}\omega_v - I_{uv}\omega_w)\omega_v = \sum M_w^{F_i} \quad (7)$$

where:

M - powertrain mass,

- $\ddot{X}_C, \ddot{Y}_C, \ddot{Z}_C$ –projections of powertrain accelerations on axes of the moving coordinate systems,
- X_i, Y_i, Z_i – projection of excitation forces and reactions of respective powertrain mounts,
- I_u, I_v, I_w, I_{uv}, I_{uw}, I_{vw} – moments of inertia for the respective coordinate axes,
- $\omega_u, \omega_v, \omega_w, \dot{\omega}_u, \dot{\omega}_v, \dot{\omega}_w$ -the angular velocities and accelerations for axes u, v and w,
- M_u , M_v, M_w – projections of torques of the excitations, engine torque, and mounts of the powertrain for axes u, v and w.

For the sake of simplification, it will be assumed that the angular motions of the powertrain are small [14] what may enable the application of the relations [9, 11, 18, 21-23]:

$$\omega_u = \dot{\varphi}\ \omega_v = \dot{\theta}\omega_w = \dot{\psi}$$

In order to calculate forces in the mounts, it is necessary to define their deformations and deformation velocities. Bearing in mind the adopted coordinate systems, motion of any point on the powertrain is given in the equation in a matrix form [9-11]:

$$r = r_C + Lr_A \quad (8)$$

where:

$$r_C = \begin{bmatrix} X \\ Y \\ Z \end{bmatrix} \quad (9)$$

$$r_A = \begin{bmatrix} a_i \\ b_i \\ c_i \end{bmatrix} \quad (10)$$

$$L = \begin{bmatrix} c2c3 & s1s2c3 & s1s3 - c1s2c3 \\ -c2s3 & -s1s2s3 & c1s2s3 \\ s2 & -s1c2 & c1c2 \end{bmatrix} \quad (11)$$

In the expression (11) ***s*** and ***c*** are cosine and sinus of the respective angles, respectively.

For small angles the following expression can be written:

$$L = \begin{bmatrix} 1 & \psi & -\theta \\ -\psi & 1 & \varphi \\ \theta & -\varphi & 1 \end{bmatrix} \quad (12)$$

Similarly, and bearing in mind the low vibration of a vehicle, the expression can be written:

$$r_0 = r_{c0} + L_0 r_B \quad (13)$$

$$r_{c0} = \begin{bmatrix} X_0 \\ Y_0 \\ Z_0 \end{bmatrix} \quad (14)$$

$$r_B = \begin{bmatrix} a_{Bi} \\ b_{Bi} \\ c_{Bi} \end{bmatrix} \quad (15)$$

$$L_0 = \begin{bmatrix} 1 & \psi_0 & -\theta_0 \\ -\psi_0 & 1 & \varphi_0 \\ \theta_0 & -\varphi_0 & 1 \end{bmatrix} \quad (16)$$

where X_O, Y_O and Z_O - coordinates relative to the origin of the movable coordinate system of the vehicle frame.

Since the dimensions of the mounts are small relative to the powertrain, the following can be written:

$$r_A \approx r_B \to r_i = \begin{bmatrix} a_i \\ b_i \\ c_i \end{bmatrix} \quad (17)$$

Deformation of the i-th mount is defined by the following matrix equation [9, 11]:

$$\Delta_i = r - r_0 = (r_c - r_{c0}) + (L - L_0)r_i \quad (18)$$

Substituting the corresponding expressions and rearranging leads to the expressions:

$$\Delta X_i = X - X_0 + b_i(\psi - \psi_0) - c_i(\theta - \theta_0) \quad (19)$$
$$\Delta Y_i = Y - Y_0 + a_i(\psi - \psi_0) + c_i(\varphi - \varphi_0) \quad (20)$$
$$\Delta Z_i = Z - Z_0 - a_i(\theta - \theta_0) - b_i(\varphi - \varphi_0) \quad (21)$$

