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Calculus of movable element durability for tyred self-propelled machines in construction

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Abstract
This article describes the calculus of movable element durability for tyred-wheel loaders and dumpers. It begins by connecting a law of distribution the given system for calculating the feasibility parameters, then it shows the calculus of the fatique life of the movable element in the tyres self-propelled machine used in construction; this is considered to be an the line vibration system, their operating functions being represented by the road microprofile (ergodic stationary process). The dynamic processes are linked with stressed done by them inside the structure during operation. The spectral theory of vibrations (being aware the static working character of the mechanism) allows the total finding out the distribution places of the dynamic stresses. The calculus method of the fatique life for the machine parts rely on the theory of cumulation phenomena of the fatique faults. By this justification, the whole process is determined; known for the process of "narrow band" type. For the new generation machines, there is overviewed the technological performances as well as the designing conditions of its components, that is axes jointed metallic structure, joint mechanism of the equipment, and the technical solutions used to improve the productivity, durability and the handlingness of the machine.

1. Introduction

This paper is aimed at calculating the durability of certain components, such as the reciprocating lever meant to ensure the vehicle safety during work. The handling mechanism loading mass is appreciated for the power steps of the installation, and it is seen as being included in the vehicle mass [5].

Within the category of the joint-type dump transport self-propelled machine for constructions and roads, there are performance equipments, such as loaders for material handling or self-propelled dumpers for construction sites that can transport the material by shovels or forks.

For these equipments there are given the testing conditions as well as the feasibility and durability test of the subassemblies and structures that are built in their construction.

2. Models for the estimate of dynamic loads being produced in construction vehicle elements, and the association of a repartition law for the calculation of durability parameters [1, 2, 3]

The vehicle power indicators take into consideration parameters such as: the power per size and weight, the consumption, efficiency and interval of the aggregate work over the working cycle.

During a working cycle, the tyre-wheeled loaders(Figure 1) perform the following operations: bucket loading, materials transport, bucket unloading into the tub, and empty bucket movement back to the pile. Articulated dampers for material transport to the site (Figure 2) use, as a rule, verified assemblies, such as high performance Diesel engines, motorized axes, hydrodynamic drives, operating systems and brakes (engine brake, retardation transmission), cabin suspension devices, or operator loading seats. The study of the vibrations specific for a transport aggregate on this cycle is made with the help of simplified dynamic models, having a certain degree of freedom, whose choice depends on the purpose of the calculation and on the number of transports permitted by the existing joints between various subassemblies.

For the articulated tyre-wheeled loader DK-2.8D, similar with the Cat 950H (or 966H) model the experimentally determined wheel load mass is of 4,600 kg for a transport speed of 7.5 km/h. In this case the loading factor βe_f, was analyzed during the working process, with a load of 30-40% per transport cycle, that is, for a load carrying transport followed by a transport with an empty tub

Fig.1 [6]

Fig.2 [8]

The association between a repartition rule and the real life system is made with the help of a reasoning which combines the physical interpretation and the experimental checking, the most important argument being given by the comparison with experimental results.

System phases are influenced by determining elements (human decisions), as well as by aleatory experiments (unexpected deviations from nominal parameters).

Taking all this into account, the behaviour of the given system during operation may be analyzed through statistical and probability methods.

The times when defects appear, as well as their interval are random measures, depending on the physical and chemical qualities of the material, on the manufacturing qualities, the prophylactics of the system as an assembly or of one of its elements, and may follow various distribution laws:

- *Normal Law*: characteristic for ageing elements, fatigue and excessive usage;
- *Reyleigh Law*: used in the durability study of dredger and loader digging cup dentures, especially when made of hard alloys. It is also called the eccentricity law.
- *Exponential Law*: models all cases of unexpected defects with an aleatory character. It is characteristic for the useful life time of an element.
- *Gamma Law*: used when the number of defects r is presumed, and the time lapse until the precise number of defect appears is examined.

- *Waibull Law*: associated with large classes of phenomena, from metallic material breaking, to durability and environment pollution.
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In the case of digging and transporting devices the probability density of transport distances S repartition is described by Waibull's Law [5]:

$$p_S = \alpha \cdot \beta \cdot S^{\alpha - 1} \cdot e^{-\beta \cdot S^{\alpha}} \tag{2.1}$$

where: - for bulldozers $\alpha = 2.1$; $\beta = 0.000408$ are the parameters of Waibull's repartition law ; the earth transport distance is S = 30 – 50 m but it can also reach 100 m. For tyre-wheeled loaders this distance is generally much smaller, but the vehicle can also be used for the load transport over larger distances.

