Engine-and-transmission installation frequency analysis of a wheeled forestry tractor

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Abstract: The article provides a frequency analysis of an engine-and-transmission installation of a wheeled forestry tractor. It can be used for machine-building enterprises of the forest complex. During the wheeled forestry tractors operation in the engine-and-transmission installation, resonance phenomena may occur when the internal frequencies of the system coincide with the external disturbance. The solution of the problem was carried out on the basis of the developed equivalent dynamic circuit, which was described by the differential equations of torsional vibration. As a result of solving these equations, the values of the engine-and-transmission installation natural frequencies were obtained. A comparative analysis of the frequencies obtained with regard to the damping properties of the system has been made, and possible resonance zones have been identified. The information obtained allows selecting the optimal damping indicators for the wheeled forestry tractor engine-and-transmission installation operation in a resonant-free mode.

Keywords: engine-and-transmission installation, resonance, torsional vibrations

Introduction.
Changes in the transmission parameters of a wheeled forestry tractor and the introduction of new technical solutions affect the inertial and stiff, damping properties of the transmission, and, consequently, its eigen frequencies values. As the researchers note, the dynamic component of the tangential traction force is the result of self-oscillations, has a certain dominant frequency, which can cause resonant oscillatory processes of the “engine-transmission-tractor with a bundle” system. The oscillatory system eigen frequencies can vary over a wide range when installing elements with a high compliance coefficient. In connection with the change in the transmission parameters or the new technical solutions introduction, for example, damping elements, it is necessary to conduct a frequency analysis of the oscillatory system and develop measures that exclude system’s resonant modes.

The studies analysis of transport and technological machines [1, 2, 3, 4, 6] shows that the resonant modes of the lugs stimulus with the bearing surface may arise at the third natural frequency of the dynamic system "engine - transmission - tractor with a bundle” natural oscillations. Therefore, an equivalent dynamic system design scheme must have at least four discrete masses bonded together by inertia-free connections, and the suspensions stiffness of transmission elements must also be taken into account. In order to analyze the simplifications of the design scheme, an analytical determination of the natural transmission frequencies was made, taking into account the reactive circuits for the seven mass systems. Figure 1 shows the design scheme, and their reduced parameters are indicated.
The basis for obtaining the differential equations of the wheeled forestry tractor transmission torsional oscillations is the Langrangian equations of the second kind:

$$\frac{d}{dt} \left( \frac{\partial T}{\partial \dot{q}_i} \right) - \frac{\partial T}{\partial q_i} + \frac{\partial H}{\partial \dot{q}_i} = Q_i(t)$$

(1)

where $T$ is the kinetic energy of the system; $P$ is potential energy of the system; $\Phi$ is dissipative function; $q_i$ is generalized coordinates; $Q_i$ is perturbing generalized forces.

For the generalized coordinates $q_i$ the angles $\phi_i$ of masses rotation fixed on the shafts are taken. They are counted from the system equilibrium deformed position, assuming that these angles in the equilibrium position are equal to zero. As generalized forces in this case, the torques $M_{i\theta}$ applied to the corresponding elements of the system are used.

![Figure 1](image.png)

**Figure 1.** Estimated equivalent 7 mass scheme of a dynamic system "engine – transmission - tractor with a bundle", taking into account reactive circuits:

- $J_1, J_2$ is the inertia moment of the engine and gearbox, respectively, at turning in the transverse plane relatively to the center of gravity under the influence of reactive moment;
- $J_3, J_4, J_5, J_6$ is the given moments of the transmission sections inertia;
- $J_7$ is inertia moment of the tractor and bundle translational moving mass;
- $C_1, C_2$ is the given to the angular stiffness of the suspension engine and transmission;
- $C_3-C_7$ is the crankshaft stiffness coefficient of the transmission sections reduced to the crankshaft axis;
- $C_{1c}, C_{2c}$ is the reduced to the angular stiffness of the tractor mass in the transverse plane;
- $k_1, \ldots, k_6$ is the damping coefficients of the corresponding transmission sections.

Substituting the found values of $T$ and $P$ into the corresponding Langrange equations and performing the corresponding transformations, a system of differential equations describing the transmission shafts torsional vibrations is obtained

$$J_1 \dot{\phi}_1 + k_2 (\phi_1 - \phi_2) + c_2 (\phi_1 - \phi_2) - k_3 (\phi_1 - \phi_3) - c_3 (\phi_1 - \phi_3) = 0;$$
$$J_2 \dot{\phi}_2 + k_3 (\phi_2 - \phi_3) + c_3 (\phi_2 - \phi_3) - k_4 (\phi_2 - \phi_4) - c_4 (\phi_2 - \phi_4) = M_1;$$
$$J_3 \dot{\phi}_3 + A_1 \dot{\phi}_3 + A_2 \dot{\phi}_4 - k_5 (\phi_3 - \phi_5) - c_5 (\phi_3 - \phi_5) = 0;$$
$$J_4 \dot{\phi}_4 + k_5 (\phi_4 - \phi_5) + c_5 (\phi_4 - \phi_5) - k_6 (\phi_4 - \phi_6) - c_6 (\phi_4 - \phi_6) = 0;$$
$$J_5 \dot{\phi}_5 + k_6 (\phi_5 - \phi_6) + c_6 (\phi_5 - \phi_6) - k_7 (\phi_5 - \phi_7) - c_7 (\phi_5 - \phi_7) = 0;$$
$$J_6 \dot{\phi}_6 + k_7 (\phi_6 - \phi_7) + c_7 (\phi_6 - \phi_7) - k_8 (\phi_6 - \phi_8) - c_8 (\phi_6 - \phi_8) = 0;$$
$$J_7 \dot{\phi}_7 + k_8 (\phi_7 - \phi_8) + c_8 (\phi_7 - \phi_8) - k_9 (\phi_7 - \phi_9) - c_9 (\phi_7 - \phi_9) = 0;$$

(2)

Differential equations of any system forced oscillations with the presence of disturbing moments applied to the elements

$$Q(t) = Q_{0i} \cos pt,$$

(3)

where $Q_{0i}$ is the amplitude of disturbance;

$p$ is frequency.

