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WORKLOAD ESTIMATION OF HARVESTERS DURING THE OPERATIONS OF WORK CYCLE

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Abstract. The research is devoted to the investigation of working efficiency and workload of harvesters. The complex criterion of their efficiency evaluation is suggested. The estimation of operational effectiveness of the harvester MLH-414 (*МЛХ-414*) in various working conditions has been made. The various styles of limbing have been analysed. The recommendations for the efficient application of a work mix have been given. Mathematical modelling of harvester loading during transport and engineering operations has been carried out. The adequacy of the designed model of the actual harvester structure has been verified. The most overloaded operations of the harvester's working cycle are defined. The ways for decreasing the emerging loadings are offered.

Keywords: harvester; modelling; dynamics; oscillations; loads.

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Introduction

In research field of dynamics of logging machines including ones based on articulated chassis, there are many wildly known works of scientists from Sweden, Finland and Japan.

In the work (Azad 2006), motion stability for articulated skidders with pile monkey in transfer operations has been analysed. Dependencies of skidder stability on various mass and geometrical parameters of chasses, peculiarities of its arrangement and travelling speed and various angles of semi-frame folding have been determined. The author makes a parallel analysis by means of a mathematical formulation of the dynamic system with the following solution by means of computers and CAD system *Adams*.

The work (Heinze 2007) dwells on the detailed examination of manipulator performance dynamics of a forwarder and considers a mathematical model of manipulator performance of forwarder Rottne (*http://www.rottne.com*). The author distinguishes hydraulic cylinders of arm, handle and telescopic link as well as metalware of arm and manipulator handle as resilient members. A modelling of the manipulator dynamics is being carried out considering the kinematic ratios between manipulator links. An examination of dynamics processes is done for a rigidly fixed manipulator not taking into account elastic properties of the wheeled chasses. The mathematical formulation of the model is based on the Lagrange equation, and the obtained differential equations are done in the system *MatLab*. The work also gives the model source code. To estimate the model adequately, the author has made an experimental research of the dynamics of manipulator that is fixed on rigid metal support. The characteristics being researched are the following: kinematic locations of manipulator links, pressure in hydraulic cylinder of arm, loads arising on the manipulator tip that have been measured by means of pressure and acceleration sensors.

Theoretical and experimental research of manipulator dynamics has been considered in the work by Löfgren (2009). When modelling, the author takes into account performance kinematic of articulated

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manipulators and peculiarities of telescopic links structures. The specific feature of experimental research was the installation of turn sensors in manipulator flexible joints. The metreage has been compared to the analogues data obtained in the result of mathematical modelling. It should be noted that experimental research has been carried out on a manipulator fixed directly on a forwarder chassis (load of specified weight was lifting and dropping when researching).

The works (Papadopoulos, Sarkar 1996; Papadopoulos *et al.* 1997) are devoted to the modelling of dynamic loading of manipulating logging machinery with 'rigid frame'. The peculiarity of the mathematical models of loader and harvester is in the coupling between electrohydraulic system of manipulator control and its bearing structure that allows analysing their performance dynamics under various control effects of the operator. The suggested models are two-dimensional and developed for vehicles with wheel arrangement 4×4.

The joint work of scholars from the Royal University of Technology (Sweden) dwells on the research of moving a forwarder articulated structure loading (Widéen *et al.* 2011). The authors consider the structure with a hinge having vertical, longitudinal and lateral horizontal pivot centres. In the system of final element analysis *Ansys*, there has been a researched deflected mode of bearing structure and an articulation pivot of semi-frame. The determination of existing loading modes is done by pre-modelling in the system *MathLab*. The options of machine loading when turning and direct moving have been studied. Herewith, the authors highlight only the static force factors without considering dynamic loading of the structure, and they do not analyse engineering loading in locked hinge.

Generalising the above written matters, it should be concluded that scientists have made a huge work in the field of parameters substantiation for logging machinery, in the research of their operational performances and dynamic loading in various operational conditions. But familiar loading research of the bearing structures of articulated logging machines is based on the examination only transfer operations. This is not enough for loading estimation of structures of multioperational manipulating machines and for establishing rational loading and operational modes of their performance.

