Vibration load of transmission units at vehicle’s motion over different roads

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Abstract. Vibrations of operating transmission units caused partly by the vehicle’s motion over the road and partly by running engine are considered in this article. Special attention is paid to the oscillations caused by the vehicle wheels interaction with the road surface. Spectral density of the road microprofile is known to be one of its primary characteristics. Its value affects the road action force on the transmission. A range of questions of the road surface force impact causing vibration load determined by engine torsional oscillations is considered in this article.

Introduction
Multi-purpose wheeled vehicles require high level quality of operation and reliability. Thorough development tests aimed to improve the construction, and transition from planned maintenance to the actual technical state maintenance are a reserve of improving reliability. All these require a wide use of control and diagnosis equipment. Consequently, it is necessary to single out characteristics allowing to accurately determine changes of the vehicle technical state at minimum expense. Specific operating conditions of the researched machines make it necessary to specify the whole range of design, test and development methods especially in ensuring optimal vibroacoustics. Thus, it is feasible to apply the design concept of transmission units with preset vibroacoustic parameters taking into account the road microprofile impact [1].

Road microprofile impact on transmission vibration load
When calculating transmissions vibration load under the influence of surface irregularities (micro-roughnesses) the following assumptions are made:
- the road profile is symmetrical with respect to longitudinal vertical plane passing through the vehicle axis of symmetry;
- roadwheel contact is pinpoint;
- roughness length impact on the exciting force is considered by the microprofile adjustment;
- the vehicle speed mathematical expectation is constant and equal to average speed;
- the equivalent dynamic system can be presented by linear analytical model;
• non-driving front axle run-over the roughness does not cause substantial oscillations in the transmission.

The analytical dynamic model applied to 4 X 2 vehicle is shown in (Figure 1). In this model the suspension guide unit provides the axle and transmission relative shift at their vertical movements according to the law:

\[ x_1 - x = \eta (\xi - z), \]

where \( \eta \) - kinematic connection coefficient of vertical and longitudinal movements of the vehicle transmission units \((-0.1 \leq \eta \leq 0.1)\).

This coefficient takes into account vertical and longitudinal oscillations of driveshaft length as the vehicle runs over different irregularities.

The driveshaft vertical and longitudinal movements are limited by the suspension kinematics and consequently can be taken into account by the corresponding rigidity and damping coefficients of its parts. Longitudinally, high-frequency movements are considered the most vibrationally-hazardous. They appear in nonsteady modes of high-speed motion over the microprofile with small amplitude irregularities. In general, they can be described by the following harmonic dependence:

\[ \phi = A(x) \sin(\omega t + \varphi), \]

where \( A(x) \) - shaft drive longitudinal oscillations amplitude.

The vehicle sprung mass \( m \) and the front axle mass take part in lengthwise movement along coordinate \( x \), and in vertical movement along coordinate \( z \) - only the front axle mass \( m_2 \),

\[ m = m_1 - m_2 + \sum r_{\text{K}i} J_{\text{K}i}; \]

\[ m_2 = \frac{m_a}{L}; \]

where \( a \) - distance from the front axle to the gravity center; \( L \) - vehicle base; \( \sum J_{\text{K}i} \) - total inertia moment of the front axle wheels.

The system parameters are shown in Figure 1:

\( J_1 \) - reduced inertia moment of rotating masses of engine, clutch, gear box, part of shaft drive;
\( J_z \) - reduced inertia moment of part of shaft drive masses, rotating parts of drive axle, wheel assemblies and wheels;

\( m_{3h} \) - drive axle unsprung mass;

\( c_{ul} \) - reduced rigidity of shaft drive and transmission gear shafts;

\( c_{ur} \) - reduced drive axle tangential tyre stiffness;

\( C_rK_r \) - reduced rigidity and damping coefficient of drive axle suspension;

\( C_{ll} \) - drive axle radial tyre stiffness.

The system mass movement is described in generalized coordinates: \( x, z, \dot{x}, \ddot{x}, \varphi, \dot{\varphi}, \ddot{\varphi}, \xi, \dot{\xi}, \ddot{\xi} \), where \( \varphi, \dot{\varphi} \) - angles of rotation of shaft drive rotating masses and gear box with moments \( J_1 \) and \( J_2 \).