Mount deformation velocities can be obtained by differentiating the previous expressions:

$$\Delta\dot{X}_i = \dot{X} - \dot{X}_0 + b_i(\dot{\psi} - \dot{\psi}_0) - c_i(\dot{\theta} - \dot{\theta}_0) \quad (22)$$
$$\Delta\dot{Y}_i = \dot{Y} - \dot{Y}_0 + a_i(\dot{\psi} - \dot{\psi}_0) + c_i(\dot{\varphi} - \dot{\varphi}_0) \quad (23)$$
$$\Delta\dot{Z}_i = \dot{Z} - \dot{Z}_0 - a_i(\dot{\theta} - \dot{\theta}_0) - b_i(\dot{\varphi} - \dot{\varphi}_0) \quad (24)$$

In the literature [19, 20] there are many mathematical models of powertrain mounts, which are essentially more or less complicated. Bearing in mind the objective of this study was a comparative analysis of thermal loads in mounts, it was estimated to be acceptable to apply more simplified expressions for approximation of forces in the mounts.

Forces in elastic mounts are assumed in the form [9]:

$$F_{ci} = c_{i1}\Delta_i + c_{i2}\Delta^2 + c_{i3}\Delta^3 \quad (25)$$

where:

- c_i, c_{i1}, c_{i2} and c_{i3} – stiffness coefficients, and
- Δ_i – relative deformation of the mount.

Damping forces in the mounts are assumed in the form [9]:

$$F_{ai} = k_{i1}\dot{\Delta} + k_{i2}\dot{\Delta}^2 sign(\Delta) \quad (26)$$

where:

- k_{i1}, k_{i2} –damping coefficients,
- $\dot{\Delta}$ - relative velocity of the mount deformation, and
- *sign* - respective mathematical function.

Powertrain vibrations also depend on the unbalance of the running engine (torques and inertial forces).

In the specific case of a four-stroke four-cylinder in-line diesel engine was used with a crankshaft whose kranks lie in the same plane (the angle 180°). Forces occur in the reciprocating mechanisms [24] that drive the piston (gas forces and inertial forces of the piston group), and centrifugal and tangential forces acting on the movable bearing of the crank shaft knee. When balancing the inertial forces of the piston group mass (in the ideal case, if a force is developed into Fourier series), the inertial forces of the second and higher orders stay unbalanced. It is noted that in case when there are differences in the masses of the piston groups for each cylinder, there are also unbalanced forces of the first order (in this considered case, the masses have been equal to each other).

Assuming that harmonics of a higher order can be ignored, the unbalanced inertial force of the observed engine can be expressed in the form [14, 24]:

$$F_{in} = 4m_r\,\omega^2\lambda \cdot \cos 2\omega t \quad (27)$$

where:

- m_r – reduced mass of the piston group,
- r – radius of the crankshaft crank,
- ω - angular velocity of the engine crankshaft,
- λ - crankshaft radius and connecting rod ratio, and
- t – time.

Using basic knowledge of the vector theory and postulates of statics, and taking into account Figs 2 and 3, the torque resulting from the inertial force is defined by the equation (8) [9]:

$$\overrightarrow{M_{Fin}} = \begin{bmatrix} \overrightarrow{u_0} & \overrightarrow{v_0} & \overrightarrow{w_0} \\ a_E & b_E & c_E \\ -F_{in}sin\gamma & \gamma & F_{in}cos\gamma \end{bmatrix} \quad (28)$$

wherein the expressions in (9) are harmonized with the Fig. 3.

Centrifugal force is partially balanced the counter-weights, or by use of other methods, of which there will be no more to say, but readers are advised to see [24].

Tangential force causes engine torque, which, due to its variation has a variable value (unevenness is partially reduced by the engine flywheel) [24].