- for scrapers: $\alpha = 1.35$; $\beta = 0.0004$; the current earth transport distance S = 50 – 500 m. The other cycle distances are: earth gathering 25-40 m and unloading 15-25 m.

3. Calculus of the fatigue interval in the moving elements of tyre-wheeled self-propelled vehicles for constructions.

Self-propelled tyre-wheeled vehicles with moving buckets for constructions (STVMB) may be considered as systems with in-line vibration, and the working functions represented by the micro-profile of the site road after stabilization – as stationary and ergodic processes. Therefore, the dynamic processes and the stresses determined by them during the work of carrying devices are completely random; STVMB working conditions in a straight line are also stationary (or quasi-stationary) and ergodic. These characteristics were proved experimentally by the study of certain working elements of the STVMB [4, 5].

For stationary and quasi-stationary processes calculation methods were devised for the fatigue interval of vehicle parts, based on the theory of cumulative fatigue defects [1, 4].

In order to understand these calculations, it is important to know the entire distribution of potential amplitude values of the dynamic stresses.

The spectrum vibrations theory, built upon the above conclusions, based on the known statistical character of working functions, allows finding, for the aggregate elements, the entire distribution of potential amplitudes specific for dynamic stresses. Therefore, in order to tie them all in one, the calculus for vibrations and fatigue interval of movement elements, in the case of STVMB under random action conditions, the principles governing the transfer from an ordered density of vibration distribution to the distribution density of stress amplitudes must be defined [4].

The same applies when the whole process must be determined, but it is only known for the "narrow band" processes, although it is fundamental in practice for the design of STVMB.

Generally speaking, the spectrum of dynamic stress densities acting at the level of STVMB wheels, but also the stresses generated by these loads have a number of peaks. This means that, during the process, heavy cycles take place and their effective frequency must be determined according to the number of extremes.

$$\omega_{ef}^{ext} = \sqrt{\frac{\int_0^\infty \omega^4 S_{sd(\omega)} d\omega}{\int_0^\infty \omega^2 S_{sd}(\omega) d\omega}} \tag{3.1}$$

where:

ω - the process frequency, 1/s ;

$S_{sd}(\omega)$ – the densities spectrum of ordered dynamic stresses (daN^2/cm^4);

It is known that, if simple cycles take place in the process, then the effective frequency is determined from the number of times it passes through the zero level (after the number of zeroes):

$$\omega_{ef}^{zero} = \sqrt{\frac{\int_0^\infty \omega^2 S_{sd}(\omega) d\omega}{\int_0^\infty S_{sd}(\omega) d\omega}} = \frac{\sqrt{2}}{\sigma_{sd}} \sqrt{\int_0^\infty \omega^2 S_{sd}(\omega) d\omega} \tag{3.2}$$

where:

σ_{sd} – the square average of ordered dynamic stresses, in daN/cm^2.

The spectrum density of such a process has only one peak. The lower the process spectrum, the more important $\omega_{ef}^{\max}$ frequency is at ω_{ef}^{zero} frequency. At the limit, when the loading (stress) process is made only of simple cycles $\omega_{ef}^{\max} = \omega_{ef}^{zero}$.

Therefore, we have the relation:

$$\beta_{ef} = \frac{\omega_{ef}^{max}}{\omega_{ef}^{zero}} \tag{3.3}$$

It may be seen as a coefficient of the process for the “narrow band” characteristic.

If β_{ef} is closer to 1, the “narrow band” characteristic of the process will be more appropriate [4].

The level of the “narrow band” process is designated for a series of bucket-equipped vehicles, as shown in Table 1.

Table 1 [4]

MATBC	**Load weight x 1000 kg**	**Working speed, in km/h**
4 BC –10	10,0	5,2
3 BC-15	11,0	5,1
DK-2,8D	4,6	7,5
950H Cat.	4,6	7,0

The level of the « narrow band » process was experimentally verified through weights which affect the vehicle wheels, as shown in table 1.

The analysis of the coefficient derived from the oscillation recording device, with the loading process characteristics (see Table 2) showed that, during the process, the relative number of heavy cycles (two, three etc.) does not exceed 30-40%.

This opens the way for an engineering study of the closeness between loading (stress) processes and the "narrow band" characteristic, for which the distribution density on the ordinate $p(S_d)$ follows a normal law, while the density distribution of amplitudes $p(Sd_a)$ is made according to Reyleigh law. Then by

$$p(S_d)=\frac{1}{\sqrt{2\pi}\cdot\sigma_{sd}}\cdot e^{-\frac{S_{d^2}}{2\sigma_{sd}^2}} \tag{3.4}$$

we have
$$P(S_{da})=\frac{S_{da}}{\sigma_{S_{da}}^2}\cdot e^{-\frac{S_d^2}{2\sigma_{S_{d_a}}^2}} \tag{3.5}$$

where:

σ_{sd} – the square average of dynamic amplitude stresses, daN/cm^2 ;
S_d, S_{da}-the ordinates and amplitudes of dynamic loads, daN/cm^2.