The system equations of motion can be described as follows:

\[
\begin{aligned}
\dot{\varphi}_1 &= \frac{r_x T}{J_2} - \left( \frac{1}{J_1} + \frac{1}{J_2} \right) M_{12}, \\
\dot{\xi} &= -b_P \ddot{\varphi}_1 + b_R, \\
\ddot{x} &= c_R \dot{\varphi}_1 - c_P \ddot{\varphi}_1 + b_T, \\
\ddot{\xi} &= a_P \ddot{\varphi}_1, \\
\dot{\varphi}_1 &= \frac{M_{12}}{J_1}.
\end{aligned}
\]  

(1)

Constraint reactions equations are written as follows:

\[
\begin{aligned}
P_p &= c_P (\dot{\varphi}_1 + \dot{\varphi}_1) + k_P (\ddot{\varphi}_1 + \ddot{\varphi}_1); \\
R &= C_{ll} (q - \dot{\xi}); \\
T_0 &= T + \bar{R} \dot{q} + \dot{R}f; \\
\bar{R} &= (m_2 + m_{3h})g; \\
T &= \frac{C_{ll}}{r_{ks}} \left( \frac{x}{r_{ks}} - \varphi_{12} - \varphi_1 \right) + \frac{k_{ll}}{r_{ks}} \left( \frac{\dot{x}}{r_{ks}} - \dot{\varphi}_{12} - \dot{\varphi}_1 \right); \\
M_{12} &= c_{ll} \varphi_{12} + K_{ll} \dot{\varphi}_{12}.
\end{aligned}
\]

Coefficients in constraint reactions equations are written in the following form: \( a_P = \frac{(m + m_{3h})}{A} \); \( b_{\varphi} = \frac{m_{\varphi}(m + m_{3h})}{m_{3h} A} \); \( b_1 = \frac{m_{\varphi}}{m_{3h} A} \); \( b_3 = \frac{m_{3h} m_{\varphi}}{m_{3h} A} \); \( b_1 = \frac{m_{3h} m_{\varphi}}{m_{3h} A} \); \( A = 2 \eta^2 (m_2 + m_{3h})^2 + m_{3h} m + m_{3h} \). Thus, at known values of mass-inertia characteristics of undercarriage unsprung parts, the vehicle geometric characteristics and elastic-rigidity properties of its suspension it is possible to calculate coefficient values in constraint reactions equations. Substituting these in constraint reactions equations, using system (1), it is possible to determine vertical and torsional oscillations character of sprung and unsprung masses as well as transmission parts. Applying up-to-date software to the obtained diagrams: MathCAD, Matlab and others, and carrying out their spectrum analysis thereafter it is possible to obtain spectral densities of vibration accelerations on the driver’s seat and on construction elements.

**Transmission parts vibration load**

The vehicle motion over any bearing surface is carried out by traction forces, realized by the propulsion unit according to the driving conditions [2]:

\[
\varphi, G_{aw} \geq F_{t_0} \geq F_f + F_w
\]

(2)
where $\varphi_r$ - traction coefficient, $G_{an}$ - the vehicle adhesion weight, $F_{d}$ - drag force, realized by the vehicle wheels, $F_{f}$ - rolling resistance force, $F_{w}$ - air resistance force.

In case of the vehicle motion at the speed of not more than $15 \frac{m}{c}$ over the hard surface having no skidding in the contact patch, air resistant forces can be disregarded because of their smallness. Rolling resistance force in general can be calculated by formula:

$$F_{f} = \sum_{i=1}^{n} f \cdot R_{z}$$  \hspace{1cm} (3),

where $n$ - number of wheels, $f$ - rolling resistance generalized coefficient; $R_{z}$ - bearing surface normal reaction on $i$ wheel.

Load force, acting in this case on tooth gears of gear boxes from resistance forces is calculated by formula:

$$F_{H_{f}} = \frac{F_{f} \cdot r_{d} \cdot i_{0}}{r_{2} \eta_{tr}}$$  \hspace{1cm} (4),

where $F_{f}$ - rolling resistance force; $r_{d}$ - wheel dynamic radius, $i_{0}$ - final drive gear ratio, $r_{2}$ - driven gear pitch circle radius, $\eta_{tr}$ - transmission reverse efficiency.

