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Investigation of Frequencies of Self Torsion Vibrations of Individual and Group Traction Drives of Locomotives

A. Klokovs*, P. Balckars, K. Ljapshins

**Riga Technical University, TMF, LV-1658, Riga, Kalku street 1, Latvia, E-mail: kabanelo@inbox.lv*

Abstract

This work presents the investigations of frequencies of the self torsion vibrations group and individual traction drive of locomotives. As an example a traction drive of diesel – train DR1 was chosen. For the system of traction drive the expressions of kinetic and potential energy have been completed. With the use of the Lagrange equations of second type the equations of motion of the system were made. The decision of the equations of motion was conducted by the mathematical simulation of dynamic processes and used the special software - MATHCAD.

The investigations of a diesel - train DR1 traction drive self vibrations are made with and without the resilient interaction (creep) of wheel with a rail.

The obtained results allow estimating the role of influencing of the resilient sliding of wheel on rails. These results also demonstrate how torsion vibration frequencies differ with the use of individual and group traction drives. It enables a rational choice of the proper traction drive.

KEY WORDS: *frequencies, torsion vibrations, traction drive, locomotive, wheels, autooscillations.*

1. Introduction

For the increasing of reliability and longevity of carriage parts and all units of stuffs and aggregates of locomotive different methods are used. One of them is that the dynamic qualities of locomotive are being improved – the dynamic forces, which operate on a main frame, frame of bogie, traction electric motors, are taken to the minimum. It's similarly necessary to reduce vibrations, operating on a power and auxiliary equipment, located in the body of locomotive. Research of vibrations can promote working out these problems in the traction transmission of locomotive.

A traction transmission of modern locomotive is a difficult unit; working under specific conditions, substantially different from the conditions of work of not only stationary transmissions but also transmissions of trackless vehicles.

The maximal dynamic loadings at vibrations in a traction drive arise up at resonance, it is therefore important to know frequencies of own vibrations of drive at the different modes of motion of a transport crew.

The insufficient theoretical elaboration of these questions resulted in the lack of a well-organized system of approaches for choice of charts and parameters of traction transmissions which would provide creation of reliable in exploitation traction drive of locomotive. As a result of it the traction drive of locomotive remains a unit unreliably working in exploitation.

Therefore the calculations conducted in the research and determination of frequencies of self torsion vibrations of traction drive of locomotives enable to confront the separate variants of individual and group traction drives of locomotive [1], (Fig. 1, a, b), under similar conditions and to show their features, dignities and failings. The purpose of work was determination of frequencies of own vibrations of individual and group traction drive of locomotive in three modes:

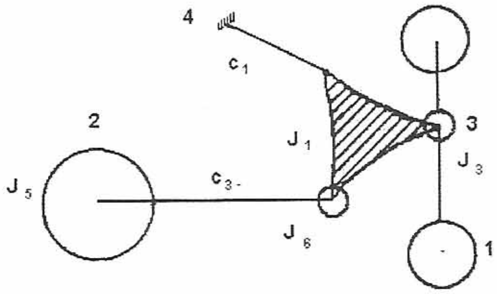
- mode of set motion, when the wheeled pair do not participate in turning vibrations;
- mode of sliding of wheels on rails, when the wheeled pair are accomplished by turning vibrations about own axes of rotation;
- mode of resilient sliding wheels on rails (phenomenon of creep).

As an example a traction drive of diesel - trains DR1 was examined.

The conducted researches of own turning vibrations frequencies of traction drive of locomotives will be given by possibility to confront the separate variants of traction drives, individual and group (Fig. 1), in identical terms and to show their features.

Calculation of the forced vibrations of individual traction drive, defined with the kinematics excitation at the vibrations of bogie frame, can be made in the following order. First of all the form of the forced vibrations of drive for frequencies suitable for the vibration frequencies of bogie frame is estimated (it is expedient to choose the «crack» of the system in the place were joining pendant a body of reducing gear to the bogie frame) [2].

