

## THE RESEARCH OF THE DYNAMIC LOAD OF THE POWER TRAIN OF THE CATERPILLAR TRACTOR CHETRA 6C-315

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### **Abstract**

*The paper describes the developed dynamic model of the caterpillar's power transmission. Its advantage is the integrated dimensional model of caterpillar propeller. Animated motion modelling provides the formation and transfer of kinematic and dynamic disturbances from caterpillar chassis, causing the greatest loadings of the power circuit sections.*

*The principle of power transmission mathematical model's differential equations forming is described. Result of the model's computational research is a set of oscillograms of the drive wheel's torque changing for different operating modes. The values of power transmission masses' natural frequencies are obtained.*

*The research of the nature of transmission sections' dynamic loading by torsional vibrations, that arise and spread on the shafting, when the linked track rewinds, is carried out. The diagrams show that local maximums of dynamic coefficients characterize the regimes with high hook load. The increase of torques for these sections happens because of approximation of the peculiar frequency of the system and the frequency of one of the harmonic components of stimulating actions. At this peak, value of the oscillation belongs to the masses, the partial frequencies of which are closest to the frequencies to the stimulation actions. Dynamic load of such sections is quite high. The material of these sections often gets fatigue damage, which can lead to breakdown. The analysis of the diagrams, made for basic operation speeds between 1,0 – 5,5 m/sec showed, that maximal peak values of dynamic coefficients appear on sections between 10 and 59 masses in the transmission. Maximal dynamic load of these sections appear at hook load from 0 to 40 kN.*

**Keywords:** *dynamic load, power train, dynamic model, torsional vibration, caterpillar tractor.*

### **1. Introduction**

It is widely known that crawler track is the generator of most dynamic load, which is transmitted from the chassis and suspension to the power train. Many authors when researching the dynamic loads on different parts of transmission suppose that torsion torque of traction wheel changes according to the harmonic rule, which is extremely simple understanding of the real life. Such admission can be explained by the fact of the difficulties of mathematical description of the real-life model of avalanche of the track, road wheels, traction wheel, considering the influence of ground and suspension. In real life the alteration of the torsion torque on the traction wheel depends on mass-inertia, elastic and dissipative parameters of the driving device, coordinates of interacting parts of the driving device and the suspension, profile of teeth of the tracking wheel and the structure of tracks, straining force of the track, speed of movement of caterpillar vehicle, physical and mechanical properties of ground and some other kinematic and dynamic properties. So, the model of the power train, which is the most close to the real life, should take into account the load of all these factors on the parts of the power train.

## 2. Model of the power train and the caterpillar

It is known [1, 3, 4, 6 and others], that during exploitation the power train is influenced by a complex of dynamic loads, which are caused by irregularity of the power of resistance to roll-over and draw-bar resistance, vertical and angular oscillation of the base on the suspension and the over-training of teeth with caterpillar. At that, the frequencies of indignations can become the same with the frequencies of the power train, which leads to the resonance and heavy increase of dynamic load for power train.

To accomplish the research of dynamic loads of main parts of the power train in the real-life exploitation the department of “Auto and tractor industry” in cooperation with Main Special Design Bureau “Promtractor” LLC developed dynamic model of the transmission of caterpillar tractor Chetra 6C-315. Fig. 1 presents the structural scheme of the power train, which was used to develop this model. The model includes 98 discrete mass with elastic and frictional bonds. Dynamic parameters of its elements were identified with the aid of calculations based on the corresponding methodology.

To describe the movement of the mass model on the basis of Lagrange's equations of the 2<sup>nd</sup> order we developed the mathematical model. Below you can see an example of description of 6<sup>th</sup> oscillating circuit's mass movement (Fig. 1), i. e. mass of the differential gear of turn, with the aid of the following system of difference equations:

$$\left\{ \begin{array}{l} J_{60}\ddot{\varphi}_{60} + k_{60}\dot{\varphi}_{60} + C_{\sum(60-64-66)}(\varphi_{60} - \frac{i_{p.r.}}{1-i_{p.r.}}\varphi_{66} - \frac{1}{i_{p.r.}}\varphi_{64}) - C_{59-60}(\varphi_{60} - \varphi_{59}) = 0; \\ J_{62}\ddot{\varphi}_{62} + k_{62}\dot{\varphi}_{62} + C_{62-66}(\varphi_{66} - \varphi_{62}) + C_{62-64}(\varphi_{64} - \varphi_{62}) - C_{60-62}(\varphi_{62} - \varphi_{60}) = 0; \\ J_{64}\ddot{\varphi}_{64} + k_{64}\dot{\varphi}_{64} + C_{64-91}(\varphi_{91} - \varphi_{64}) - C_{\sum(60-64-66)}(\varphi_{60} - \frac{i_{p.r.}}{1-i_{p.r.}}\varphi_{66} - \frac{1}{i_{p.r.}}\varphi_{64}) = 0; \\ J_{66}\ddot{\varphi}_{66} + k_{66}\dot{\varphi}_{66} + C_{66-68}(\varphi_{68} - \varphi_{66}) - C_{\sum(60-64-66)}(\varphi_{60} - \frac{i_{p.r.}}{1-i_{p.r.}}\varphi_{66} - \frac{1}{i_{p.r.}}\varphi_{64}) = 0; \\ J_{68}\ddot{\varphi}_{68} + k_{68}\dot{\varphi}_{68} + C_{68-70}(\varphi_{70} - \varphi_{68}) - C_{66-68}(\varphi_{68} - \varphi_{66}) = 0; \end{array} \right. \quad (1)$$

where:

$J_i$  – momentum of inertia of discrete masses,

$C_{i-j}$  – torsional stiffness of elastic bonds,

$k_i$  – the coefficients of the buffing of masses' oscillation,

$\ddot{\varphi}; \dot{\varphi}, \varphi$  – the coefficients of acceleration, speed and, movement of masses during oscillation.

The model allows considering the influence of longitudinal and angular oscillations of the body of transmission against the base on the load on the power train, and also the oscillation of the body on the suspension. To do this the model includes two additional links with reactive bonds.