In the absence of precise information, it will be assumed that the torque acting on the powertrain can be described by expression [9]:

$$M = -M_e \cdot i_0 \cdot i_m\,(0.95 + 0.1 \cdot rnd) \quad (29)$$

where:

- M_e – engine torque,
- i_0 – driving axle ratio,
- rnd – random numbers uniformly distributed in the interval [0,1].

The vibration of powertrain impacts the vibration of the vehicle frame, which is of random nature [9,14]. Bearing in mind that the complexity of the vehicle spatial model exceeds the needs of this study, it is estimated to be appropriate in this specific case not to use the frame vibration excitations based on the vehicle model, but to use already adopted broadband excitation functions in the following form:

$$excitation = \max\,(rnd - 0.5) \tag{30}$$

where:

- max = 0.01 m, rad
- *rnd*- has the same meaning as mentioned in description of the engine torque.

Projections of the generalized forces include all the components of forces and torques of the corresponding mount in the direction of the observed axes (for mounts 1 to 4), the engine inertial forces and torques and unbalanced inertial forces, as well as the active suspension force which, does not have the corresponding torque due to the assumption that it acts in the powertrain C.G.

Bearing in mind the expressions (1-27), differential equations of motion of the powertrain can be written in the form:

$$M\ddot{u} = \sum X_i \tag{31}$$

$$I_u\ddot{\varphi} = \sum M_F^u \tag{32}$$

$$M\ddot{v} = \sum Y_i \tag{33}$$

$$I_v\ddot{\theta} = \sum M_F^v \tag{34}$$

$$M\ddot{w} = \sum Z_i \tag{35}$$

$$I_w\ddot{\psi} = \sum M_F^w \tag{36}$$

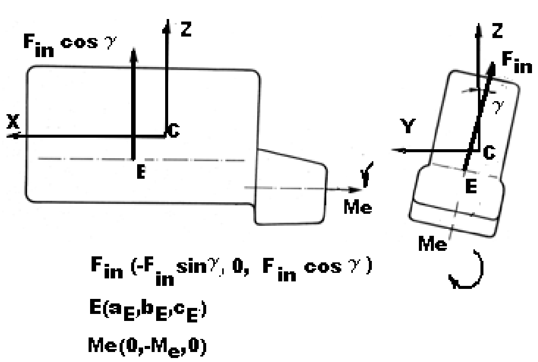

Fig. 3 Inertial force and engine torque

where:

F_{in} – resulting inertial force of piston group,

M_e – engine torque,

E – acting point of the resulting force and its coordinates related to the moving coordinate system,

γ - engine inclination (in the observed case 0°).

3. THERMAL LOAD OF MOUNTING SYSTEM

Due to the relative motion of sprung and unsprung masses, mechanical work is being done in mounts, which is equivalent to the amount of produced heat, Q [2]. Mechanical work (the amount of heat) is defined by the expression [2-7]:

$$A = \int_0^s F_m(t)\cdot dz_{rel} = \int_0^T F_m(t)\cdot \dot{z}_{rel}\cdot dt\,, \tag{37}$$

where:

- $F_m(t)$ - is elasto-damping force in the mount,
- z_{rel} – is deformation of the mount, eqns (19 - 21),
- $\dot{z}_{rel}$ - relative velocity of deformation (time derivatives of displacements) given by eqns (22-24) and
- t – is time.

Mechanical power, $P(t)$, that is equivalent to heat flux, is the first derivative of mechanical work with respect to time:

$$P(t) = \frac{dA}{dt} \tag{38}$$

Average power, P_{av}, is given by expression [10]:

$$P_{av} = \frac{A}{T} = \frac{1}{T}\int_0^T P(t)\cdot dt\,, \tag{39}$$

where T is the monitoring period. Average power turns into heat, with dominant convection [10]:

$$P = \alpha\cdot S\cdot \Delta\tau\,, \tag{40}$$

where:

- α - is heat transfer coefficient,
- S - convection area and
- $\Delta\tau$ - is temperature difference between the mounts and surrounding air.