Calculation of the forced vibrations of individual traction drive, defined with the kinematic excitation at the vibrations of bogie frame, can be made in the following order. First of all the form of the forced vibrations of drive for frequencies suitable for the vibration frequencies of bogie frame is estimated (it is expedient to choose the «crack» of the system in the place were joining pendant a body of reducing gear to the bogie frame). Using information on whipping of the axle box hanging and forms of vibrations of bogie frame, amplitude of mass 4 (on the generalized chart of system) at regular kinematics excitation or some other revolting function is set, and then the form of drive vibrations amplitude is taken into account at determination of vibrations of the armature of electric motor and dynamic moments on the areas of rotor. For the resonance vibrations it's necessary to take into account a friction in the system.



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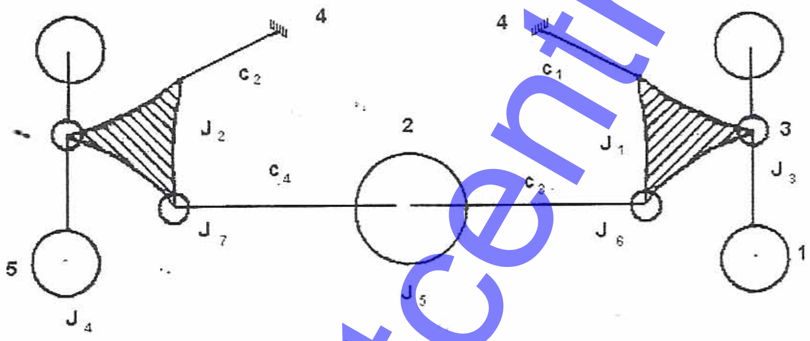

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Fig. 1 Calculation chart of traction drive: a) individual traction drive, b) group traction drive. 1, 5 – wheeled pairs: J_1, J_2 - moments of inertia of reducing gears; 2 – electric engine; J_5 - moment of inertia of engine; J_3, J_4 – moments of inertia of wheeled pairs; 3 - body of reducing gear; 4 – bogie frame; J_6, c_3 and J_7, c_4 - moments of inertia and inflexibilities of cardan shaft; c_1 and c_2 - inflexibility of reactive tractions

At the calculations of self frequency of torsion vibrations of traction drive the followings assumptions are adopted [3].

1) A bogie frame, body of reducing gears, reactive tractions, shafts of engine and reducing gears, cog-wheels of reducing gears, wheeled pair and their axes are absolutely solid bodies, because the parameters of inflexibility of them are considerably higher than the parameters of inflexibility, connecting their resilient elements.

2) Allowing the comparative character and ultimate purpose of researches it's possible to scorn:
   - by gaps in connections of units of the system;
   - by the kinematics errors of the gear;
   - by influence of moving of body on the free vibrations of bogie.

3) Connection of reducing gears with the bogie frame is carried out only through reactive tractions, because cardan shafts have the axial moving in spline connections.

4) The rotating parts of drive do not have an imbalance.

5) The assist longitudinal symmetry of overspring structure of bogie, i.e. the center of vibrations of galloping coincides on length of bogie with the centre at gravity of overspring structure.

6) The examined mode of motion a diesel - train, when the speed of movement and moment on the shaft of engine is permanent.

On the basis of the first point of the accepted assumptions the reducing gear of the wheeled pair can be presented as an element with one degree of freedom, position of which is simply determined with the angle of turn of its body in relation to the axis of rotation of the wheeled pair. Such rotary moving body of reducing gear can be examined as it portable motion. Then corners of turn of elements, being in the body of reducing gear, about own axes rotations corresponds their relative motion. Consequently, the elements of traction drive, being in the body of reducing gear of the wheeled pair (except for the driven toothed wheel on the axis of the wheeled pair), participate in difficult motion. Positions of the revolved parts of traction drive proper the shaft of engine and wheeled pair simply determined the corners of turn them in relation to the own axes of rotation.

Research of free torsion vibrations in a individual and group traction drive was conducted, when a moment on the shaft of engine and speed of movement of diesel engine is permanent, but the wheeled pair here accomplish rotary vibrations in relation to own axes due to forces of resilient co-operation (creep) in the areas of contact of wheels with rails.