The inertial mass 87 presents the body of the transmission in the oscillative circuit, which includes the elements of the final reduction gear's planetary train. This allows showing the links between longitudinal and angular oscillations of the body on the base and torsional oscillation of the suspension. Final reduction gear of the machine is made as planetary gear with deferred crown gear, the flexibility of the reactive of bonf for which is defined as torsional flexibility of final drive's fixing to the body of transmission.

To analyze the influence of kinematic and dynamic loads to the load of power train, which form the torsion torque of the traction wheels, we developed two variants of a solid model of the caterpillar for the tractor Chetra 6C-315: with 6 individual tracking wheels (Fig. 2) and with 6 tracking wheels and equalizer bars (Fig. 3).

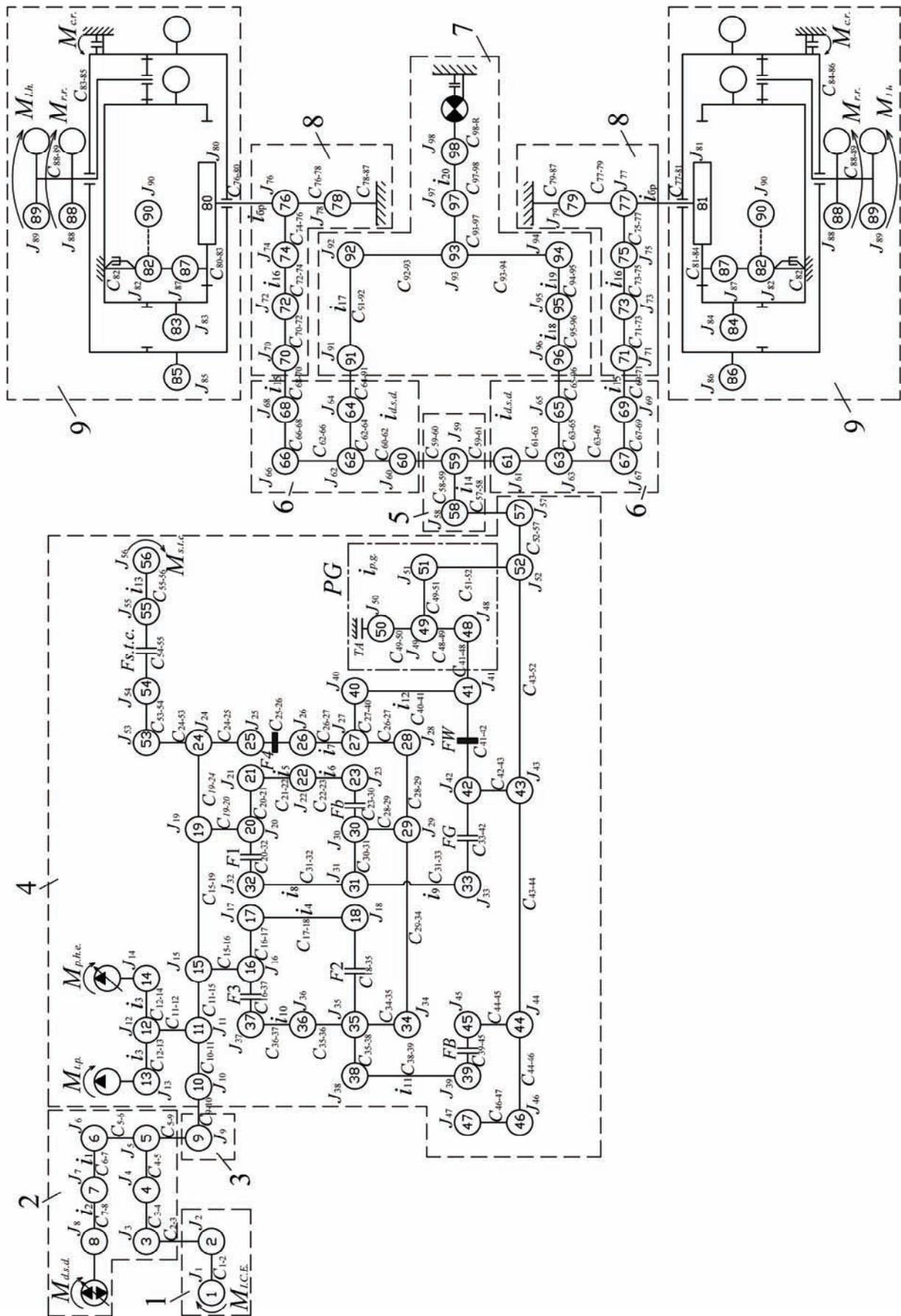


Fig. 1. Structural scheme of the power train of the caterpillar tractor Chetra 6C-315

The models were developed on the basis of modern CAD/CAE system «Universal mechanism 6.0», developed in the Bryansk State Technical University [20]. The same system was used to create the models of tractors with these caterpillars (Fig. 4, 5) and the complex of calculations of dynamic load of power train was carried out. The first phase included the modelling of linear motion of the tractor with constant speed. To describe the particularities of the ground we used the model of Becker taking the sagging into account.