The loading factor, the square average of the load and the amplitude of the wheel load, derived from the calculus and from experiments in [4], are presented in Table 2.

Table 2 [4]. Loading factor, square average of the load and amplitude of the wheel load

Vehicle type	Wheels	Narrow band coeffi-cient	Square average of loads on ordinate 1000daN	Load amplitude x1000 daN			
				Average	Square average	Average	Square average
				Experiment	Calculated	Experiment	Calculated
DK-2,8D (950H)	front bridge	1,34	3,94	2,87	4,94	2,71	2,58
	back bridge e	1,31	2,76	1,97	3,46	1,85	1,81
4BC-10	front	1,25	1,2	0,78	1,50	0,74	0,78
	back	1,21	1,4	0,83	1,75	0,86	0,92

In this case, the average m_a, and the square average of the amplitude may be found from the square average of ordinate σ_{sd} of the dynamic stresses.

$$m_a = 1,253\sigma_{sd}, \sigma_{sda} = 0,655\sigma_{sd} \quad . \tag{3.6}$$

In order to check that such a transfer takes place (when we consider the transformation undergone by the machine during its operation over the working cycle), the statistical characteristics of empirical and theoretical laws were decisive for the distribution of ordinate densities and amplitudes which influence the vehicle wheels, working together (for the calculus of important coincidences of the square average values of load amplitudes). Thus it was easy to determine the possibility of passing from the distribution of the ordinates, based on the spectrum oscillation theory, to the distribution of stress amplitudes, necessary for determining the fatigue interval.

For a simpler calculation, in this case, the study of the loading – unloading procedures was replaced by the study of several equivalent transport procedures, which generate the same dynamic loads (or unloads). The total amount of mechanical-mathematical operations allowing an exchange of loading-unloading procedures with transport ones, as dynamic action positions [4] are known under the name of "vehicle transformation". Here, the vehicle transformation has an energetic character and is followed by the shift of input functions. The approximation for bucket loading vehicles is based on the fact that these input functions for loading-unloading procedures take the form shown in [4]:

$$F_1(t) = h_1 - \frac{1}{c_R}\left\{\frac{1}{2BL}\sum_{j=1}^{n} P_{z_j}\cdot\left[-x_j L + 2y_j B + (1-\beta_s)BL\right] + \frac{1}{2B}\sum_{j=1}^{n} P_{x_j}\left(Z_i + R_R\right) - \right\};$$

$$F_2(t) = h_2 - \tfrac{1}{c_R}\left\{\frac{1}{2BL}\sum_{j=1}^{n} P_{z_j}\left[-x_i L - 2y_j B + (1-\beta_s)BL\right] + \frac{1}{2B}\sum_{j=1}^{n} P_{x_j}\left(z_j + R_R\right)\right\};$$

$$F_3(t) = h_3 - \frac{1}{c_R}\left\{\frac{1}{2B}\left[\sum_{j=1}^{n} P_{z_j}\left(x_i + \beta_s B\right) - \sum_{j=1}^{n} P_{x_j}\left(z_j + R_R\right)\right]\right\}$$

$$F_4(t) = h_4 - \frac{1}{c_R}\left\{\frac{1}{2B}\left[\sum_{j=1}^{n} P_{z_j}\left(x_i + \beta_s \cdot B\right) - \sum_{j=1}^{n} P_{x_1}\left(z_j + R_R\right)\right]\right\}$$

(3.7)

where: $F_j(t)$- the drive function for the wheel « j » of the vehicle, in cm;

h_j- the ordinate of the micro-profile for the wheel « j » of the vehicle, in cm;

c_R- the complex road hardness, daN /cm;

B,L- the track width and vehicle wheel base, in cm;

Px_j, Pz_j- the reactions taking place in the bolts fixing the working element (bucket) to the vehicle frame, in daN;

β_s – the relative safety coefficient for the position of the weight centre of the vehicle;

R_R- the distance given by the wheel angle radian, in cm

Note that the terms of the transport regime, increasing for the hole form of the road profile, tend to zero, therefore Fj(t) =hj. From the above mentioned dependencies it results that during the loading-unloading procedures, the dynamic processes, as well as the transport ones may be equivalent to working regimes in a “narrow band”. In this case the statements made are true, and along with the above mentioned laws, the laws of transfer from the ordinates distribution to the distribution of load (stresses) amplitudes are also applicable.

