

Research of dynamic loading in a drivetrain by means of mathematical modeling

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Abstract. This paper describes the development of the initial full dynamic model of a caterpillar agricultural tractor 'Chetra 6C-315' drivetrain, the processes of the model reduction; the mathematical apparatus for defining loads acting on drivetrain shafting elements. The main results of computational researches of dynamic loadings of drivetrain elements in various tractor operation modes are presented.

1. Introduction

Despite the continuous improvement of modern tractors drivetrains constructions, dynamic loading of drivetrains remains at a high level. It results in accumulation of fatigue damage and premature failures of details subjected to the highest loading. It especially concerns machines which were tested insufficiently and their construction was not enhanced properly. For example, agricultural tractor Chetra-6C315 (drawbar category 6) was designed on the base of an industrial tractor (drawbar category 9) construction. At that, constructions of some units, particularly a gearbox and a steering mechanism, were taken from a 9-tonn machine to a 6-tonn one without any changes. So, the development of an adequate mathematical model of the drivetrain and the search of ways for decreasing dynamic loadings in the drivetrain of this tractor in various operation modes by the created model is a topical goal.

2. Development and specifics of the dynamic model

The structural scheme of the tractor drivetrain created on the base of analysis of technical documentation was used as the base for development of a dynamic model. The full initial model includes 98 lumped masses with elastic and friction links. Dynamic parameters of its elements were defined by computations according to the method described in [1].

As far as the initial dynamic model of the drivetrain has complicated the branched structure with the large number of masses, which have high partial frequencies (above 10000 Hz), simplification of the initial model was made. The initial model was changed to the equivalent model with fewer moving masses. For simplification, Rivin's method was used. This method is based on changing of a separate elementary two-mass oscillation system with the highest partial frequency to a one-mass system by combination of two masses into one and changing the links compliance of the combined mass.

The reduced dynamic model is presented and described in [2].

The advantage of the reduced model is providing analysis of:

- a mutual influence of torsional vibrations in drivetrain and longitudinal-angular vibrations of the tractor frame;



- ### 3. Development and research of the mathematical model

Main elements of subsystems are blocks corresponding to inertial masses of the system. These blocks are enumerated in accordance with figure 1. Other blocks represent external and internal impacts. The model provides researches of the drivetrain loading both by separate impact factors and by their mutual influence.

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corresponding block provides researching of speed and power characteristics of power flow through the planetary mechanism both when any element is stopped, and during rotation of all elements with various angular speeds. Also, vibration processes generated between connected elements and damping characteristics of their link are taken into account.

4. Dynamic parameters of loading modes

The general loading mode of pull and transport machines could be presented as a sum of quasistatic and dynamic components of the torque [1-4]. A static component is simple to calculate – necessary data are engine power characteristic, conditions of a propulsor and work equipment interaction with the ground.

A dynamic component is considered by means of the coefficient of external dynamic loads k_{dv} calculated from relation:

$$k_{dv} = 1 + \frac{P_A}{P}, \quad (1)$$

where P – tangential load corresponding to the rated torque, P_A – average dynamic load [2]. Tangential load is calculated from expression:

$$P = \frac{2 \cdot M}{m \cdot z}, \quad (2)$$

where M – rated torque, m – average modulus, z – gear teeth number.

$$P_A = \sqrt{P_a \cdot (2 \cdot D - P_a)}, \quad (3)$$

$$P_a = \frac{P_1 \cdot P_2}{P_1 + P_2}, \quad (4)$$

where P_1 – force in gearing in case of absence of elastic deformation of teeth, shafts and other elements; P_2 – force in gearing during steady rotation of gears and bending of teeth in the range of a step errors value.

Dynamic factor D is calculated from expression:

$$D = \Delta \cdot b_c \cdot C \cdot \cos^2 \beta, \quad (5)$$

where Δ – rated factory inaccuracy of gear, C – specific teeth stiffness, b_c – average value of the rim, β – pressure angle.

The model research allows one to obtain oscillograms of toques on every areas of the drivetrain, to define maximal and minimal dynamic components, and the average value of the static component.

5. Research of dynamic loading of the drivetrain from the sprocket and caterpillar chain rewinding unevenness

The plan of the computational research is presented further. On the calculation start in dialog mode rated values of external and internal forces and torques, the character of loads are entered. In accordance with drawbar force calculation, the operation mode of the engine incase of the maximal torque is set [5, 6]. For this research, it was assumed that the drawbar load is constant in time and equal to the load on maximal possible drawbar power. Modelling time was set equal to 10 sec. During that time, all transient processes in the drivetrain will end. The range of the sprocket and caterpillar chain rewinding frequencies is defined by the range of tractor working speeds. The relation between these parameters is the following:

$$V = \frac{\omega \cdot 2\pi \cdot R_s}{z}, \quad (6)$$

where ω – frequency of sprocket and caterpillar chain rewinding, R_s – sprocket radii, z – teeth number of the sprocket.

Oscillograms of elastic torques on each area of the drivetrain were obtained as the result of modeling. An example of the oscillogram is presented in figure 2.