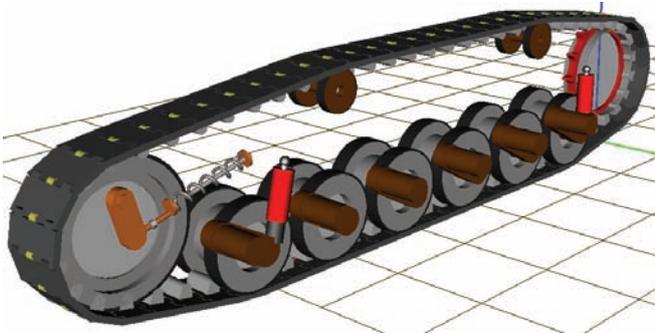


Fig. 2. Model of caterpillar with torsion suspension

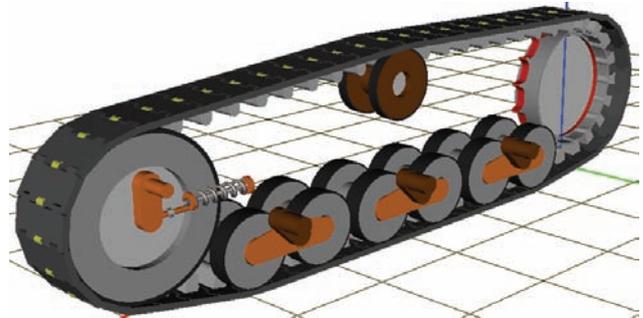


Fig. 3. Model of caterpillar with centrepoint suspension

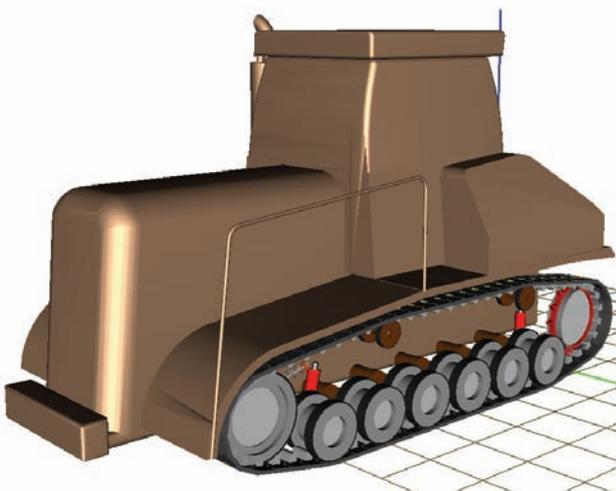


Fig. 4. Model of tractor with torsion suspension

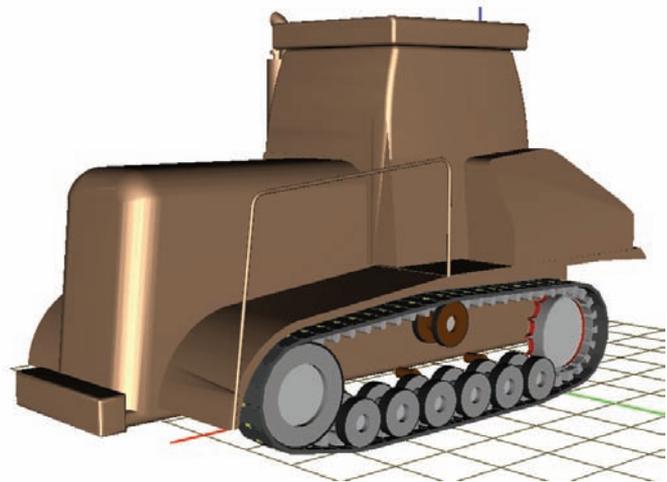


Fig. 5. Model of tractor with centrepoint suspension

### 3. Research results

#### 3.1. The analysis of the changes of momentum on the traction wheel

As a result of modelling we received the dependencies of torsion torque's changes on the traction wheel from the angle of turn for main speeds of motion (from 1.0 till 9.5 m/sec). Some of oscillograms are shown on the figure 6.

More complicated analysis of the changes of torsion torque for main speed between 1.0 till 9.5 m/sec is shown on the 3D diagram (fig. 7), where  $x$ -axis presents the speed of the motion of the tractor  $V$  (m/sec),  $y$ -axis – the angle of turn of the traction wheel (per one tooth),  $z$ -axis – the change of the torsion torque on the traction wheel (N·m).

The received data was used to assign the disturbing action when examining of dynamic loads of transmission, the received dependences were arranged as Fourier's series with the aid of standard libraries of *Matlab* [21].

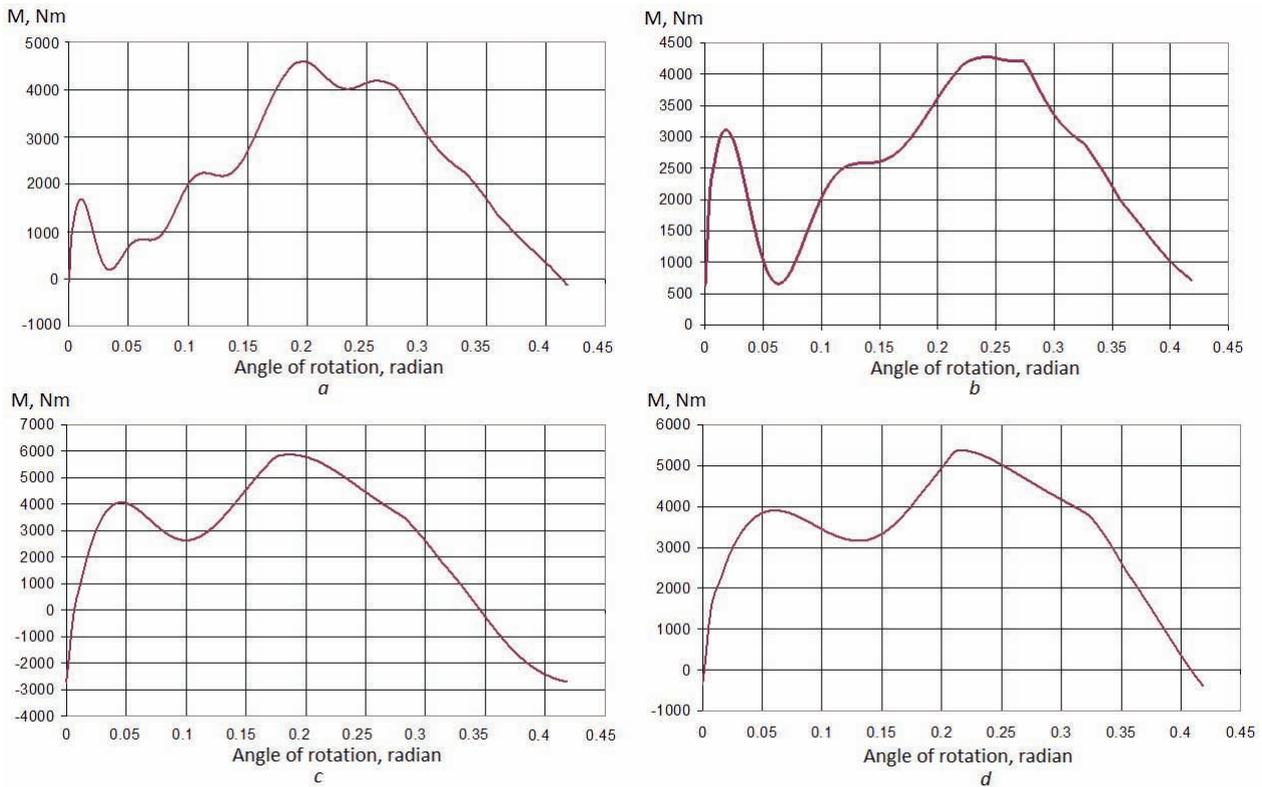


Fig. 6. The changes of the torsion torque for the traction wheel with the speeds: a – 2.5 m/sec; b – 5 m/sec; c – 8 m/sec; d – 9.5 m/sec