After the corresponding transformations, the equations for the torques in the sections are obtained:

$$M_{i\theta} = C_1 \ (\phi_2 - \phi_1),$$
$$M_{i\theta} = C_2 \ (\phi_2 - \phi_4),$$
$$M_{i\theta} = C_{n-1} \ (\phi_n - \phi_{n-1}).$$

(4)

After substituting the values $\phi_1, \phi_2, \ldots, \phi_n$ in formulas (2,4), the following equations are obtained:

$$M_{i\theta} = C_1 \ (a_2 - a_1) \ \cos pt,$$
$$M_{i\theta} = C_2 \ (a_3 - a_2) \ \cos pt,$$
$$M_{i\theta} = C_{n-1} \ (a_n - a_{n-1}) \ \cos pt.$$

(5)
For the frequency analysis of the vibrational transmission system of a wheeled forestry tractor, the transmission and frequency functions use is of interest. The differential equations system is the torsional vibrations equations. Solving this system of equations with zero right-hand side, the natural oscillations of each inertial mass can be found. The solution of the system with the right-hand side, which describes the disturbing moment applied to the elements, allows determining the forced oscillations of the system under the action of disturbing forces that periodically change over time. These vibrations increase the dynamic transmission loading.

**Acknowledgements**

The result of solving these equations, taking into account external disturbances, is presented in a tabular form. The table shows the engine natural frequencies and disturbance frequencies of the "engine - transmission - tractor - wood bundle" system at the theoretical speed of the skidding system in various gears for seven mass dynamic design scheme. The dominant operating range of the engine crankshaft rotational speed of a wheeled forestry tractor is from 1800 ... 2000  min\(^{-1}\), in short-term modes - 1200 min\(^{-1}\), and in cases of short-term engine overloads, the speed can be reduced to 800 min\(^{-1}\).

The operator’s desire to obtain maximum productivity results in the fact that the main part of the transport operations machine time, the engine works when the control lever is set to the maximum fuel supply. This is accompanied by a relatively stable rotational speed of the engine crankshaft, regardless of operating conditions and tractor power saturation.

In order to assess the engine speed the crankshaft speed utilization is defined:

\[ K_p = \frac{n_e}{n_{\text{opt}}}, \]  

(6)

where \(n_e\) is mathematical expectation of engine crankshaft speed; \(n_{\text{opt}}\) is engine crankshaft speed at rated power.

The theoretical movement speed is calculated with the engine speed utilization coefficient \(K_p = 0.8\).

It is known that when the disturbing force frequency coincides with the natural oscillations frequency of the system, resonant oscillations occur. These oscillations are maintained indefinitely due to the action of a disturbing force and, therefore, are of great practical importance [5,7].

The amplification factor \(\beta\) is the ratio [4]

\[ \beta = \frac{a}{a_0} \]  

(7)

where \(a\) is the steady-state amplitude of the disturbing frequency strength, \(a_0\); \(a_0, a = h/p^2\) is the deviation of the system from the equilibrium position under the constant force action;

\[ h = H/m, H \] is the amplitude;

\(m\) is the mass.

Table 1 – Eigen frequencies values of the system "engine-transmission-tractor-bundle» of the 7 mass system

| Parameters | Transmissions |
|-----------|---------------|
| Eigen frequency mode, rad/s | I | II | III | IV |
| 4,7 | 4,703 | 4,703 | 4,703 |
| 8,05 | 8,903 | 7,873 | 5,371 |
| 58,32 | 56,4 | 33,34 | 28,85 |
| 142,8 | 243,9 | 447,77 | 802,6 |
| 304,1 | 365,4 | 530,31 | 877,7 |
| 4632 | 4190 | 3940 | 3460 |

For small damping coefficient values, the maximum value of the coefficient \(\beta\) is observed at a point lying very close to the resonance; therefore, it is quite acceptable to take its value at resonance as the maximum value of the coefficient \(\beta\).

Therefore, the analysis of the “engine – transmission – tractor – bundle” system eigen frequencies and the lugs synchronous excitation frequencies should be carried out not only at the resonance, when \(aP = 1\), but also in the resonance area, in which the coefficient \(\beta\) has a significant value; in this research a resonance area (0.75-1.25) \(aP\) is taken.
Figure 2. The resonance curve of the oscillatory circuit:

- $P$ is eigen frequencies;
- $\omega$ is imposed frequency;
- $\beta$ gain factor

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Conclusions

Analysis of the eigen frequencies and the system exciting frequencies allows to draw the following conclusions:

- the engine-and-transmission installation can operate in the resonance zone in lower gears: the eigen frequency is 56.54 rad/s with a crankshaft speed in the range of 1200 min$^{-1}$ in first gear and 140.6 rad/s with a speed of 2000 min$^{-1}$ in second gear during the interaction of the wheels lugs with the bearing surface;
- the engine-and-transmission installation can operate in the resonance zone in higher gears (3rd and 4th gear) with a crankshaft rotation frequency in the range of 1200 ... 2200 min$^{-1}$.

In order to exclude resonant modes in the transmission of a wheel skidder, the possibility of installing elements with greater flexibility is considered.

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