As already noted, the analysis of heat transfer from the mount has not been carried out in the paper, because the values of α_i and S_i are not known, and it requires very extensive experimental studies to determine these values, which will certainly be the subject of special attention in the future.

Since all four power train unit mounts had the same characteristics, the analysis of collective thermal loads was conducted.

4. NUMERICAL SIMULATION AND ANALYSIS OF RESULTS

On the basis of the expressions (1-40), it can be seen that the differential equations describing the spatial vibration of the vehicle powertrain are non-linear and should be solved numerically, by use of the method Runge-Kutta, and by use of a software developed by the authors, in Pascal. The integration was carried out with the time increment of 0.0001 s, in 524288 points. This enabled reliable analysis in the domain 0.019-5000 Hz [25]. It is obvious that the domain is much broader than range of excitations from the running engine and the entire vehicle powertrain. Integration of differential equations is carried out in case of using conventional and hydraulic mounts.

The parameters of the observed vehicle and its powertrain given in Tab. 1, and the coordinates of the connection points (mounts) in Tab. 2.

Table 1. Characteristic parameters of the vehicle FAP 1213 and its powertrain

Maximal engine power, kW	100
Maximal engine speed, min^{-1}	2600
Maximal engine torque, Nm	428
Engine speed at max. engine torque, min^{-1}	1300
Vehicle mass, kg	12000
Powertrain mass, kg	1680
Moments of inertia Ix/Iy/Iz, kgm^2	85/35/72
Driving axle gear ratio, -	3.83
Transmission ratio in direct gear, -	1

Table 2. Engine mounts coordinates

Mount position →	a, m	b, m	c, m
Mount 1	0.5	0.4	0.1
Mount 2	0.5	-0.4	0.1
Mount 3	-0.5	0.4	0.1
Mount 4	-0.5	-0.4	0.1

Mechanical and hydraulic mounts have identical stiffness in the direction of the axes X, Y and Z, which are determined by the tests performed in FAP [8], as shown in Table 3.

Table 3. Stiffness of the applied conventional and hydraulic mounts

	c_{i1}, N/m	c_{i2}, N/m^2	c_{i3}, N/m^3
X	1200000	250000	60000
Y	1200000	250000	60000
Z	1200000	250000	60000

In the absence of accurate data, damping characteristics of the mounts are approximately defined on the basis of the support stiffness and mass carried by those [11], and are given in Tab. 4.

Table 4. Assumed damping characteristics of the mounts

	k_{i1} (mechanical/ hydraulic), Ns/m	k_{i2} (mechanical/ hydraulic), Ns^2/m^2
X	620/62000	1/100
Y	620/62000	1/100
Z	620/62000	1/100

In this paper it is assumed that the frame has six identical excitations in the time domain (eq. 30). For the illustration, vertical vibration excitations of the powertrain, derived from the frame vibration are shown in Fig. 4. It is obvious that the excitations are very dynamic, and so large thermal load of the powertrain mounts should be expected.

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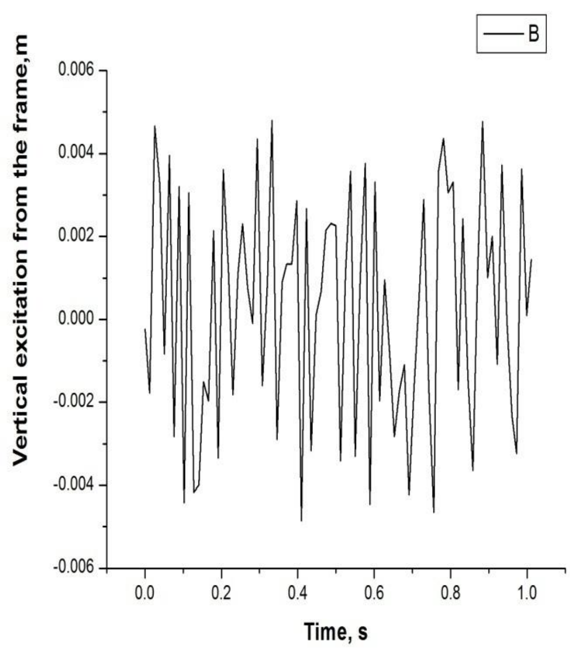