The peculiarities of operational conditions and labour subject, a vast number of performed engineering operations and their duration, the presence of articulation pivot of semi-frames and systems of its block, the specific features of turning mechanisms and the different support structures of the manipulator determine the main differences in modelling of dynamics of manipulating logging machinery on articulated chassis. This kind of machines is considered to be the most promising logging vehicles today.

Materials and methods

The working cycle operations of harvesters considerably differ in duration (Sirén, Tanttu 2001; Sen'kin 2006; Teterina 2009; Spinelli, Visser 2008; Wang, Greene 1999; Eliasson *et al.* 1999), energy content (Asmolovskij, Klokov 1996; Rakut'ko 2008; Fedorenchik *et al.* 1997; Gellerstedt 2002) and the influence on machine elements (Budevich 2006; Snopok 2008; Papadopoulos, Sarkar 1996). These characteristics are affected by the ways of performing working cycle operations, natural and industrial operating conditions and specification of harvesters.

The choice of machines' effective parameters and the ways of performing working cycle operations can be done with the help of assessment criterion of tandem machines effectiveness considering the dynamic loading on their elements (Goronovskij, Golyakewich 2010).

The criterion of Energy Potential of Efficiency (EPE) is developed for the evaluation of multioperational logging machines' effectiveness. This criterion is determined as the ratio of useful power N to the duration of engineering cycle operations T_c^h.

Similar criterion has been used before for the evaluation of unioperational transport road-building machines (Ginzburg *et al.* 1986; Zhukov *et al.* 1989). It was determined by the expression of the useful work performed by the tractor per unit of time, considering portion of working operations during the total time of the cycle.

It should be noted that harvesters are multioperational machines. On moving operations they don't perform useful work. Energy efforts on their engineering operations vary within a wide range. That's why the value of useful power should be included in the original expression for EPE determination as the product of mathematical expectation of its constituents:

$$EPE = \left(M_1^h \cdot \omega_1^h \cdot t_1^h + M_2^h \cdot \omega_2^h \cdot t_2^h + F \cdot v \cdot t_4^h + M_3^x \cdot \omega_3^x \cdot t_3^h \cdot n_1\right) \cdot \frac{n_2}{T_c^h}, \quad (1)$$

where: t_1^h – time needed for cutting a tree; t_2^h – time needed for logging a tree to the place of bucking; t_4^h – time needed for limbing; t_3^h – time needed for bucking; M_1^h, ω_1^h – rotary moment of saw tire thrust on a tree and its angular velocity while felling a tree; M_2^h, ω_2^h – rotary moment of manipulator and its angular velocity; F, v – strength and velocity of grabbing a tree through the harvester head; M_3^h, ω_3^h – rotary moment of saw tire thrust on a tree and its angular velocity while bucking a tree; T_c^h – the total time of harvester working cycle; n_1 – a number of kerfs for bucking of one tree trunk; n_2 – a number of trees being treated from the one side of engineering stand which is dependent on the type of cutting and the density of crop.

The strength, F, needed for grabbing a tree feeding it by the mills of harvester head, is determined as the sum of engineering resistance forces to drag a tree (Fedorenchik, Turlaj 2002), dragging resistance F and tree's inertial force F_{in}. Due regard for the tree's inertial force is essential for large tree trunk volume, where it influences much on the value of the maximum of dragging velocity.

$$F = \frac{P_l + (q + Q_w) \cdot \mu + F_c}{2} + F_r + F_{in}, \qquad (2)$$

where: P_l – resistance force to limbing in harvester head; q – total pressure of rolls on the tree being treated; Q_w – weight of the tree; μ – friction factor of trunk rolling along feeding rolls of the head; F_r – resistance force in the roll journals.

The ability to perform the engineering operations of various styles depends on the operator's skills. Thus, experienced harvester operators apply oncoming feeding of a tree to harvester head during limbing.

In case of dragging a tree with a work mix, forces F_r and F_{in} do not affect the total dragging force because the tree is in rest. Determining the power used for limbing in a work mix, it is much more efficient to calculate the rotary moment of manipulator and the angle velocity of its turn.

Comparative efficiency of such a work mix for harvester MLH-414 (*МЛХ-414, http://www.belarus-tractor.com*) is shown in Fig. 1 (the comparison is given with the arm of crane of 5 m).

