THE INFLUENCE OF JET LINKS' PARAMETERS ON THE STRESS LOADING OF TRACTOR'S POWER TRAIN SITES

Zahid Godzhaev

Federal Centre of Machinebuilding Research and Testing
NATI Street 13, 142322 Nowy Byt, Chechov District, Moscow Region, Russia
tel.:+7 (495) 1021256, fax: +7 (495) 5460736
e-mail: godj@tsr.ru

Victor Shekhovtsov, Nickolay Sokolov-Dobrev,
Vladimir Shevtchuk, Mikhail Lyashenko

Volgograd State Technical University, Auto-Tractor Faculty
Lenin avenue 28, 4000131 Volgograd, Russia
tel.:+7 (8442) 248116, fax: +7 (8442) 248116
e-mail: shehovtsov@vstu.ru, ts@vstu.ru

Abstract

In the article are described the results of calculation researches of influence of ruggedness of fastening to a frame in cross-section and longitudinal planes of case parts of a power chain of a caterpillar, and also ruggedness of communication with a frame in the same planes of a combustion engine on dynamic stress loading of sites of a power train during movement with different speeds. Except the specified ruggedness of fastening, this dynamic loading depends on parameters of shifting jerks which take place in a power shafting owing to non-uniformity of action of the basic operational loadings – oscillations of traction resistance, cross-section and axial vibrations of a framework of a tractor on a suspender, rewinds of a caterpillar chain, non-uniformity of rotation of a shaft of the engine. Researches have shown that at the expense of steering of ruggedness of fastening to an engine bed and case parts of transmission it is possible to essentially lower dynamic stress loading of separate sites of transmission in movement modes when details of these sites are loaded as much as possible. The circuit solution is offered for realization in a fastening design to a frame of the case of transmission of caterpillars of VT family which will allow to lower stress loading of transmission details on certain speeds of movement.

Keywords: *transport, caterpillar transport, power train, dynamic stress loading, mathematical model, results of researches*

1. Introduction

It is known, that the cases of units and mechanisms, serving by bearing parts for rotating details, perceive from these details the jet moments, n a direction opposite to the torque. In this connection in power circuit links with jet connections are formed. Participation of these links in oscillatory process, which is taking place owing to non-uniform action of operational loadings, effects the stress loading of the power train. [1] Details' durability can appreciably depend on the degree of this influence.

2. Dynamic model of a power train

The communication of all units, involved in power flux transmission and participating in the oscillative motion from non-uniformity of exploatational loadings' influence, is detailed shown on the increased spatial model (Fig. 1). For the necessary researches the dynamic model of a VgTZ VT-100 tractor's power train is developed (Fig. 2).