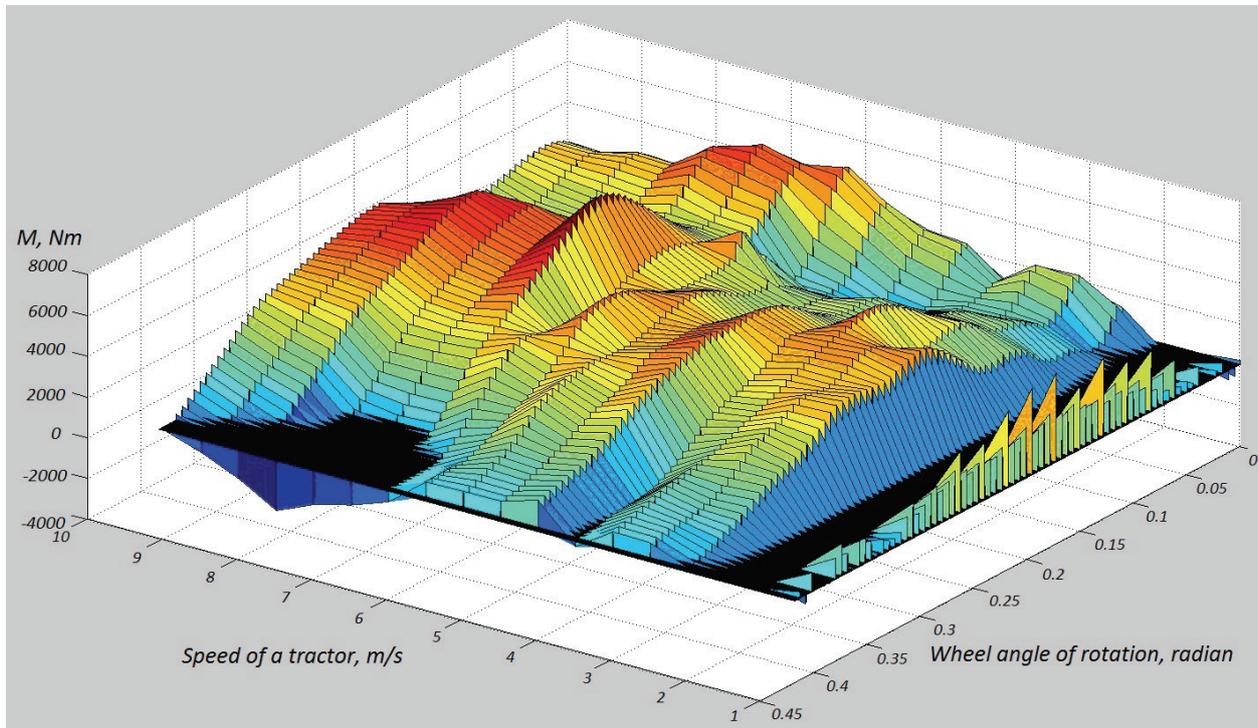


Fig. 7. Changes of the torsion torque for the traction wheel with different speeds of tractor's movement

The results of the conducted research are shown in the Table 1. For each speed of tractor, we received the value of the torque for the traction wheel up to 6<sup>th</sup> harmonic component. Table 1 shows the frequencies of each received harmonic component and the intercourse of the torque at this frequency to the torque at the frequency of the 1<sup>st</sup> harmonic component.