At such raising of task the system of group drive has two generalized coordinates more than in the mode of motion without taking into account resilient deformation in the area of contact of wheels with rails. These generalized coordinates determine position of the wheeled pair at rotary vibrations in relation to the own axes of rotation.

In research the equations of kinetic energy of traction drive were made:

$$T = \frac{1}{2} I_3 (\dot{\varphi}_3)^2 + \frac{1}{2} I_1 (\dot{\varphi}_1)^2 - I_7 \dot{\varphi}_1 \dot{\varphi}_3 + \frac{1}{2} I_5 (\dot{\varphi}_5)^2 - I_6 \dot{\varphi}_2 \dot{\varphi}_4 + \frac{1}{2} I_2 (\dot{\varphi}_2)^2 + \frac{1}{2} I_4 (\dot{\varphi}_4)^2 \quad (1)$$

where: I_1, I_2 are moment of inertia of the first and second reducing gear; I_5 is moment of inertia of engine billow; I_3, I_4 are moment of inertia first and second wheeled pairs; I_6, I_7 are moment of inertia of the first and second cardan shaft; $\dot{\varphi}_1, \dot{\varphi}_2$ are speed of the rotatory moving of the first and second reducing gears; $\dot{\varphi}_3, \dot{\varphi}_4$ are speed of the rotatory moving of the first and second wheeled pairs; $\dot{\varphi}_5$ is speed of the rotatory moving of engine billow; $\dot{\varphi}_6, \dot{\varphi}_7$ are speed of the rotatory moving of the first and second cardan shafts.

Taking into account the values at gum elements of reactive tractions- equation for potential energy of traction drive

$$\begin{aligned} \Pi = {} & \frac{1}{2}(\dot{c}_1 + \ddot{c}_3)\varphi_1^2 + \frac{1}{2} c_3 i_p^2 \varphi_3^2 - \dot{c}_3 i_p \varphi_1 \varphi_3 - \dot{c}_3 \varphi_1 \varphi_5 + c_3 i_p \varphi_3 \varphi_5 + \\ & + \frac{1}{2}(c_3 + c_4)\varphi_5^2 + c_4 i_p \varphi_4 \varphi_5 - \dot{c}_4 \varphi_2 \varphi_5 - \dot{c}_4 i_p \varphi_2 \varphi_4 + \frac{1}{2} c_4 i_p^2 \varphi_4^2 + \frac{1}{2}(\ddot{c}_2 + \ddot{c}_4)\varphi_2^2 \end{aligned} \quad (2)$$

where $c_1, c_2, c_3, c_4, \dot{c}_1, \ddot{c}_3, \ddot{c}_4$ are parameters of inflexibility of damper gum resilient elements of the first and second reducing
gears reactive traction, cardan billows, connecting the billow of engine with the entrance billow of the first and second reducing gears; i_p is gear-ratio of reducing gear; φ_1, φ_2 are corners of turn of the first and second reducing gears; φ_3, φ_4 are corners of turn of the first and second wheeled pairs; φ_5 is corners of turn of engine billow.

Usually researching torsion vibrations group or individual traction drives of locomotives the following is accepted, that the wheeled pair at motion of her on a railway in the conditions of the set motion of locomotive don't participate in the torsion vibrations at the system of drive. At such assumption influence of creep on the torsion vibrations of drive is not taken into account.

In conjunction with the increasing power of locomotives, including diesel – trains being on motive axes, ignoring forces of creep can tell on authenticity of results of the investigations. Therefore the researches of drives like in this work are made both with and without taking into account the creep.

For the receipt of differential equations representing the free torsion vibrations of individual and group drive and vertical vibrations of bogie frame this work used the Lagrange equation of the second type [1]:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial q^*}\right) - \frac{\partial T}{\partial q} + \frac{\partial \Pi}{\partial q} = 0 \quad (3)$$

where q, q^* are generalized co-ordinates and speeds.