For EPE function describing the harvester functioning with the combination of operations, it is taken into account that after oncoming feeding of harvester head to a tree, it is necessary to skid the tree to the place of bucking, and this operation cannot be regarded as useful work but only increases the time of the cycle.

The change in working efficiency in pine forest stand with the estimated productivity of the second class (where the trunk volume from 0.10 to 0.80 m^3) varies greatly. So when the tree trunk is less than

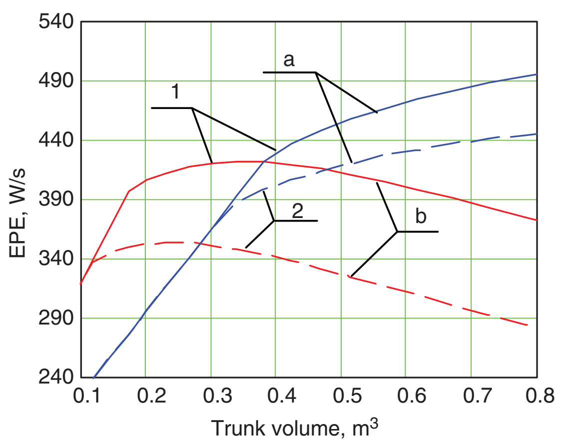

Fig. 1. The effectiveness of various styles of limbing in different operational conditions: a – with the work mix; b – without the work mix; 1 – power of harvester head drive is 50 kW; 2 – power of harvester head drive is 40 kW

0.17 m^3, the effectiveness of harvester exploitation falls greatly. The intensive efficiency decline is stipulated be the limits in maximum speed of dragging trees (5 m/s for harvester head Kesla 20 RH (*http://www.kesla.fi*) installed on the described harvester). Using the work mix, we can also observe a sharp decrease in effectiveness, but this time the trunk volume is less than 0.4 m^3. This is the result of velocity limits in the rotation of manipulator Kesla 1395 H (*http://www.kesla.fi*).

EPE function indicates the efficiency of the work mix application while working in the forest stands with tree trunks more than 0.38 m^3. The work in this area is characterised by the maximum acceptable speeds of limbing. In this case, resistance force to tree dragging and inertial force of a tree are significant at full speed. On the contrary, the moment of rotary inertia is small and resistance force to tree dragging with a work mix does not affect the power being used here.

It should be noted that if the trunk volumes are more than 0.75 m^3, the exploitation of such a harvester is awkward because of the limits in the force of dragging rolls of the harvester head (18 kN).

The decline in efficiency of the harvester head drive power affects greatly the effectiveness of its performance in common engineering operations. So while working in the forest stand with the volume of long-tailed timber of 0.30 m^3, the fall in the harvester head drive power from 50 to 40 kW results in the decline in the effectiveness of harvester operation to 13%. In case of working with a work mix, harvester head drive power has no influence on the work in the forest stand with trunk volume up to 0.32 m^3. The limit of effective exploitation of the work mix decreases in this case to the volumes of 0.28$\div$0.30 m^3.

Diverse effects of the harvester head drive power on the effectiveness of various styles of limbing are explained by the dependence of the maximum speed of tree cleaning not only from the power and speed of the harvester head operation but also from rotation characteristics of manipulator drive.

The examined EPE function does not consider random character of the stress appearance in the process of engineering cycle as well as the distribution of operational conditions. Considering such distribution will enable to use EPE criterion more accurately for determination of effective parameters of multioperational machinery for the certain logging enterprises. It should be marked that EPE function is intended for estimation of machine parameters' effectiveness in the given circumstances but does not allow comparing logging efficiency when these parameters are changed.

In order to raise the efficiency of harvester operation in forest stands with trunk volume more than 0.17 m^3, it is possible to increase maximum power of the harvester head and acceptable dragging force of the rolls. But such an approach is interfaced with the overloading of the bearing structure of the harvester and its drive.

Rational modes of machine performance are established on the basis of EPE function as well as acting loading modes. The dynamic loading (Aleksandrov 1983) of specialised wheeled harvesters 4×4 and 6×6 can be examined within the mathematical model given in Fig. 2. The mathematical model has been developed personally by S. Golyakewich.