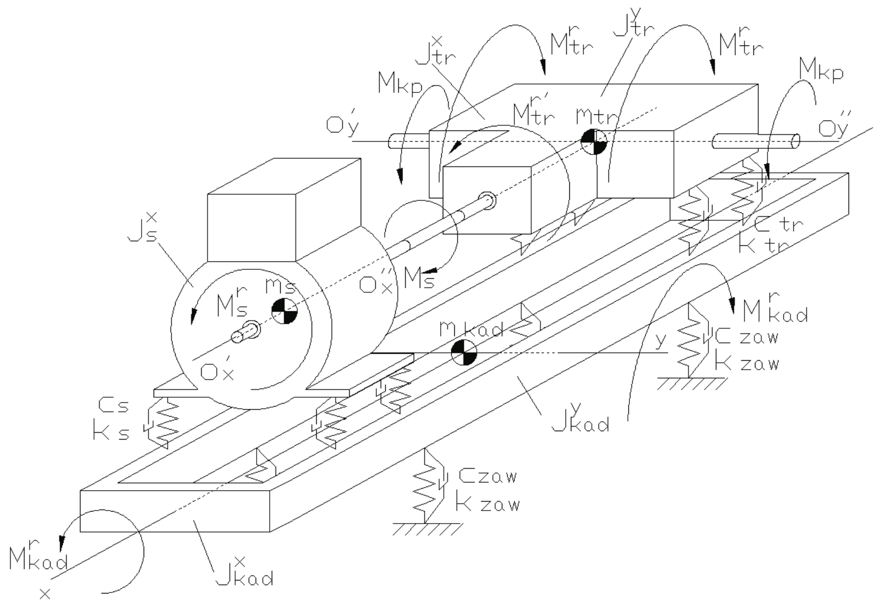


Fig. 1. The scheme of the spatial dynamic model

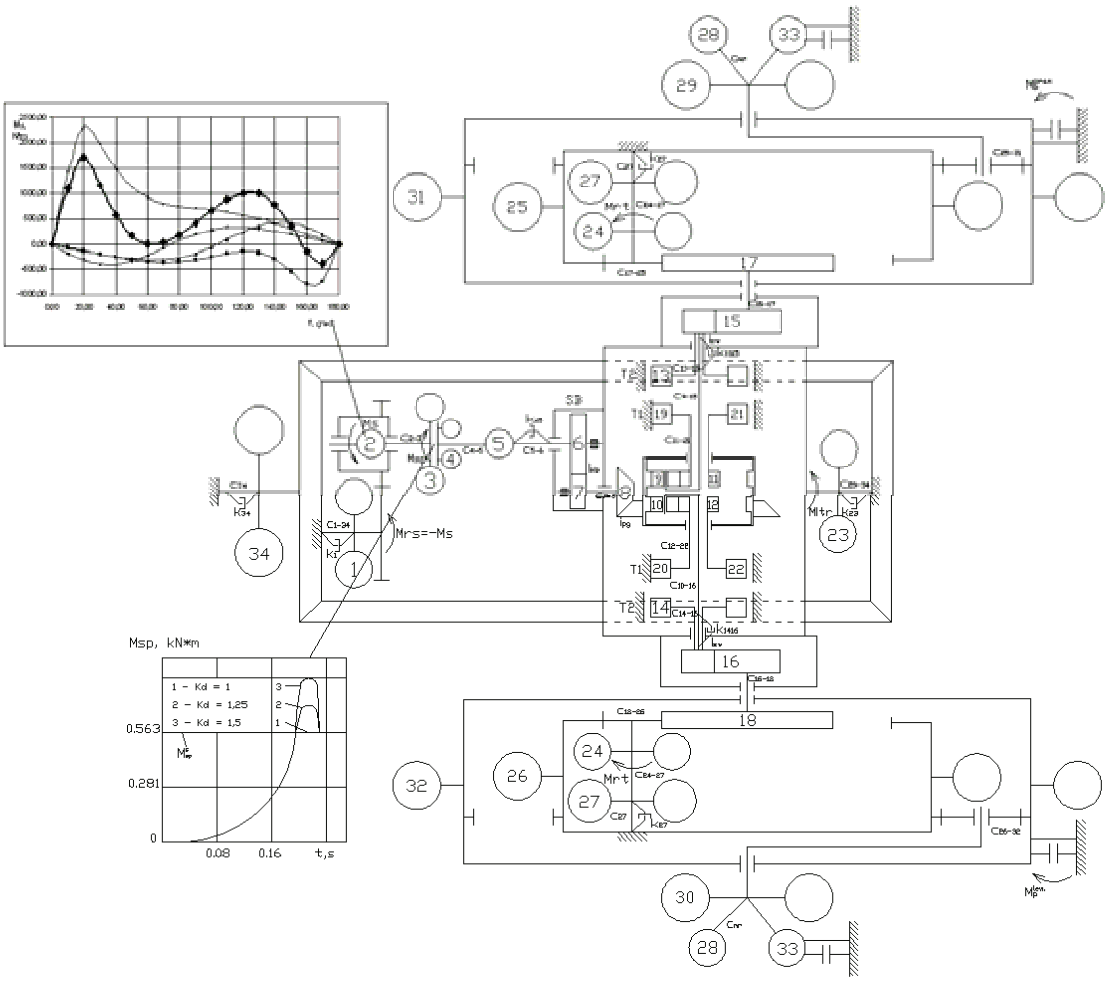


Fig. 2. The scheme of the dynamic model of the power train

The model includes 34 concentrated weights, connected by elastic and frictional communications. The values of its' elements' dynamic parameters are shown in the Table 1.

Its' values are defined by a settlement way according to a technique resulted in work [1].

Tab. 1. Dynamic parameters of model's elements

Element sign	Element name	Value
Moment of inertia of concentrated masses, kg·m²		
I_1	The engine concerning a longitudinal shaft	133.3
I_2	Crankshaft of the engine with resulted weights of crank gear	0.675
I_3	Flywheel with leading elements of a clutch coupling	2.025
I_4	Conducted elements of a clutch	0.767
I_5	The front hinge of a drive line	0.019
I_6	Leading elements of a gear	0.062
I_7	Conducted elements of a gear	0.141
I_8	Final drive and a swinging mechanism gear	0.847
I_9, I_{10}	Satellites and a planet carrier of a swinging mechanism	0.057
I_{11}, I_{12}	Solar pinion gear of a swinging mechanism	0.004
I_{13}, I_{14}	Pulley of a stopping brake	0.487
I_{15}, I_{16}	Final drive	1.798
I_{17}, I_{18}	A driving wheel	5.754
I_{19}, I_{20}, I_{21}, I_{22}	A pulley of a brake of a solar pinion gear	0.263
I_{23}	Case of transmission concerning a longitudinal axis	208.4
I_{24}	Case of transmission concerning a transverse axis	247.0
I_{25}, I_{26}	Caterpillar chain and rotating details of the running gear, reduced to a driving wheel axis	39.33
I_{27}	Framework of a tractor relative to a transverse axis	15800
I_{28}	Tool relative to a driving wheel axis	7660
I_{29}, I_{30}	Half of weight of a tractor framework in progressive motion relative to a driving wheel axis	4887.5
I_{31}, I_{32}	Basic branch of a caterpillar chain together with a ground at slipping	37.2
I_{33}	Framework of a tractor relative to a vertical axis	12500
I_{34}	Framework of a tractor relative to a longitudinal axis	8800
Torsion ruggedness of communications, N·m/rad		
c_{1-34}	The engine and a frame at longitudinal-angular oscillations	46336
c_{2-3}	Engine crankshaft	2500000
c_{3-4}	Clutch	9900
c_{4-5}	Clutch and cardan shaft	325000
c_{5-6}	Cardan shaft	6580
c_{6-7}	Gear-box	33850
c_{7-8}	Gear-box and final drive	1587550
c_{8-9}, c_{8-10}	A crown-wheel and a planet carrier of a swinging mechanism	1094340
c_{8-11}, c_{8-12}	A crown-wheel and a solar pinion gear	1380295
c_{11-21}, c_{12-22}	A crown-wheel and a pulley of a planetary brake	249723
c_{9-15}, c_{10-16}	Shaft of the rear bridge	60600
c_{13-15}, c_{14-16}	Shaft of a pinion gear of final drive and pulley of a stopping brake	7290000
c_{15-17}, c_{16-18}	Final drive and driving wheel	6370000
c_{17-25}, c_{18-26}	Leading site of the caterpillar, reduced to a driving wheel axis	120000000
c_{23-34}	The transmission case at the longitudinal-angular oscillations, reduced to a driving wheel axis	1819453

Element sign	Element name	Value
Torsion ruggedness of communications, N·m/rad		
$c_{24\text{-}27}$	The transmission case at the cross-section-angular oscillations, reduced to a axis of a main drive shaft of a gear-box	1419521
c_{27}	Suspenders at cross-section-angular oscillations of the framework, reduced to a driving wheel axis	487654
$c_{25\text{-}31}$, $c_{26\text{-}32}$	Ground under a seating in the horizontal plane, reduced to a driving wheel axis	from dependence *
C_{tool}	The tool with a tractor, reduced to a driving wheel axis	19455
c_{34}	Suspender at longitudinal-angular oscillations of the framework, reduced to a driving wheel axis	381763

* ground's carrying capacity q_s depends on the displacement deformation h_{gr} and general deformation h. The correlation is $q_s = h \cdot q_0 / h_{gr}$ [2, 3], where q_0 – average pressure of a track on the ground.