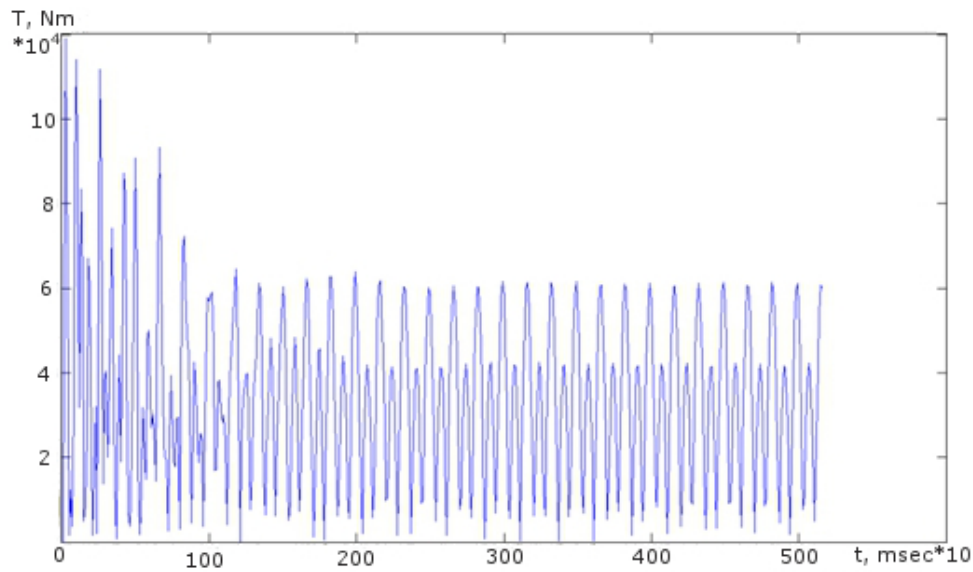


Figure 2. The oscillogram of the elastic torque on 20th area (planetary mechanism in differential steering mechanism) at the tractor speed 0.75 m/sec (sprocket and caterpillar chain rewinding frequency – 5 Hz).

Oscillograms were processed by means of the program: maximal M_{\max} and minimal M_{\min} values of the elastic torque were calculated, and the value of the average M_c torque on each area was calculated according to the following relation:

$$M_c = M_{\min} + \frac{M_{\max} - M_{\min}}{2}. \quad (7)$$

Calculation of the dynamic factor is as follows:

$$k_d = \frac{M_{\max}}{M_c}. \quad (8)$$

Research results are relations of loads dynamic factor variations on areas in the range of tractor working speeds. An example of those relations of the dynamic factor variation is presented in figure 3.

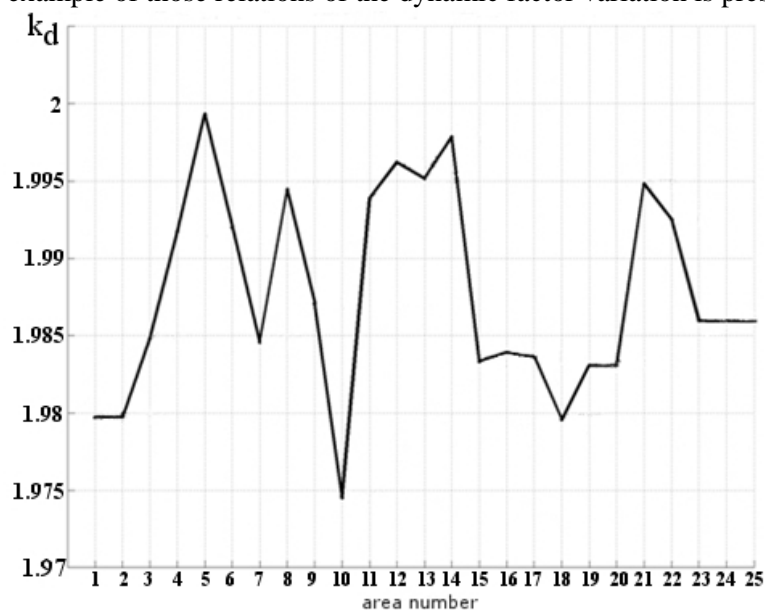


Figure 3. Values of the dynamic factor on areas at tractor speed $V=2.26$ m/sec.

Also, the range of computational researches for defining the pattern of the torsional vibrations transfer through shafting was performed. These torsional vibrations are generated by unevenness of operation loads and rewinding of the caterpillar chain.

6. Conclusion

- Analysis of the obtained graphics allows identifying areas with the highest dynamic loading on main tractor operation modes. Modes with the highest loads on areas lying near loads source (final drive and differential turn mechanism), and modes where the highest loading is on distant areas (propeller shaft, shafts and a gear of the gearbox) were found.
- Analysis of torsional vibration transferring through shafting shows that loads with frequencies of 0 – 12 Hz from the sprocket go through the whole shafting almost without amplitudes decreasing. Additional dynamic loading in this case is 50 – 120% of the activation torque. Additional loadings from caterpillar chain rewinding with frequencies 12-24 Hz on rear axle areas rises to 90-120% of the activation torque. Amplitudes of high-frequency loadings from engine harmonics are significant on areas which are close to the engine, but behind the gearbox and farther, these amplitudes are close to zero.
- Shafting is practically transparent for transferring of low-frequency vibrations. Construction measures are necessary for limiting (by means of installing elastic elements or vibration dampers) action of low-frequency vibrations on areas behind the final drive.

References

- [1] Giancarlo G. 2009 *Vibration dynamic and control* (Springer science: Buisness media)
- [2] Shekhovtsov V 2013 *The Archives of Automotive Eng.* **59** 85-97
- [3] Shekhovtsov V, Sokolov-Dobrev N and Potapov P 2016 *Procedia Eng.* **150** 1239-44
- [4] Popp K, Schiehlen W 2010 *Ground vehicle dynamic* (Berlin: Springer-Verlag Heidelberg) p 348
- [5] Rajamani R 2012 *Vehicle dynamics and control* (Springer)
- [6] Rill G 2012 *Road vehicle dynamics: fundamentals and modeling* (CRC press. Taylor and Francies group)
- [7] Shevchuk V L 2013 *Modern High Technologies* **2** 44-49