The model takes into account the torsional stiffness of hinge in semi-frames' joint, the construction peculiarities of manipulator's rotary support platform, the availability of telescopic section in manipulator and the availability of oscillation damper in the rotator design of harvester head.

While modeming the following assumptions are taken: measures of wheels stiffness are described by the appropriate functions from their deformation; stiffness of hydraulic systems of manipulator's support platform pitch, the given stiffness of a jib, handle and telescopic section are constant, damping characteristics of system elements are proportional to the first deformation derivative while element bonds in the model are holonomic. During engineering operations, we consider the contact of wheels and support surface to be localised at a point, while during transport operations, the smoothing exposure on the mover takes place (in accordance with the technique, Yacenko 1978).

On the basis of the dynamic diagram (Fig. 2) the mathematical description has been made in the form of quadric equations system by Lagrange. The solution of the system is made in the suit *MathCAD* 15 according to Runge–Kutta method with the irregular interval of integration.

In accordance with the model shown in Fig. 1, values of internal force factors are determined under dependences:

$$M_{rot.w} = C_{w7} \cdot (-Z_{11} + Z_{12}); \tag{3}$$

$$M_{ben.w} = M_{pm} \cdot \left(g + \ddot{Z}_9 - \ddot{Z}_{10} \cdot (d_2 - d_5)\right) \cdot d_2 \times \cos\varphi + R_{w6} \cdot \left(d_4 \cos\varphi - \frac{b_1}{2} \sin\varphi\right) + R_{w5} \cdot \left(d_4 \cos\varphi - \frac{b_1}{2} \sin\varphi\right), \tag{4}$$