Fig. 4 Illustration of the frame excitation

Results of the dynamic simulation are shown in the time domain. To illustrate, Fig. 4 gives the variation of displacements in case of use of hydraulic mounts. Analysis of the data concerning classical and hydraulic mounts, partially shown in Fig. 5, in the time domain, shows that the type of the applied mounts affects the character and the amplitude of the monitored displacement of the powertrain.

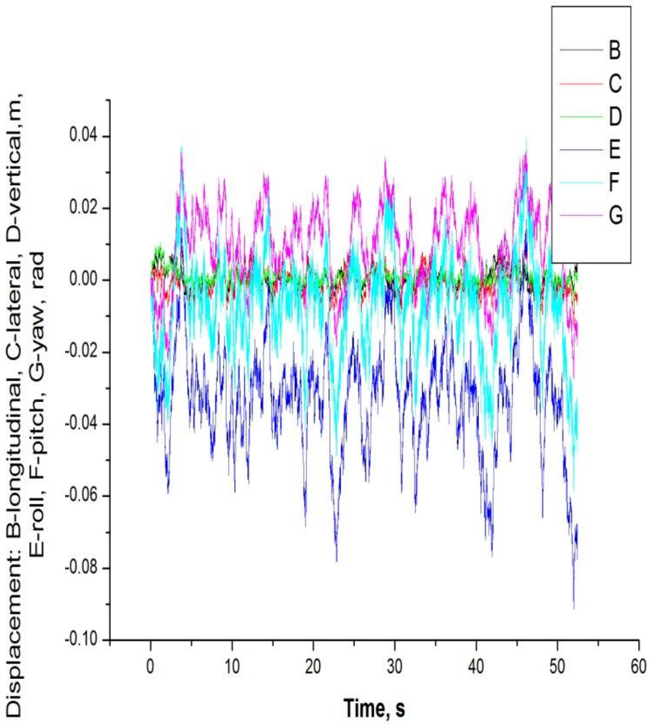

Fig. 5 Powertrain displacements in case of hydraulic mounts

Summarized thermal loads (for all forces and torques components and for all four mounts) are calculated using the Eqs (2-40) and the results are shown in Figs 6 and 7, wherein the values in Fig. 6 for the amount of heat are given in the log scale.