After transformations differential equations of traction drive free torsion vibrations that are received with allowance forces of resilient coupling rails with wheels:

$$\begin{aligned} & I_1 \cdot \ddot{\varphi}_1 - I_7 \cdot \ddot{\varphi}_3 + c_7 \cdot \varphi_1 - \dot{c}_3 \cdot i_p \cdot \varphi_3 - \dot{c}_3 \cdot \varphi_5 = 0 \\ & I_3 \cdot \ddot{\varphi}_3 - I_7 \cdot \ddot{\varphi}_1 + 2B_K \cdot \dot{\varphi}_3 + c_3 \cdot i_P^2 \cdot \varphi_3 - \dot{c}_3 \cdot i_p \cdot \varphi_1 + c_3 \cdot i_P \cdot \varphi_5 = 0 \\ & I_5 \cdot \ddot{\varphi}_5 + c_5 \cdot \varphi_5 + c_3 \cdot i_p \cdot \varphi_3 + c_4 \cdot i_P \cdot \varphi_4 - \dot{c}_3 \cdot \varphi_1 - \dot{c}_4 \cdot \varphi_2 = 0 \\ & I_4 \cdot \ddot{\varphi}_4 - I_6 \cdot \ddot{\varphi}_2 + 2B_K \cdot \dot{\varphi}_4 + c_4 \cdot i_P^2 \cdot \varphi_4 - \dot{c}_4 \cdot i_p \cdot \varphi_2 + c_4 \cdot i_P \cdot \varphi_5 = 0 \\ & I_2 \cdot \ddot{\varphi}_2 - I_6 \cdot \ddot{\varphi}_4 + c_6 \cdot \varphi_2 - \dot{c}_4 \cdot i_p \cdot \varphi_4 - \dot{c}_4 \cdot \varphi_5 = 0 \end{aligned} \quad (4)$$

In the system of differential equations of traction drive free torsion vibrations are plugged characterizing the resilient sliding of wheels on the rails parameters B_K, representing the so-called phenomenon of creep.

Generalized force, that proper the generalized co-ordinate of the rotatory moving of wheel of drive in relation to the own axis of rotation [3]:

$$Q = -k \frac{R_k^2}{V} \dot{\varphi} = -B_k \dot{\varphi} \quad (5)$$

where k is coefficient of creep; V is speed of wheel; R_k is radius of wheel; B_k is coefficient, such taking as coupling wheel with a rail.

Table

Values of traction drive, used in work

Parameters of drive	Diesel - train DR 1
Moments of inertia of the rotary parts of traction drive, $kgfms^2$	
I_1, I_2	15,4
I_3, I_4	15,6
I_6, I_7	1,8
I_5	2,8
Parameters of inflexibility between the elements of traction drive, kgf/m	
c_1, c_2	$(0.5-2.2)10^6$
c_3, c_4	$6.2 \cdot 10^4$
Coefficient, such taking as coupling wheel with a rail (creep), $kgfms/rad$	
$2B_k$	$50.7 \cdot 10^4$

Received values of torsion vibrations frequencies:

- individual traction drive without the resilient coupling between wheels and a rail (creep):

$$f_1 = 36.37Hz,\ f_2 = 58Hz$$

- individual traction drive with the resilient coupling between wheels and a rail (creep):

$$f_1 = 27.53Hz,\ f_2 = 37.42Hz,\ f_3 = 58Hz$$

- group traction drive without the resilient coupling between wheels and a rail (creep):

$$f_1 = 27Hz,\ f_2 = 44.4Hz,\ f_3 = 48.4Hz$$

- group traction drive with the resilient coupling between wheels and a rail (creep):

$$f_1 = 0.03Hz,\ f_2 = 0.1Hz,\ f_3 = 1Hz,\ f_{4,5} = 23Hz$$

2. Conclusions:

Solution of the system of equations of motion for a drive with the parameter presented in table 1 showed, that the least values of self frequencies of torsion vibrations prevailed at group traction drive, because this drive has five masses that reduce the frequencies of vibrations.

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