Tab. 1. The results of the conducted research

Speed, m / s km / h	Frequency, Hz Amplitude, Nm	Number of harmonic component					
		(Fnd)1	h2	h3	h4	h5	h6
		Frequency, Hz Relative amplitude, %					
<u>1.0</u> 3.6	<u>6.53</u> 1211.62	<u>6.53</u> 100%	<u>13.06</u> 10.19%	<u>19.59</u> 9.4%	<u>26.12</u> 6.5%	<u>32.65</u> 4.68%	<u>39.18</u> 3.41%
<u>1.5</u> 5.4	<u>9.79</u> 1511.38	<u>9.79</u> 100%	<u>19.59</u> 4.59%	<u>29.38</u> 8.92%	<u>39.18</u> 3.81%	<u>48.975</u> 3.06%	<u>58.77</u> 2.3%
<u>2.0</u> 7.2	<u>13.06</u> 2589.86	<u>13.06</u> 100%	<u>26.12</u> 17.04%	<u>39.18</u> 8.92%	<u>52.24</u> 3.86%	<u>65.3</u> 1.14%	<u>78.36</u> 5.67%
<u>2.5</u> 9.0	<u>16.33</u> 1945.5	<u>16.33</u> 100%	<u>32.65</u> 7.67%	<u>48.97</u> 4.78%	<u>65.3</u> 4.19%	<u>81.625</u> 10.94%	<u>97.95</u> 2.02%
<u>3.0</u> 10.8	<u>19.59</u> 1502.74	<u>19.59</u> 100%	<u>39.18</u> 16.95%	<u>58.77</u> 8.36%	<u>78.36</u> 16.9%	<u>97.95</u> 9.11%	<u>117.54</u> 5.25%
<u>3.5</u> 12.6	<u>22.86</u> 1628.97	<u>22.85</u> 100%	<u>45.71</u> 19.76%	<u>68.56</u> 13.87%	<u>91.42</u> 7.71%	<u>114.27</u> 9.39%	<u>137.13</u> 11.11%
<u>4.0</u> 14.4	<u>26.12</u> 1791	<u>26.12</u> 100%	<u>52.24</u> 24.29%	<u>78.36</u> 8.5%	<u>104.48</u> 5.03%	<u>130.6</u> 12.62%	<u>156.72</u> 7.5%
<u>4.5</u> 16.2	<u>29.39</u> 1785.03	<u>29.39</u> 100%	<u>58.77</u> 12.53%	<u>88.15</u> 8.22%	<u>117.54</u> 11.96%	<u>146.92</u> 11.36%	<u>176.31</u> 6.49%
<u>5.0</u> 18	<u>32.65</u> 1276.35	<u>32.65</u> 100%	<u>65.3</u> 21.8%	<u>97.95</u> 10.47%	<u>130.6</u> 13.84%	<u>163.25</u> 8.12%	<u>195.9</u> 3.06%
<u>5.5</u> 19.8	<u>35.92</u> 1181.27	<u>36.5</u> 100%	<u>71.83</u> 17.59%	<u>107.745</u> 36.82%	<u>143.66</u> 23.31%	<u>179.57</u> 13.43%	<u>215.49</u> 6.67%
<u>6.0</u> 21.6	<u>39.18</u> 1328.11	<u>39.18</u> 100%	<u>78.36</u> 54.19%	<u>117.54</u> 26.79%	<u>156.72</u> 25.19%	<u>195.9</u> 10.63%	<u>235.08</u> 8.68%
<u>6.5</u> 23.4	<u>42.45</u> 1597.40	<u>42.45</u> 100%	<u>84.89</u> 54.75%	<u>127.33</u> 56.01%	<u>169.78</u> 18.30%	<u>212.22</u> 12.66%	<u>254.67</u> 4.95%
<u>7.0</u> 25.2	<u>45.71</u> 3290.08	<u>45.71</u> 100%	<u>91.42</u> 29.25%	<u>137.13</u> 30.81%	<u>182.84</u> 14.18%	<u>228.55</u> 9.87%	<u>274.26</u> 6.81%
<u>7.5</u> 27	<u>48.98</u> <u>2589.83</u>	<u>48.98</u> 100%	<u>97.95</u> 49.18%	<u>146.92</u> 37.44%	<u>195.9</u> 17.49%	<u>244.87</u> 9.91%	<u>293.85</u> 5.70%
<u>8.0</u> 28.8	<u>52.24</u> 3002.39	<u>52.24</u> 100%	<u>104.48</u> 32.14%	<u>156.72</u> 27.46%	<u>208.96</u> 10.22%	<u>261.2</u> 4.16%	<u>313.44</u> 1.89%
<u>8.5</u> 30.6	<u>55.51</u> 2454.32	<u>55.51</u> 100%	<u>111.01</u> 28.61%	<u>166.51</u> 22.25%	<u>222.02</u> 4.13%	<u>277.52</u> 2.06%	<u>333.03</u> 1.47%
<u>9.0</u> 32.4	<u>58.77</u> 1824.78	<u>58.77</u> 100%	<u>117.54</u> 35.02%	<u>176.31</u> 16.19%	<u>235.08</u> 3.16%	<u>293.85</u> 1.57%	<u>352.62</u> 2.02%
<u>9.5</u> 34.2	<u>62.04</u> 1423.29	<u>62.04</u> 100%	<u>124.07</u> 41.62%	<u>186.10</u> 17.56%	<u>248.14</u> 3.6%	<u>310.17</u> 2.5%	<u>372.21</u> 1.81%

### 3.2. The spectrum of the peculiar frequencies of the power train

To receive the spectrum of the peculiar frequencies we recorded the oscillograms of the elastic torques on the sections of free system's oscillation after short impulse action. At this, we made the admission about the absence of friction in the system. Each of the oscillograms was analyzed with Fourier analysis with the aid of standard libraries of *Matlab*. The examples of oscillograms for different sections are shown on the Figure 8.

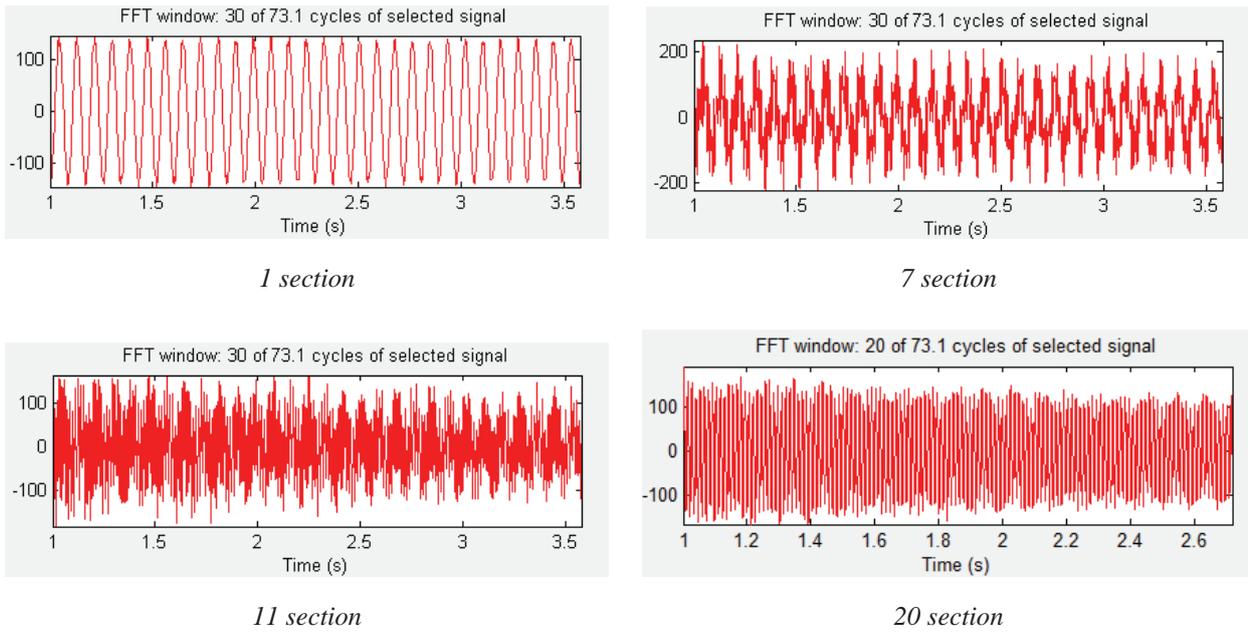


Fig. 8. Oscillograms of the elastic torques on the sections

The peculiar frequencies (0-300 Hz) were determined by peak value of Fourier density (Fig. 9).