3. Mathematical model of the power train

To create a mathematical model of scheme's functioning Lagrange's equations of the second kind are used. Obtained on its' basis differential equations system, that describes the interaction of the power train's elements, is given by:

$$1\begin{cases} J_1\ddot{\varphi}_1 + k_1\dot{\varphi}_1 + c_{1-34}(\varphi_1 - \varphi_{34}) = -M_{rs}, \\ J_2\ddot{\varphi}_2 + k_2\dot{\varphi}_2 + c_{2-3}(\varphi_2 - \varphi_3 - \varphi_1) = M_s, \end{cases}$$

$$2\begin{cases} J_3\ddot{\varphi}_3 + k_3\dot{\varphi}_3 + c_{3-4}(\varphi_3 - \varphi_4) - c_{2-3}(\varphi_2 - \varphi_3 - \varphi_1) = 0, \text{ at } c_{3-4}(\varphi_3 - \varphi_4) < M_{sp}, \\ J_3\ddot{\varphi}_3 + k_3\dot{\varphi}_3 + M_{sp} - c_{2-3}(\varphi_2 - \varphi_3 - \varphi_1) = 0, \text{ at } c_{3-4}(\varphi_3 - \varphi_4) > M_{sp}, \\ J_4\ddot{\varphi}_4 + k_4\dot{\varphi}_4 + c_{4-5}(\varphi_4 - \varphi_5) - c_{3-4}(\varphi_3 - \varphi_4) = 0, \text{ at } c_{3-4}(\varphi_3 - \varphi_4) < M_{sp}, \\ J_4\ddot{\varphi}_4 + k_4\dot{\varphi}_4 + c_{4-5}(\varphi_4 - \varphi_5) - M_{sp} = 0, \text{ at } c_{3-4}(\varphi_3 - \varphi_4) > M_{sp}, \\ J_5\ddot{\varphi}_5 + k_5\dot{\varphi}_5 + c_{5-6}(\varphi_5 - \varphi_6) - c_{4-5}(\varphi_4 - \varphi_5) + c_{\sum(5-8)}(\varphi_5 - \varphi_8 i_{(5-8)} - \varphi_{23}(1 - i_{(5-8)})) = 0, \end{cases}$$

$$3\begin{cases} J_6\ddot{\varphi}_6 + k_6\dot{\varphi}_6 + c_{6-7}(\varphi_6 - \varphi_7 i_{sb} - \varphi_{23}(1 - i_{sb}) \pm z_{sb}) - c_{5-6}(\varphi_5 - \varphi_6) = 0 \\ \text{at } |\varphi_6 - \varphi_7 i_{sb} - \varphi_{23}(1 - i_{sb})| \geq z_{sb}, \\ J_6\ddot{\varphi}_6 + k_6\dot{\varphi}_6 - c_{5-6}(\varphi_5 - \varphi_6) = 0 \text{ at } |\varphi_6 - \varphi_7 i_{sb} - \varphi_{23}(1 - i_{sb})| \leq z_{sb}, \\ J_7\ddot{\varphi}_7 + k_7\dot{\varphi}_7 + c_{7-8}(\varphi_7 - \varphi_8 i_{pg} - \varphi_{23}(1 - i_{pg}) \pm z_{pg}) - c_{6-7} i_{sb}(\varphi_6 - \varphi_7 i_{sb} - \varphi_{23}(1 - i_{sb}) \pm z_{sb}) = 0 \\ \text{at} |\varphi_7 - \varphi_8 i_{pg} - \varphi_{23}(1 - i_{pg})| \geq \pm z_{pg}, \\ J_7\ddot{\varphi}_7 + k_7\dot{\varphi}_7 - c_{6-7} i_{sb}(\varphi_6 - \varphi_7 i_{sb} - \varphi_{23}(1 - i_{sb}) \pm z_{sb}) = 0 \\ \text{at } |\varphi_7 - \varphi_8 i_{pg} - \varphi_{23}(1 - i_{pg})| \leq \pm z_{pg}, \end{cases}$$

$$4\begin{cases} J_{13}\ddot{\varphi}_{13} + k_{13}\dot{\varphi}_{13} - c_{13-15}(\varphi_{15} - \varphi_{13}) = 0 \text{ at } M_{h13} < c_{13-15}(\varphi_{15} - \varphi_{13}), \\ J_{14}\ddot{\varphi}_{14} + k_{14}\dot{\varphi}_{14} - c_{14-16}(\varphi_{16} - \varphi_{14}) = 0 \text{ at } M_{h14} < c_{14-16}(\varphi_{16} - \varphi_{14}), \\ J_{13}\ddot{\varphi}_{13} = 0 \text{ at } M_{h13} > c_{13-15}(\varphi_{15} - \varphi_{13}), \\ J_{14}\ddot{\varphi}_{14} = 0, \text{ at } M_{h14} > c_{14-16}(\varphi_{16} - \varphi_{14}), \end{cases}$$

$$5\begin{cases} J_{19}\ddot{\varphi}_{19} + k_{19}\dot{\varphi}_{19} - c_{11-19}(\varphi_{11} - \varphi_{19}) = 0 \text{ at } M_{\text{hpo}19} < c_{11-19}(\varphi_{11} - \varphi_{19}), \\ J_{20}\ddot{\varphi}_{20} + k_{20}\dot{\varphi}_{20} - c_{12-20}(\varphi_{12} - \varphi_{20}) = 0, \text{at } M_{\text{hpo}20} < c_{12-20}(\varphi_{12} - \varphi_{20}), \end{cases}$$