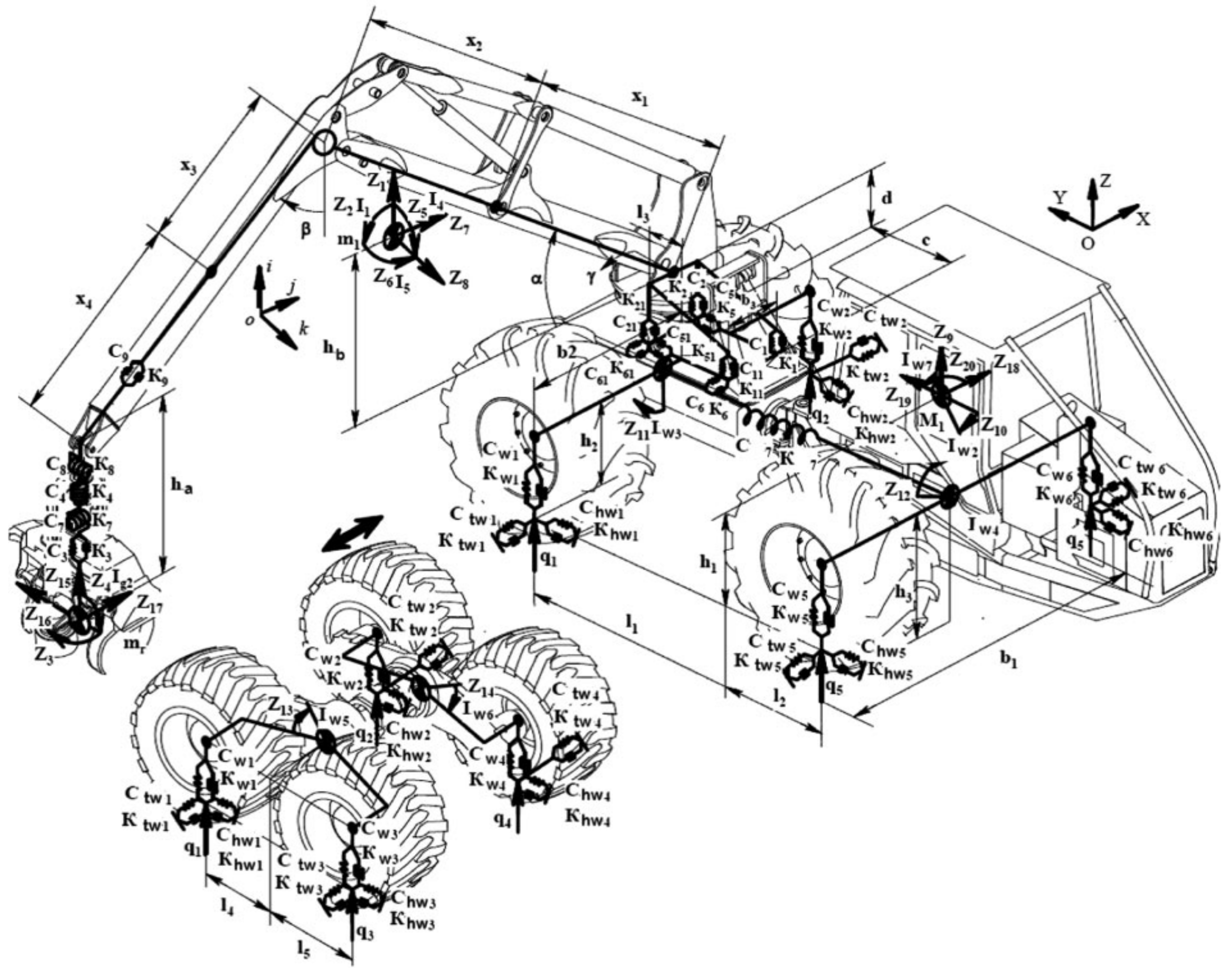

Fig. 2. Diagram of spatial dynamic model of harvester 4×4 and 6×6

$$R_{rh} = -M_{pm} \cdot \left(g + \ddot{Z}_9 - \ddot{Z}_{10}(d_2 - d_5)\right) + R_{w5} + R_{w6}; \quad (5)$$

$$R_{w5} = C_{w5} \cdot \left(Z_9 - Z_{10} \cdot \left(d_4 \cdot \cos\varphi - \frac{M_{pm} \cdot d_2 \cos\varphi - M_{em} \cdot d_1}{M_{pm} + M_{em}} - \frac{b_1}{2} \cdot \sin\varphi \right) + Z_{12} \cdot \left(d_4 \cdot \sin\varphi + \frac{b_1}{2} \cdot \cos\varphi \right) - q_5 \right); \quad (6)$$

$$R_{w6} = C_{w6} \cdot \left(Z_9 - Z_{10} \cdot \left(d_4 \cos\varphi - \frac{M_{pm} \cdot d_2 \cdot \cos\varphi - M_{em} \cdot d_1}{M_{pm} + M_{em}} + \frac{b_1}{2} \cdot \sin\varphi \right) - .Z_{12} \times \left(-d_4 \cdot \sin\varphi + \frac{b_1}{2} \cdot \cos\varphi \right) - q_6, \quad (7)$$

where: R_{w5}, R_{w6} – vertical support reactions under the wheels of logging machine power module; R_{em} – vertical reaction at the hinge; M_{em} – weight of engineering module; M_{pm} – weight of power module; φ – angle of module folding; d_1, d_2 – distance between axis of vertical hinge and gravity centres of engineering and power module with $\varphi = 0°$ correspondently; d_3, d_4 – distance between axis of vertical hinge and axis of engineering and power module with $\varphi = 0°$, correspondently; d_5 – distance between axis of vertical hinge and gravity centre of machine (positive values are corresponded to gravity centre shift towards front axis); $q_1 \div q_6$ – deflections of microprofile surface of movement under correspondent wheels of left and right sides of machine.

The adequacy estimation of the mathematical model developed for the engineering operations was being carried out by comparing measures of the theoretical and experimental values in the hinge of semi-frame joint of the harvester MLH-414 and the vertical accelerations on its harvester head while for transport operations by the comparing their fixed, smoothed spectral densities. The criterion X^2 (Bendat, Piersol 1966) for spectral densities of bending moments in the hinge with the significance level $\alpha = 0.05$ and the value of hypothesis domain $\chi^2_{n,\alpha}$ results in 43.77 is 23.50÷30.2 LS. This means that the compared spectre of experimental and theoretical realisations of bending moments in the hinge are equivalent even alongside with the estimation of other characteristics concerning adequacy of the developed mathematical model of the harvester.