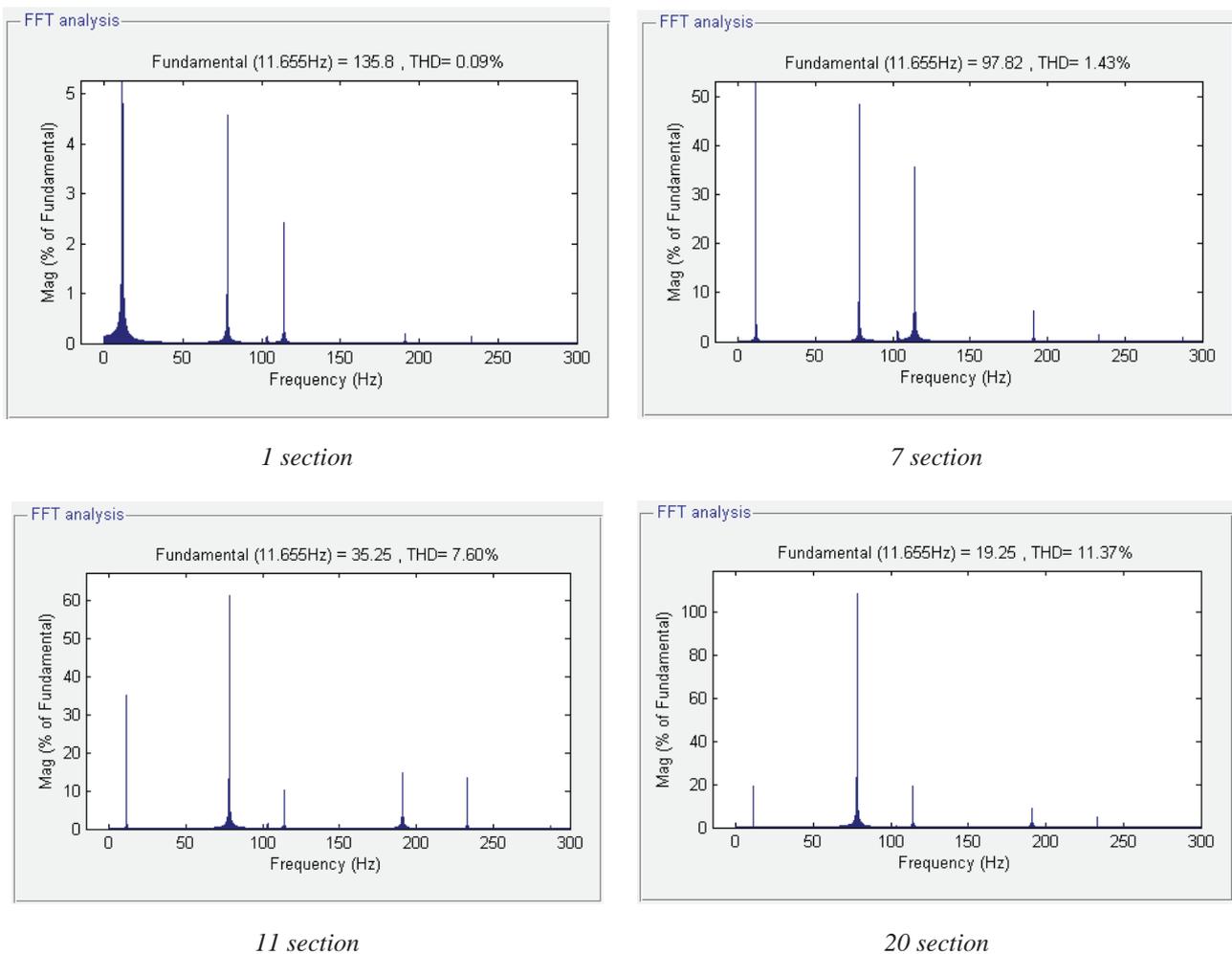


Fig. 9. Fourier density of torque on the sections

The received peculiar frequencies are shown in the Table 2.

Tab. 2. Peculiar frequencies of the power train

No.	1	2	3	4	5	6	7	8
Frequency. Hz	11.65	78.33	103.17	114.25	149.61	191.06	233.29	287.36

### 3.3. Finding of the coefficients of the dynamic loads on the sections

General operation regime for the tractor's transmission can be defined as a sum of dynamic and quasi-static components of the torsion torque [1]. Static component can be easily found if you know the speed characteristic of the engine and the conditions of interaction of ground and caterpillar.

The dynamic component is defined with the aid of dynamic coefficient  $k_{дБ}$ , which can be found with the following formula:

$$k_{дБ} = 1 + \frac{P_A}{P}, \quad (2)$$

where:

$P$  – circumferential force, which corresponds to the ultimate moment,

$P_A$  – average dynamic load [2].

Circumferential force can be found with the formula:

$$P = \frac{2M}{mz}; \quad (3)$$

where:

$M$  – ultimate moment,  $m$  – average modulus,  $z$  – number of teeth of the wheel,

$$P_A = \sqrt{P_a(2 \cdot D - P_a)}; \quad (4)$$

$$P_a = \frac{P_1 \cdot P_2}{P_1 + P_2}, \quad (5)$$

where:

$P_1$  – power, emerging after catching if there is no elastic deflection of teeth, trains and so on;

$P_2$  power, emerging after catching at smooth running of gear wheels and whipping of teeth at step's error rate.

Dynamic factor  $D$  is defined with the formula:

$$D = \Delta \cdot b_c \cdot C \cdot \cos^2 \beta, \quad (6)$$

where:  $\Delta$  – calculated production error of the gear wheel;

$C$  – modulus to density of teeth;

$b_c$  – average width of ring gear;

$\beta$  – gearing angle.