$$
6\quad
\begin{cases}
J_{23}\ddot{\varphi}_{23}+k_{23}\dot{\varphi}_{23}+c_{23-34}(\varphi_{23}-\varphi_{34})-c_{\sum(5-8)}(1-i_{(5-8)})(\varphi_5-\varphi_8 i_{(5-8)}+\varphi_{23}(1-i_{(5-8)}))- \\
-c_{6-7}(1-i_{sb})(\varphi_6-\varphi_7 i_{sb}-\varphi_{23}(1-i_{sb}))-c_{7-8}(1-i_{pg})(\varphi_7-\varphi_8 i_{pg}-\varphi_{23}(1-i_{pg}))- \\
-\dfrac{(1-i_{(5-8)})}{i_{(5-8)}}J_8\ddot{\varphi}_8-\dfrac{(1-i_{sb})}{i_{sb}}J_7\ddot{\varphi}_7-\dfrac{(1-i_{pg})}{i_{pg}}J_8\ddot{\varphi}_8=0, \\
J_{24}\ddot{\varphi}_{24}+k_{24}\dot{\varphi}_{24}+c_{24-27}(\varphi_{24}-\varphi_{27})-c_{17-25}(\varphi_{17}-\varphi_{24}-\varphi_{25}/r_{kp})-c_{18-26}(\varphi_{18}-\varphi_{24}-\varphi_{26}/r_{kp})- \\
-c_{\Sigma(8-17)}(1-i_{(8-17)})(\varphi_8-\varphi_{17}i_{(8-17)}-\varphi_{24}(1-i_{(8-17)}))-c_{\Sigma(8-18)}(1-i_{(8-18)})(\varphi_8-\varphi_{18}i_{(8-18)}-\varphi_{24}(1-i_{(8-18)}))- \\
-c_{8-9}(1-\dfrac{1-k_{po}}{k_{po}})\left(\varphi_8-\varphi_9\dfrac{1-k_{po}}{k_{po}}+\varphi_{11}\dfrac{1}{k_{po}}-\varphi_{24}(1-\dfrac{1-k_{po}}{k_{po}})\right)- \\
-c_{8-10}(1-\dfrac{1-k_{po}}{k_{po}})\left(\varphi_8-\varphi_{10}\dfrac{1-k_{po}}{k_{po}}+\varphi_{12}\dfrac{1}{k_{po}}-\varphi_{24}(1-\dfrac{1-k_{po}}{k_{po}})\right)- \\
-c_{8-11}(1-\dfrac{1}{k_{po}})\left(\varphi_8-\varphi_9\dfrac{1-k_{po}}{k_{po}}+\varphi_{11}\dfrac{1}{k_{po}}-\varphi_{24}(1-\dfrac{1}{k_{po}})\right)- \\
-c_{8-12}(1-\dfrac{1}{k_{po}})\left(\varphi_8-\varphi_{10}\dfrac{1-k_{po}}{k_{po}}+\varphi_{12}\dfrac{1}{k_{po}}-\varphi_{24}(1-\dfrac{1}{k_{po}})\right)- \\
-c_{15-17}(1-i_{zw})(\varphi_{15}-\varphi_{17}i_{zw}-\varphi_{24}(1-i_{zw}))-c_{16-18}(1-i_{zw})(\varphi_{16}-\varphi_{18}i_{zw}-\varphi_{24}(1-i_{zw}))- \\
-\dfrac{(1-i_{(8-17)})}{i_{(8-17)}}J_{17}\ddot{\varphi}_{17}-\dfrac{(1-i_{(8-18)})}{i_{(8-18)}}J_{18}\ddot{\varphi}_{18}+\dfrac{1}{1-i_{po}}J_9\ddot{\varphi}_9+\dfrac{1}{1-i_{po}}J_{10}\ddot{\varphi}_{10}-(i_{po}-1)J_{11}\ddot{\varphi}_{11}- \\
-(i_{po}-1)J_{12}\ddot{\varphi}_{12}-\dfrac{(1-i_{zw})}{i_{zw}}J_{17}\ddot{\varphi}_{17}-\dfrac{(1-i_{zw})}{i_{zw}}J_{18}\ddot{\varphi}_{18}=0,
\end{cases}
$$

where: J_i – moment of inertia of the concentrated masses,
c_{i-j} – torsion rigidity of its' elastic connections,
k_i – masses' oscillating damping factors,
$\varphi_i, \dot{\varphi}_i, \ddot{\varphi}_i$ – movement, speeds and accelerations of the masses in oscillatory motion.

In mathematical model the 1st block of the equations system describes conditions of combustion engine's elements loading; 2nd – of clutch coupling elements; 3rd – of gear-box elements; 4th – of stopping brakes elements, 5th – of planetary brakes elements; 6th – of transmission case oscillations.

4. Description of model's jet links

The first jet link allows taking into account engine's oscillations on a frame at action on its framework of a jet moment counterbalancing a torque (on Fig. 1 weight 1 with communications). The engine is presented as two inertial masses – a body mass and a mass of crankshaft's details. In this case it's rational to consider the body's oscillations relative to the longitudinal axis O_X'-O_X'' (Fig. 2). The moment of inertia of the engine's body relative to the crankshaft's axis and its fastening rigidity when oscillating the frame are recounted with use of the experimental data, determined in work [5].

While in service the transmission case makes angular fluctuations concerning a frame in cross-section and longitudinal vertical planes. Therefore it is presented in the form of two separate inertial weights participating in these movements. The second jet link of model considers coherence of its fluctuations in a cross-section plane with shifting jerks of elements of a shafting from a main drive shaft of a gear-box to a final drive (on Fig. 1 weight 23 with communications). The jet moment influences on the case, aspiring to turn it relative to a longitudinal axis and depending on a reduction ratio in a gear-box.

Coherence of longitudinal-angular fluctuations of the case on a frame with shifting jerks of a site of the shafting from a final drive to driving wheels is considered by the third oscillatory circuit (on Fig. 1 weight 24 with communications). Linear ruggedness of fastening of the case of transmission to a frame [6] is counted and reduced to angular ruggedness concerning cross-section and longitudinal shafts. Values of moments of inertia of the weights participating in fluctuations concerning these shafts are calculated by a technique [7] by means of program complex AutoCAD.

The tractor framework is presented in the form of four inertial weights connected by the differential. The moment of inertia of a framework concerning a longitudinal axis is displayed by weight 34. The engine and the transmission case form an oscillatory circuit with this weight on which jet forces operate at angular oscillations in a cross-section plane with this weight.

Moment of inertia of a framework of a tractor with longitudinal-angular oscillations on a suspender is displayed by weight 27 on the scheme. The jet moment is transferred to weight through the transmission case, aspiring to turn a framework on some angle at tractor oscillations in a longitudinal vertical plane. The use of the nonlinear characteristic of a suspender at longitudinal-angular oscillations of the tractor, experimentally received in work [8] is built into a model. Forward moving weight of a framework of a tractor is presented by two concentrated weights 29 and 30 which are connected with the help of differential with rotating details of running system (weight 25 and 26) and a ground (weights 31 and 32). Division of weights is made for modelling of dynamic processes in a tractor power transmission at turning movement and slipping.

The moment of inertia of weights 25 and 26 is led to an axis of a driving wheel and is a moment of inertia of the rotating and forward moving details of running system, and the moment of inertia of weights 31 and 32 is a moment of inertia of weight of a ground under a seating. At movement of a tractor without slipping these weights are braked, the moment is transferred to weights 29 and 30 through differential communication. At excess by the moment of the engine of the moment of clutch, the moment part is transferred to weights 31 and 32, and they start moving.