The characteristic feature of the specialised harvesters is in their having the bearing structure of hinge joint of semi-frames (Doronin *et al.* 2005; Titov *et al.* 2008). Its workload during engineering operations depends greatly on the stiffness of the wheels used (Ryskin, Provotorov 1971) and the block system of the horizontal hinge (Sadovskij *et al.* 1972; Kochnev *et al.* 1990). Equation (8) describes the experimentally established stiffness of the wheels 30.5÷32.0 LS used in the harvester being under examination in the working range of loads (Pirson validation criterion of approximation is 0.966):

$$C_{wi} = 2 \cdot 10^6 \cdot \Delta w_i + 98150, \quad (8)$$

where: w_i – deformation of *i*-th tire [m]; $2{\cdot}10^6$, 98150 – empiric coefficients having dimension [N/m^2] and [N/m], respectively.

The performance of the manipulator aside the longitudinal axis of the engineering module is accompanied with the uneven deformation of its wheels, the bias of the module towards the manipulator's work and the increase of torsion moment in the hinge of semi-frames joint. Moreover, according to Fig. 3, it should be taken into consideration the cases of hinge loads before and after separation of the engineering module wheel which is the farthest from the harvester head. In the first case, the increase of the vertical load on the manipulator results in the rise of the engineering module bias. This process is accompanied with the increase of the torsion moment in the hinge.

The value of torsional moment depends greatly on the stiffness of the block system. So if the vertical load on the manipulator is 5.88 kN, then the value of torsional moment changes from 18.1 to 25.7 kN·m, when the stiffness of the hinge system block is $5.0{\cdot}10^5 \div 2.0{\cdot}10^6$ N·m/rad. If the other conditions are equal, the reduction of the hinge block system from $1{\cdot}10^6$ N·m/rad to $2{\cdot}10^6$ N·m/rad decreases its loading on the torsion by 8.2 kN·m, while the separation of engineering module wheel of harvesters comes at overturning moment which is less by 22.8 kN·m. After the separation of engineering module wheel, the stiffness of the hinge system block influences the pivot angel of the module when the required restoring moment is created.

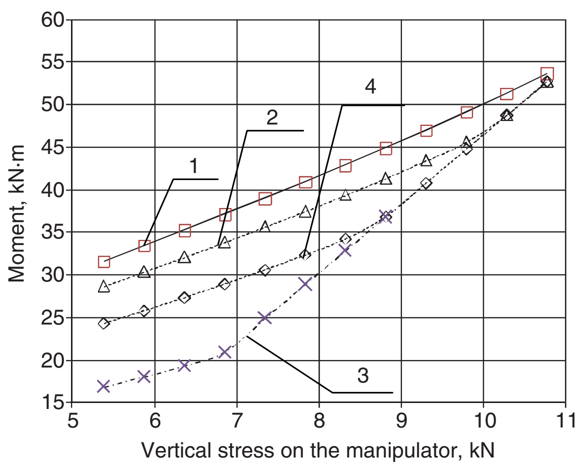

Fig. 3. Changes of hinge workload by means of the torsional moment under different stiffness of the block system: 1 – $C = 2.0{\cdot}10^6$ N·m/rad; 2 – $C = 1.5{\cdot}10^6$ N·m/rad; 3 – $C = 5.0{\cdot}10^5$ N·m/rad; 4 – $C = 1.0{\cdot}10^6$ N·m/rad

On the bases of the mathematical model created and according to the dependences given in Fig. 3, it is possible to establish rational values for the stiffness block system of semi-frames in the harvester, providing minimum values of the torque in the hinge within the required level of stability of harvester engineering semi-frame. Besides, the creation of the block systems with large stiffness (more than $2.0 \cdot 10^6$ Nm/rad for the harvesters of the given type) and the possibility of automatic control of the moment of stability in operation are considered to be an important approach in the development of hinge joint bearing construction. This will make possible the obtaining of the required stability of the harvester automatically with the minimum of loading on bearing construction elements.

In order to increase the stability of the harvester while working in the extended protrusion, operators often make semi-frames folding towards the manipulator's work. In this case, the loading of the hinge joint by the torsion and bending moments varies as it is shown in Fig. 4.

The bearing construction suffers from the vast dynamics loads while the harvester is moving through the cutting area and single asperities like stumps, piles of cutting residues and fallen trees. The modelling allowed determining the loads acting on the hinge in the semi-frame joints moving through single asperity (Fig. 5).