The program complex, which was used for this research, allows to get oscillograms of torsion torques of each section of power train and to determine the max and min dynamic components, and also the average value of the static component.

### 3.4. Analysis of the load on sections of striking interaction of traction wheels and caterpillar

To fulfil the analysis we made the following admissions:

1. The engine speed does not change if the extra loads increase.
2. Hook load is constant and does not change during the modelling.
3. Tractors move evenly and constantly, on the flat ground, there are no longitudinal and angular, vertical and other oscillations.
4. Resisting torque for the traction wheels is determined beforehand and it is chosen from the database depending on the speed of movement.
5. The frequency of resisting torque's change is proportional to the set speed.

The research is carried out for all the set hook loads (0 – 80 kN) with the step 4 kN and full range of set speeds (1 – 9.5 m/sec) with the step 0.5 m/c. As a result, we received the set of oscillograms of elastic torques for those sections of transmission, which transmits the torsion torque. Most common examples are shown on the fig. 10-11. When analyzing we found out average statistic and max dynamic torque for each section. As a result, the dynamic coefficient and average torques were defined for all set regimes of operation. The values of dynamic coefficient and average torque for some sections are shown on fig. 12 and 13.

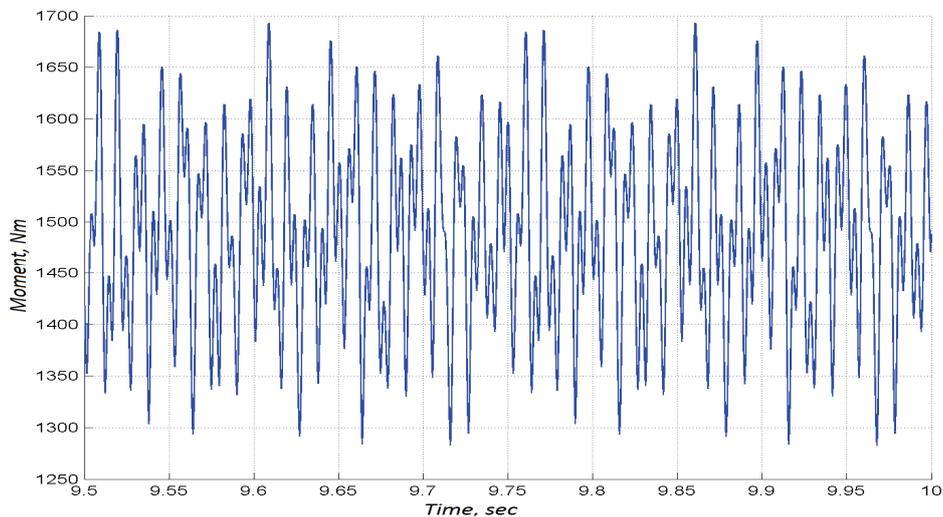


Fig. 10. Oscillogram of the torque on section 3-4 at the speed of motion  $V=3$  m/sec with hook load  $F_{hl}=70,12$  kN

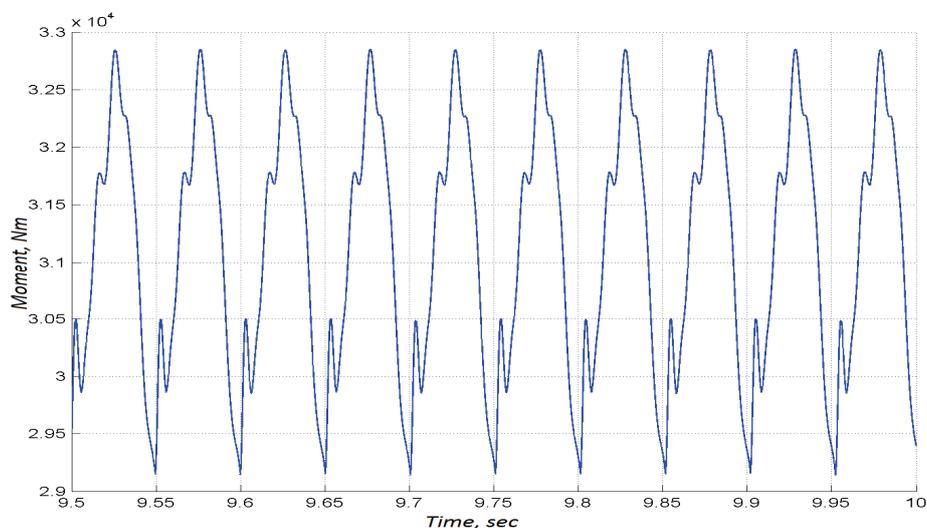


Fig. 11. Oscillogram of the torque on section 59-65 at the speed of motion  $V=3$  m/sec with hook load  $F_{hl} =70,12$  kN

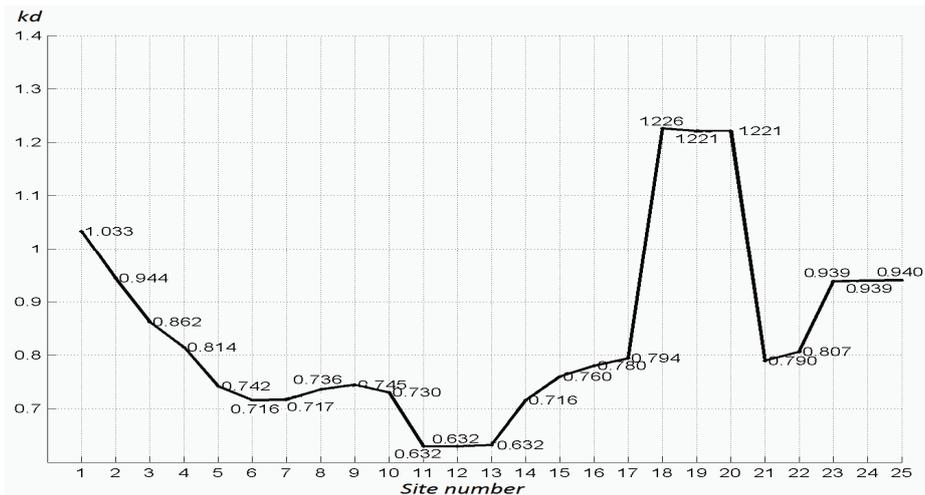


Fig. 12. Dynamic coefficient values on the sections at the speed of motion  $V=3$  m/sec with hook load  $F_{hl}=70,12$  kN