The hinged tool is entered into dynamic model (weight 28). The tool is connected by means of differential with weights 29 and 30, simulating division on boards of a moment of inertia of a forward moving weight of tractor's framework, and also with weight 27 for modelling of communication of longitudinal-angular oscillations of the tool in transport position with longitudinal-angular oscillations of a tractor's framework. The law of change of traction resistance of the tool undertakes from experimental data [9] where the traction effort is presented by harmonious components.

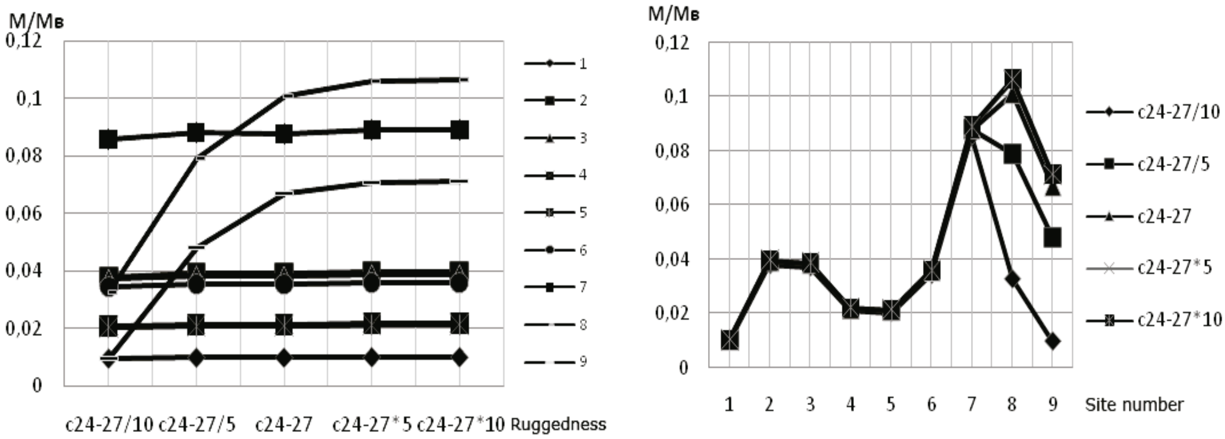
5. Research technique

At model use settlement probe for definition of influence of oscillations on bearing parts of not rotating weights of the engine and transmission on stress loading of power transmission sites is executed at action of indignations from hood loadings, non-uniformity of a twisting moment of the engine, oscillations' of a framework on a suspender, resistance to rolling, regearings of driving wheels with a caterpillar chain and other revolting factors. Stress loading of a shafting from influence of each of factors was thus defined at initial size of cross-section and longitudinal ruggedness of fastening of case-shaped parts and at its increase and reduction in 5 and 10 times. The revolting moment with the set frequency was put to certain weight under the harmonious law with individual amplitude.

For the given engine and transmission, frequency of the revolting moment on driving wheels changes in limits from 10 to 30 Hz on the first drive, from 32 to 36 Hz on the second, from 38 to 42 Hz on the third and from 45 to 48 Hz on the fourth. On the schedules illustrating character of stress loading of a shafting at oscillations with these frequencies, following designations of sites are accepted: 1 – a crankshaft; 2 – a clutch; 3 – a site between a clutch and a cardan shaft; 4 – a cardan shaft; 5 – a gear-box; 6 – a site between a gear-box and a final drive; 7 – a site between a crown-wheel and a planet carrier of a swinging mechanism’s planetary train; 8 – a shaft of the rear bridge; 9 – a site between final drive and a driving wheel.

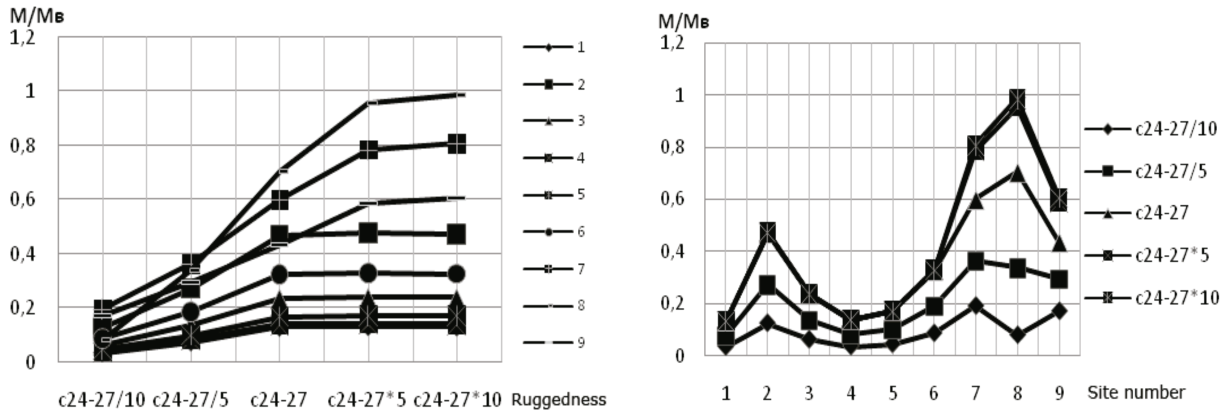
6. Influence of longitudinal ruggedness of fastening of the transmission case on stress loading of its sites

Selective results of definition of the maximum moments on each site of a shafting at the appendix of the individual moment serially to driving wheels with frequencies 10, 24, 26 and 34 Hz from an aforementioned frequency range are resulted in drawings 3 – 6.