The motion of harvesters is accomplished in unlocked horizontal hinge. In this case, the torsion moment is small enough and occurs only with some resistance of hydraulic liquid circulation in the block system.

It should be noted that overcoming of barriers with the speeds higher than 1.94 m/s change of value of bending moment and in vertical reaction in the hinge of semi-frames is not more than 3.5%.

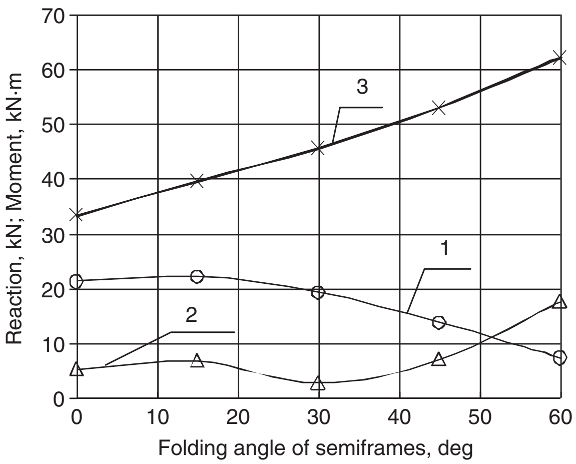

Fig. 4. Changes in the loading of the semi-frames' hinge joint in the various position on the horizontal plane and boom of the manipulator of 9.3 m: 1 – vertical reaction; 2 – bending moment; 3 – torsional moment

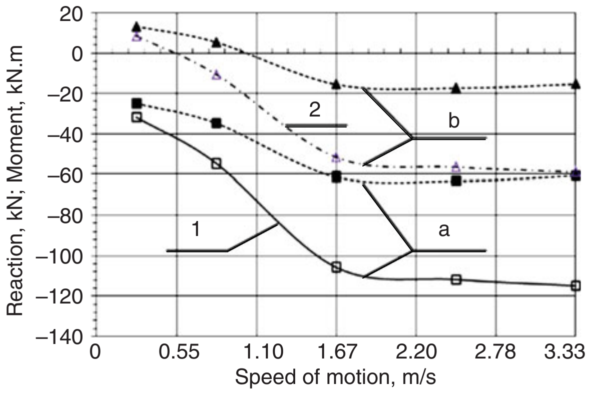

Fig. 5. Changes in the loading of the semi-frame's hinge joint while moving through roughness of 0.25 m: a – a wheel of energetic module; b – a wheel of engineering module; 1 – bending moment; 2 – torsion moment

Moreover, the model suggested and the results obtained on its basis are used while creating the bearing constructions of harvesters, making the choice of the rational arrangment of parameters and the parameters of hinge block system and explanting of the effective ways of engineering operations, depending on natural and industrial operating conditions.

Results

The examined criterion of energy potential efficiency enables to make the complex estimation of harvesters' drive and their structure parameters in various natural and operational conditions.

Its peculiarity is the combined regard for energetic and time constituents of harvester operation efficiency and the possibility of introduction of engineering and structural limiting factors.

The researches proved that exploiting the harvester MLH-414 for limbing in pine forest stands with the estimated productivity of the second class, and it is very efficient for the manipulator to use the oncoming feeding of harvester head by the tree with a trunk volume more than 0.38 m^3. With lower trunk volumes the common style of limbing is more reasonable.

The decline in harvester head power from 50 to 40 kW leads (in the given case) to the decrease in harvester efficiency by 13%. While working with a work mix such a decline has an impact only on operation effectiveness in forest stands with trunk volume more than 0.32 m^3. The performance of harvester MLH-414 without a work mix in the forest stands with long-tailed tree volume more than 0.78 m^3 is awkward because of the limits in the dragging force of rolls.

The experimental researches carried out proved the hinge joint of semi-frames to be overloaded. The examination of the workload of the harvester MLH-414 while performing the prior tension of the tree proved that the workload in the hinge joint of semi-frames are dependant greatly on the parameters of driver rigidity and operator's skills. The model suggested and the results obtained on its basis are used while creating the bearing constructions of harvesters, making the choice of the rational arrangment of the parameters and the parameters of hinge block system and explanting of effective ways of engineering operations, depending on natural and industrial operating conditions.

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