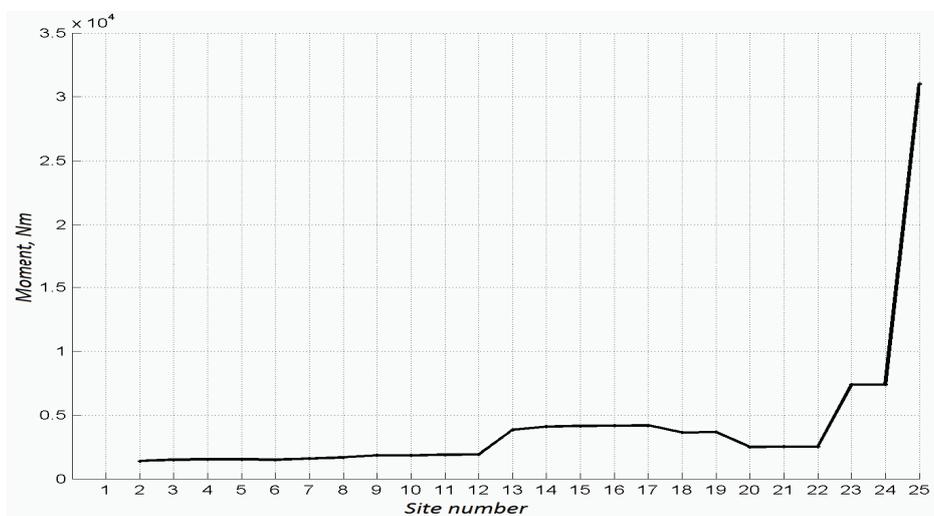


Fig. 13. Values of the average torque of elastic power on sections at the speed of motion  $V=3$  m/sec with hook load  $F_{hl}=70,12$  kN

The received data allowed making 3D diagrams, which had the full information about dynamic and average static load of sections of transmission at set regimes of operation. Some of these 3D diagrams are shown on the fig. 14-15.

The 3D diagrams show clearly the local maximums of dynamic coefficient for some regimes of loading. The most extreme values are found for the regimes with the minimal hook load. This can be explained by the fact that for these sections the maximum values of elastic torques because of torsional oscillation are much higher than the transmitted torque, which is low at low hook load.

The diagrams show that local maximums of dynamic coefficients characterize the regimes with high hook load. The increase of torques for these sections happens because of approximation of the peculiar frequency of the system (Table 2) and the frequency of one of the harmonic components of stimulating actions (Table 1). At this peak, value of the oscillation belongs to the masses, the partial frequencies of which are closest to the frequencies to the stimulation actions. Dynamic load of such sections is quite high. The material of these sections often gets fatigue damage, which can lead to breakdown.

The analysis of the diagrams, made for basic operation speeds between 1,0 – 5,5 m/sec showed, that maximal peak values of dynamic coefficients appear on sections between 10 and 59 masses in the transmission. Maximal dynamic load of these sections appear at hook load from 0 to 40 kN.

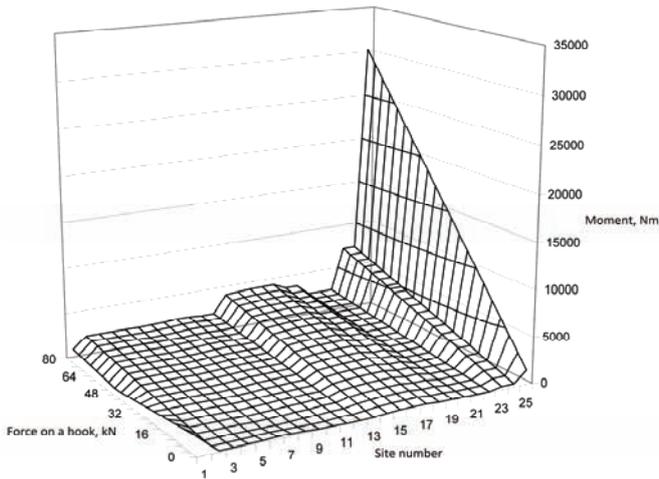


Fig. 14. Value of average torque of elastic power on the sections at the speed of motion  $V = 1$  m/sec

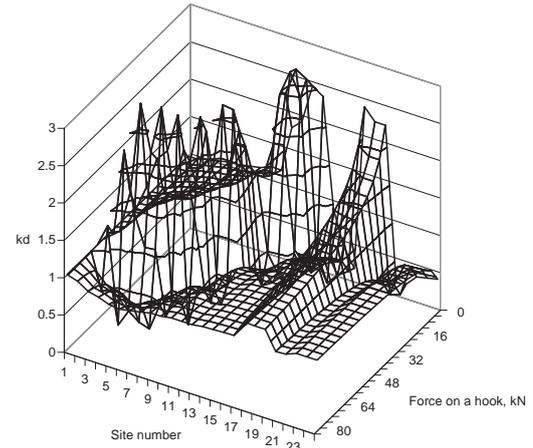


Fig. 15. Value of dynamic coefficient on the sections at the speed of motion  $V = 5,5$  m/c

#### 4. Main results

1. We developed the dynamic model of the power train of the caterpillar tractor Chetra 6C-315, which allow examining the dynamic load of the sections at different regimes of operation. The model is advantageous in comparison with other models because it includes the spatial model of the caterpillar. The animating modelling of motion for this model includes the forming and transmission of the complex of kinematic and dynamic indignations, which lead to maximal dynamic load of the power chain.

2. We made some research of the model. We got the set of oscillograms of torsion torque change on the traction wheel for different regimes of operation. We defined the peculiar frequencies of oscillation of power train's masses. We also fulfilled a research of dynamic load on transmission sections depending on the torsional oscillation, which appear and are transmitted through shaft line when moving the caterpillar.

3. The results of the research allow estimating the circumstances of operation of transmission of designed tractor at different internal and external loads. The results allow specifying the project calculations of parts, and also recommending the measures to lower dynamic loads on the transmission.

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