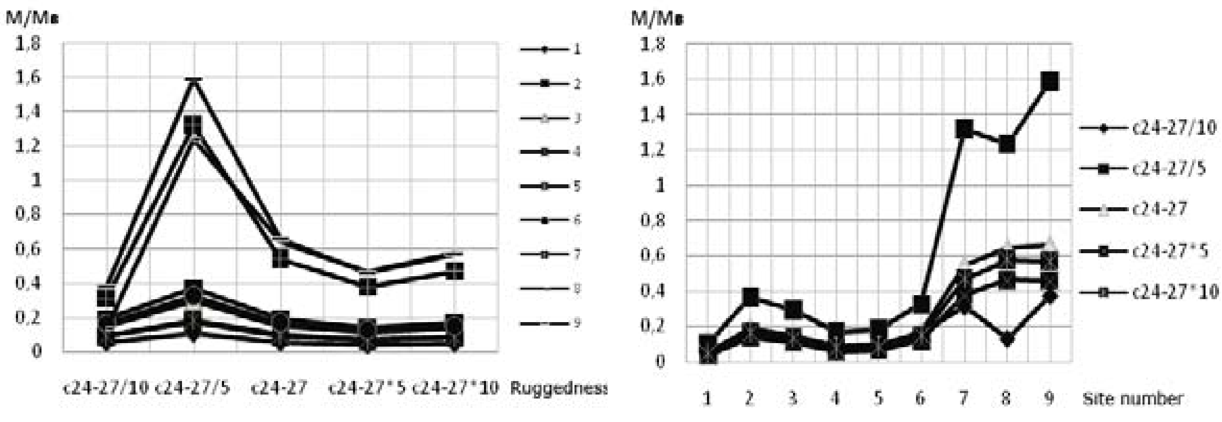
a) *b)*

Fig. 3. Change of stress loading of sites at frequency of indignation of 10 Hz on 1 drive



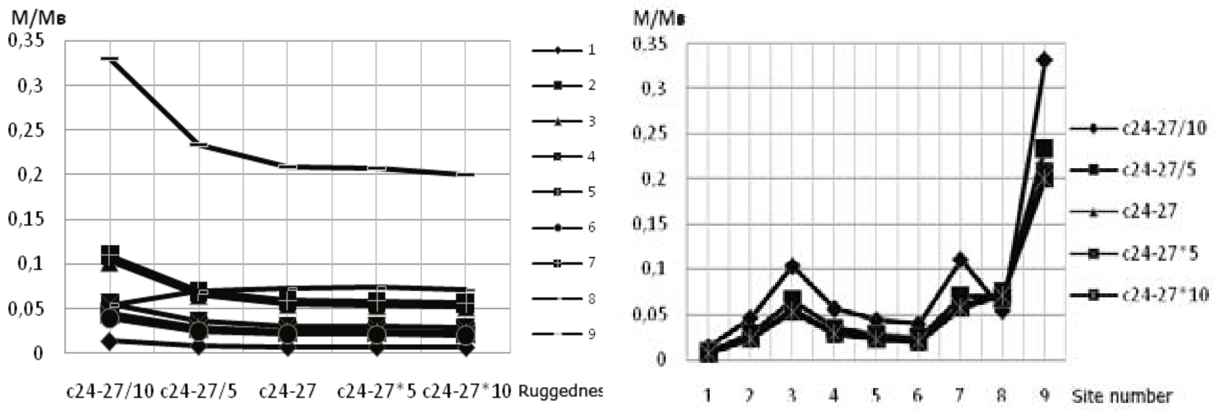
a) *b)*

Fig. 4. Change of stress loading of sites at frequency of indignation of 24 Hz on 1 drive



a) *b)*

Fig. 5. Change of stress loading of sites at frequency of indignation of 26 Hz on 1 drive



a) *b)*

Fig. 6. Change of stress loading of sites at frequency of indignation of 34 Hz on 2 drive

Curves 1 – 9 on Fig. 3 – 6 *a)* illustrate change of stress loading of sites 1 – 9 of a shafting at change of ruggedness of fastening of jet weights. Fig. 3 – 6 *b)* illustrates change of stress loading of sites at passage on them of oscillations of corresponding frequency.

Percentage of the moments on sites at the changed ruggedness of communications of jet weights in 5 and 10 times in relation to experimentally defined, equal 1419500 Nm/rad [15], are specified in table 2.

Tab. 2. Change of stress loading of sites of a power transmission at various longitudinal ruggedness of fastening of transmission to a frame

Frequency, Hz	Ruggedness	Site number								
		1	2	3	4	5	6	7	8	9
10	C/10	-2.9	-2.9	-2.8	-2.8	-2.8	-2.8	-2.3	**-67.7**	**-85.6**
	C/5	0.2	0.2	0.2	0.2	0.2	0.2	0.5	**-21.7**	**-27.9**
14	C/10	**-13.1**	**-13.1**	**-13.1**	**-13.08**	**-13.1**	**-13.1**	**-12.8**	**-70.4**	**-24.5**
	C/5	-5.1	-5.1	-5.0	-5.0	-5.0	-5.0	-5.0	**-25.6**	**-53.8**
18	C/10	**-23.6**	**-23.6**	**-23.6**	**-23.6**	**-23.6**	**-23.6**	**-23.4**	**-73.3**	8.5
	C/5	-8.9	-8.9	-8.9	-8.9	-8.9	-8.9	-8.8	**-28.4**	0.6
22	C/10	**-44.8**	**-44.8**	**-44.8**	**-44.8**	**-44.8**	**-44.8**	**-43.0**	**-80.3**	**-16.0**
	C/5	**-20.7**	**-20.7**	**-20.6**	**-20.6**	**-20.7**	**-20.7**	**-19.4**	**-36.7**	**-7.2**
26	C/10	-0.8	-0.8	-0.8	-0.8	-0.8	-0.8	**-42.4**	**-79.3**	**-43.2**
	C/5	95.2	95.2	95.2	95.2	95.2	95.2	143.8	92.7	141.9
	5C	**-26.8**	**-26.8**	**-26.8**	**-26.8**	**-26.8**	**-26.8**	**-30.5**	**-28.2**	**-30.4**
	10C	**-11.6**	**-11.6**	**-11.6**	**-11.6**	**-11.6**	**-11.6**	**-13.6**	**-10.4**	**-13.6**
30	C/10	249.4	249.4	249.6	249.6	249.6	249.6	448.9	105.4	369.7
	C/5	29.1	29.1	29.1	29.1	29.2	29.2	35.0	7.5	29.4
34	C/10	85.6	85.6	85.8	85.8	85.8	85.8	85.0	**-25.8**	58.0
	C/5	18.0	18.0	18.0	18.0	18.0	18.1	17.5	-5.3	11.9
38	C/10	31.4	31.4	31.4	31.4	31.4	31.4	31.4	**-42.9**	14.6
	C/5	9.0	9.0	8.9	8.9	9.00	9.1	9.1	**-11.0**	4.4
44	C/10	140.0	139.9	140.2	140.2	140.1	140.2	140.2	-6.7	60.9
	C/5	24.0	24.0	24.0	24.0	23.9	23.9	23.9	-0.4	10.8
46	C/10	139.9	139.9	140.2	140.2	140.1	140.2	140.2	-6.7	60.9
	C/5	24.0	24.0	24.0	24.0	24.0	24.0	24.0	-0.4	10.8

In Table 2 the bold type allocates values of the moments on the sites which stress loading essentially decreases in comparison with stress loading at nominal ruggedness. On dark background are shown the relations of the moments on the sites which stress loading considerably increases in comparison with stress loading at nominal ruggedness of communications of jet weights.