Drag force source $F_{s_{x}}$, realized by wheels, is a torque output, generated by the engine, its mathematical model is presented in the form of [3]:

$$T_{e}(t) = \overline{T_{e}} + \sum_{k=1}^{l} \sum_{l=1}^{l} T_{e} \cdot \sin \left( \omega_{t} \cdot t + \psi_{r} - \beta_{j} \right)$$  \hspace{1cm} (5),

where $T_{e}$ = $\overline{\sum_{t=1}^{n} T_{e} - T_{e}}$ - average torque output, developed by the engine; $\overline{T_{e}}$ - average torque output value, developed in one engine section; $T_{e}$ - shaft rotation moment of resistance in an engine section; $k$ - section number in working order; $l$ - number of sections; $\beta_{j}$ - crank angle at a time period between the start of power stroke in the first section and the start of power stroke in $j$ - section of the engine.

Gear teeth interaction in gear boxes is affected by exciting force, composed of engine perturbation actions and rolling resistance forces, loading the transmission and engine. Total exciting force can be found by formula:

$$F(t) = \frac{T_{e}(t) \cdot T_{H_{f}}}{r_{1}} = \frac{T_{H_{f}}}{r_{2}}$$  \hspace{1cm} (6).

To solve tasks of units design with preset vibroacoustic features it is required to build dynamic models of the vehicle transmission and tooth gear. It is necessary to study vibration processes in the transmission as a whole and in its separate units. For this case, the diagram of tooth gear equivalent dynamic model is shown in (Figure 2).

It is possible to describe dynamic motion of interacting helical gears on qualitative level by analytical mechanics equation written in Lagrange form of the second kind. For the system in question having 12 degrees of freedom, when determining kinetic and potential energy as well as Rayleigh dissipation function, these equations are written as differential equations of the second order and their classical solutions – as an equation system of harmonic oscillations. From the physical standpoint this system is a superposition of oscillations, emanating from the teeth contact zone.

Coefficients of rigidity, damping and mass-inertia characteristics of tooth gears are usually determined in the course of specific experiments. The example of calculation results is provided by a dependency diagram of the spectral peak envelope of vibroacoustic signal of the third speed gears in...
«Gazelle» vehicle, preserving and violating lubricant in the contact patch depending on the load torque in Figure 3.

Figure 2. Diagram of equivalent dynamic model of helical gear

Figure 3. Diagram of the spectral peak envelope of vibroacoustic signal of the third speed in «Gazelle» vehicle preserving (1) and violating lubricant in the contact patch (2) load torque values in gear: a) \( 0.25 T_{\text{max}} \); b) \( T_{\text{max}} \).
Experimental verification of numerical calculations is carried out on special test benches, comprising a power unit and a hydraulic or any other similar loading device simulating resistance to motion. Measuring complexes to record operational vibroacoustic characteristics are manufactured in a large variety by companies RFT, Brue & Kjaer etc. Sensors are mounted in maximum emission places. For gear boxes these are spots on the body near bearing units. Research of the impact of the amount of lubricant in the contact patch when the «Gazelle» gear box is in operation on the third speed was carried out on such test bench by a measuring complex having characteristics similar to Brue & Kjaer, at the torque load value of 25% from the maximum. The diagram, obtained as a result of the experiment is shown in (Figure 4).

**Figure 4.** Diagram of spectrum of vibroacoustic signal of the third speed in «Gazelle» vehicle, obtained in the course of experiment, in the violation of lubricant in the contact patch for load torque values in gear, equal to $T_{\text{max}}$.

**Conclusion**

Vibroacoustics of transmission units operation, their signal being processed in the appropriate manner, are informative control values; calculation and experimental data can be used for development and optimization of design solutions, providing the designed units with vibroacoustics values within the prescribed limits and the vehicle as a whole – with the acceptable smooth ride features.

**References:**

[1] Leliovsky K Y, Belyakov V V, Ogorodnov S M, Bushueva M E 2006 Applying vibroacoustic diagnostics when designing the vehicle power train units *Izvestiya Ijon RF* 16 44-46
[2] Belyakov V V and Kulyashov A P 2004 Cross-country transport and technological vehicles *N Novgorod: TALAM* 960
[3] Weitz V L 1976 Machine assemblies dynamics with internal combustion engines *L.: Mashinostroenie* 384
[4] Leliovsky K Y, Belyakov V V and Ogorodnov S M 2008 Improving «Gazelle» gear boxes design according to their operational vibroacoustics *News of Higher educational Institutions. Series «Mashinostroenie»* 8 49 - 56