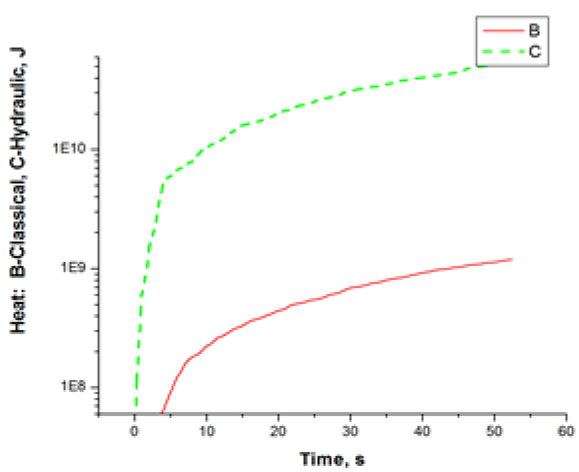

Fig. 6 Quantity of heat (mechanical work) depending on the type of mount

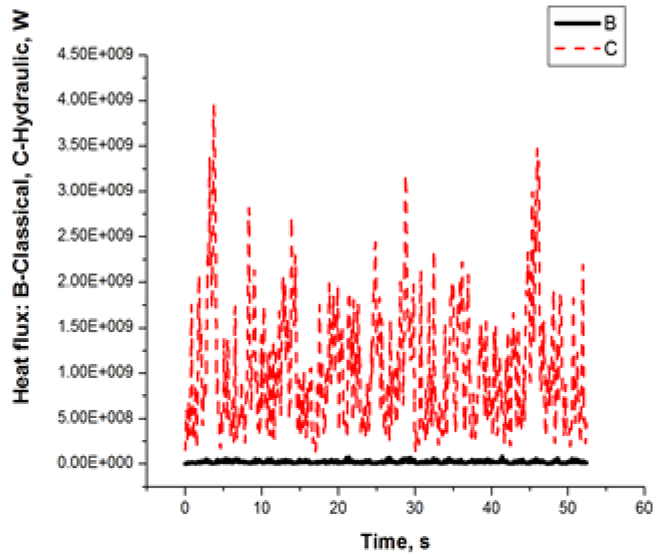

Fig. 7 Heat flux depending on the type of mounts

Table 5. Statistical data on heat flux

Mounts	Min, W	Max, W	Mean value, W	Effective value, W
Classical	2.494 10^2	2.973 10^8	2.271 10^7	3. 438 10^7
Hydraulic	2.570 10^2	1347 10^{10}	1.038 10^9	1. 609 10^9

The analysis of Fig. 6 shows that mechanical mounts suffer from approximately 45 times lower thermal loads then the hydraulic (for 52 s, classical about 1.19x109 J, and hydraulic about 53.69x109 J). This is understandable bearing in mind that hydraulic mounts, in addition to hysteresis in the rubber, have the additional fluid flow within the mount. Fig. 6 shows that the amount of heat increases with time and that in the absence of cooling, they would experience a degradation of the shape and characteristics.

Fig. 7 shows how the heat flux depends on the time. It is obvious that it changes stochastically over time, so for the sake of the analysis, it is necessary to calculate some characteristic statistical values given in Tab. 5 [25,26].

Analysis of the data from Tab. 5 shows that the heat flux is much higher in hydraulic, that in the conventional suspension of powertrain. This indicates a greater possibility of degradation of characteristics of hydraulic mounts compared to the classical. Therefore they are bigger in size than the classical. It should be pointed out that such high values of thermal flow are a result of rigorous excitations of vehicle frame that were used in the simulation. In practice, they are significantly milder, and consequently the real thermal load of the powertrain mounts. However, the research is carried out to determine the ratio of the thermal load of classic and hydraulic supports, with by use of models, so the obtained results can be adopted only as an orientation, which is acceptable in the initial design phase of the truck. It should be mentioned that hydraulic mounts have significantly better performance when vibrations of the powertrain are in question [9].

5. CONCLUSIONS

The research shows that mechanical models can perform analyses of the impact of powertrain supports features on their thermal loads. Analyses showed that the hydraulic supports are significantly more exposed to heat load than the conventional. Bearing in mind the thermal loads, as well as a more complex structure, the less frequent application of hydraulic supports at the freight vehicles powertrain is expected. In order to verify the obtained results the experimental research should be carried out in the future.

Based on the research results the following conclusions can made:
The proposed model of powertrain can be used for the simulation of thermal load of freight vehicle powertrain mounts,
The hydraulic mounts of powertrain are subjected to a significantly greater thermal load than the conventional, and
Bearing in mind the extent of the existing thermal loads, as well as a more complex design, the application of hydraulic powertrain mounts is understandably less frequent in freight vehicles powertrain suspension.

Nomenclature
z- vertical vibration, [m]
A – work, [J]
I_u, I_v, I_w - moment of inertia, [kgm^2]
f – frequency, [Hz],
t – time, [s],
Q - quantity of heat, [J]
Fm -force in the shock absorber, [N]

Greek symbols:
φ- roll, [rad]
θ- pitch, [rad]
ψ- yaw of the powertrain, [rad]
x – reference for x-axis, ***y*** – reference for y-axis, and ***z*** – reference for z-axis

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