At the analysis of the constructed schedules it is revealed that the increase in longitudinal ruggedness in 5 and 10 times at oscillations with all frequencies, except 26 Hz, does not lead to appreciable change of stress loading of sites – it changes no more than on 5 %. Therefore in table 1 influence of increase in ruggedness is shown only on frequency of 26 Hz.

The analysis of oscillations with all considered frequencies shows that influence of longitudinal ruggedness on stress loading of sites is shown as follows:

1. At tractor movement on the first drive with speeds from 1.8 m/s (frequency of regearing of driving wheels with a caterpillar is equal 10 Hz) to 4.32 m/s (24 Hz) reduction of longitudinal ruggedness of fastening conducts to drop of stress loading of the most loaded sites on 13 – 88 %, and the ruggedness more decreases, the stress loading more decreases. Hence, on these modes it is necessary to reduce longitudinal ruggedness of fastening.
2. At movement on speeds from 4.32 m/s to 8.29 m/s (frequency of regearing 24 – 46 Hz) reduction of longitudinal ruggedness leads to increase in stress loading of almost all sites on 20 – 360 %. Together with it, the ruggedness increase leads to insignificant reduction of stress loading and consequently is inexpedient. Hence, on these modes of operation of a tractor longitudinal ruggedness of fastening of transmission to a frame is better be not to changing.

Probes of influence of cross-section ruggedness of fastening of the case of transmission on stress loading of its sites are executed also at excitation of oscillations on driving wheels and from outside the engine. Probes have shown that influence of this ruggedness on stress loading of sites is inessential.

7. Influence of ruggedness of fastening of the case of the engine on stress loading of sites of a power chain

Results of research are shown in Fig. 7. In this figure dependence of stress loading of sites of a shafting of a power train on total influence of harmonious components of a twisting moment with the account is resulted at modelling of cross-section-angular oscillations of a framework of the engine and without their account.

The analysis of the received data says that at the expense of elastic fastening of the engine to a frame, peak dynamic loadings in a power transmission, caused by shifting jerks because of non-uniformity of action of a twisting moment, decrease. Comparison of level of stress loading of sites testifies to it at the account of ruggedness of fastening of the engine (the broken line second from below) and without the account (the uppermost line). At a correct choice of ruggedness of this communication on separate sites of transmission it is possible to lower stress loading on 6 – 40 % that is it is rather considerable.

8. Necessity of changing of elastic characteristics of fastening of case parts to a frame

Results of the researches testify that at the expense of purposeful change of elastic characteristics of fastenings of case-shaped parts it is possible to reach essential drop of stress loading of elements of a power train. Thus the greatest effect can be achieved at the expense of steering of ruggedness of an engine mount and the transmission case in a longitudinal plane.

Power train stress loading depends on ruggedness of an engine mount linearly, therefore it is clear that the more softly its suspender is the more stress loading decreases without dependence from a high-speed mode. More difficult is the problem with a suspender of the case of the transmission which ruggedness should change at change of movement speed.

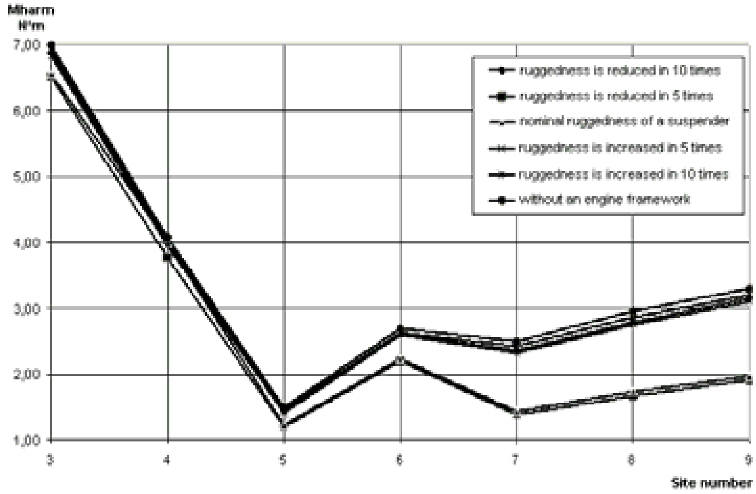
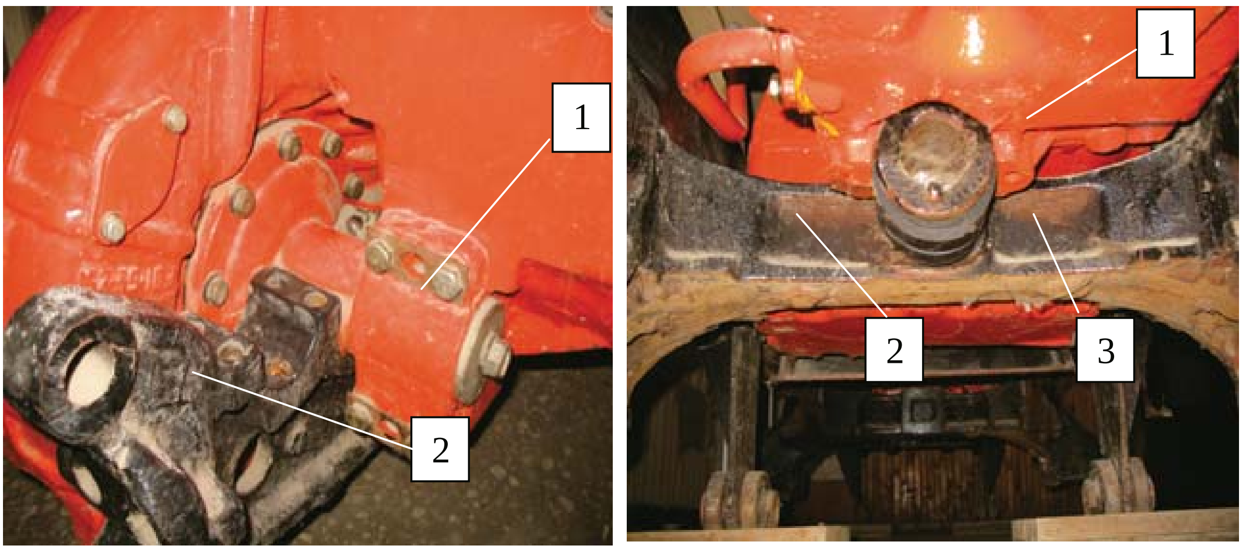


Fig. 7. Change of stress loading of sites of a power train at total influence of harmonious components of a twisting moment of the engine

The transmission case fastens to a frame in three places. These are two symmetrically located bearing parts 1 on rear brackets 2 (Fig. 8, *a*), and one on a wrist pin of 1 rear cross-section wooden block 2 (Fig. 8, *b*). Case of transmission fastens to a wrist pin 1 with a ski lift (on Fig. 8, *b* the ski lift is removed) on two bolts through a rubber gasket 3. Front communication allows reducing dynamic loading both on the case, and on transmission details at longitudinal-angular oscillations (see Fig. 3 – 6) of transmission case within elastic deformation of a layer pad (3-5 mm). Rear bearing parts of the case of transmission are rigid that, however, is dictated by constructive necessity. The pliability of these communications at oscillations can be provided basically only at the expense of elastic deformation of rear brackets 1 (Fig. 8, *a*) and their communications with longitudinal longerons of a frame on which they are welded. In comparison with front ruggedness, ruggedness of rear communications is above on some degrees. However they also define basically ruggedness of communication with a frame of transmission case in a longitudinal plane. Hence, it's necessary to work out actions for perfection of a design of rear bearing parts of transmission which would provide a case rear elastic enough communication with a frame (or would allow to turn on hinges on a certain angle concerning frame brackets) at oscillations and at the same time would allow to keep ruggedness of fastening to the case of transmission of the case of final drive.



a) *b)*

Fig. 8. Transmission installation on a frame: a) on brackets; b) on a cross-section wooden block

Circuit study of one of fastening variants is presented on Fig. 9.

Input in the scheme of additional elastic elements allows the transmission case at jet moment action in a longitudinal plane to turn on some corner concerning a transverse beam axle passing through a shaft of driving wheels, at the expense of deformation of springs 6. Thus ruggedness of fastening of the case to a frame at longitudinal-angular oscillations will be to depend in defining degree on ruggedness of springs. Introduction of a design of fastening with the similar scheme will allow to lower essentially stress loading of some power transmission sites on operational modes.

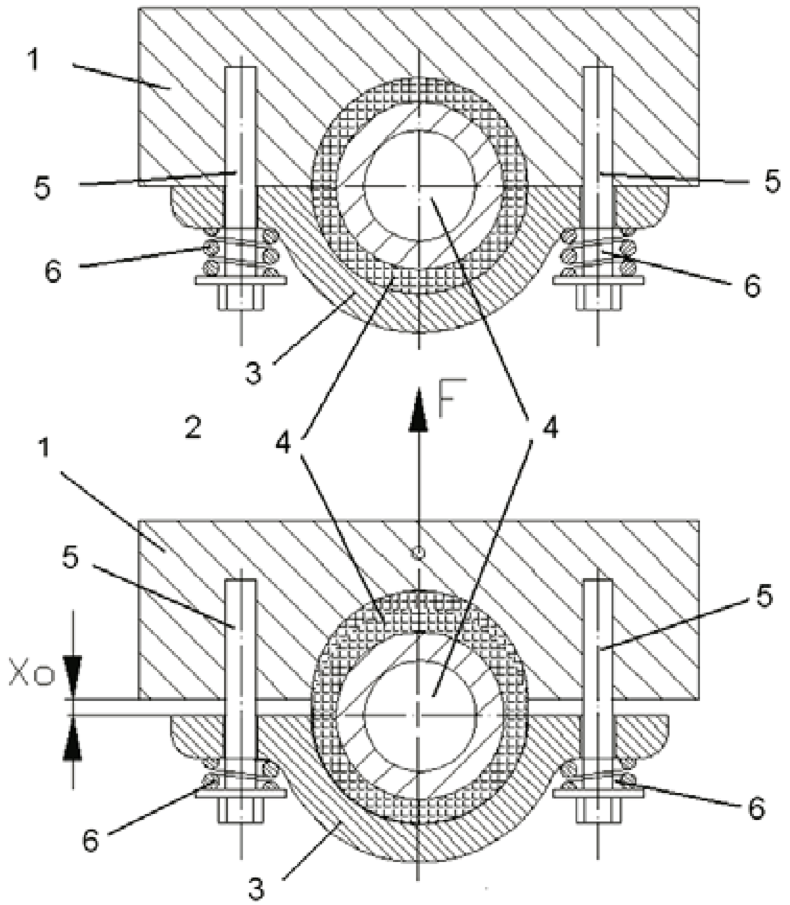


Fig. 9. The scheme of fastening of transmission with additional elastic elements (1 – the transmission case; 2 – a wrist pin of a cross-section wooden block; 3 – a ski lift; 4 – a rubber gasket; 5 – bolts; 6 – springs)

9. Conclusions

1. Dynamic and mathematical models of a power transmission of tractors of public corporation VGTZ of VT family public corporation are created, allowing considering influence on dynamic stress loading of drive of coherence of shifting jerks in transmission with vertical and angular oscillations of case-shaped parts of the engine, transmission and a tractor framework.
2. Settlement researches have shown that reduction of cross-section ruggedness of fastening of the engine allows to lower dynamic stress loading of separate sites of transmission on 6-40 %; change of cross-section ruggedness of fastening of the case of transmission does not influence stress loading of sites, and drop of longitudinal ruggedness at movement with speeds from 1.8 to 4.32 m/s leads to stress loading drop on 13-88 %. The circuit decision of fastening of the case of transmission with the reduced longitudinal ruggedness is offered.

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