If the distribution density of amplitudes, as well as the spectrum density of the dynamic stress ordinates is known, the method given in [4] can also be applied for the calculus of the fatigue interval of working parts of the STVMB.

Generally speaking, the working cycle of the STVMB is made of 4 main procedures: loading, loaded vehicle transport, unloading and vehicle transport without a load. Fatigue effects add up during all these four procedures. The calculation method shown above permits the calculus of the fatigue interval, presuming that the working regime does not stop, as if a single movement would take place.

For the vehicle work this fatigue interval is selectively called T_c.

Obviously, each vehicle element is bound to have 4 selectively chosen fatigue intervals, characteristic for the 4 procedures of the working cycle. The element under focus is entirely characterized by a single fatigue interval, which shall be called the general interval T_g, and which is determined for the real cycle made of the four procedures. Knowing the selectively chosen fatigue intervals Tcj, as well as the continuity of the cycle operation a_{0pj}, presuming that the intensity of adding up fatigue defects is linear, proportional to the time of the selective interval, we get the equation for the general calculus of the fatigue interval :

$$T_g = \left(\sum_{j=1}^{n} \frac{a_{op\,j}}{T_{cj}} \right)^{-1} \qquad (3.8)$$

Under real operation conditions the fatigue interval is dispersed, because its result must be thus chosen as to be smaller than the average resulting from the calculus.

The general fatigue interval, calculated under the condition of independent values distribution, shall be called a Gamma-percent Tgγ, and shall be read in the formula:

$$T_g \gamma = T_g \cdot S_\gamma(\nu) \qquad (3.9)$$

The distribution function of $S_\gamma(\upsilon)$ depends on the type of distribution specific for the fatigue interval, and the size of the variation coefficient γ, for which the average of the fatigue interval is true.

The value of $S_\gamma(\upsilon)$, if υ şi γ are known, is determined according to data from specialty literature [1, 2, 4]. It is determined through experiments on similar vehicles, and if there are no such data, then it is recommended that γ = 80%, υ = 0.4. In this case $S_{\upsilon}(\upsilon)$ = 0.65.

Thus, according to the above shown method, the fatigue interval was calculated for the swing lever 4BC-10 of the self-propelled vehicle [4]. During its movement on site roads the following data were obtained:

V, in km/h	7,5	13,5	17,5
Tg, in h	2360	52,3	4,17
T_{υ} (with $S_{\gamma}(\upsilon)$= 0,66), in h	1557	38,0	0,75

The data obtained are checked with the results recorded during the operation of transport vehicles of the same type.
Thus, the possibility of calculating the fatigue interval of STVMB elements was proved.

4. Structure of the joint chassis machines[6,7]

The condition of the durability and trials calculus are under part I.

The axes are such designed as to ensure the durability necessary for all operating conditions. The axis of the rigid side is mounted on the frame and supports the wheeled-loader mass and the transmitted twisting moment, as well as the value of the soil reaction against the wheels and traction forces-braking for loading/unloading the transport operation (with or without stress).

The back axes is design to oscillate in cross plan, +- 13 degrees. The 4 wheels keep on the contact with the soil during the run ensuring a corresponding traction and a very good stability of the car.

The integrated breaking system (IBS) diminishes the oil temperature and ensures a smooth run of the gearing. IBS has a direct impact on the axes durability, mainly on the breaks when run on long distance and/or strong breakings are used.

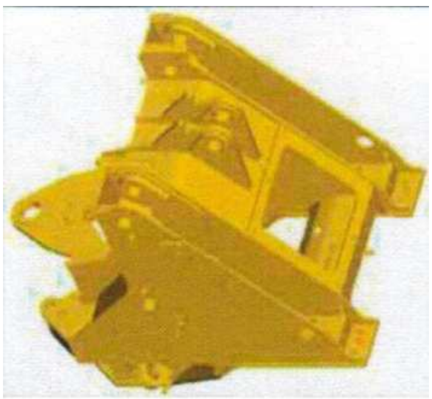

Fig.3 [7]

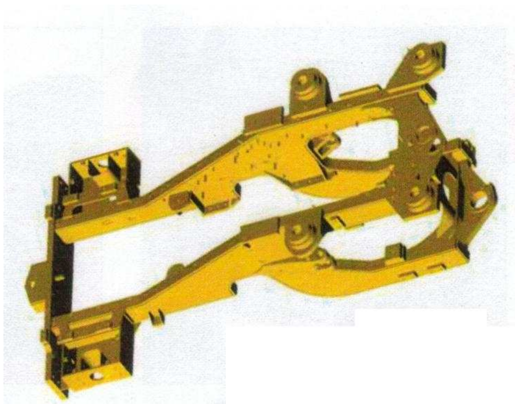

Fig.4 [7]

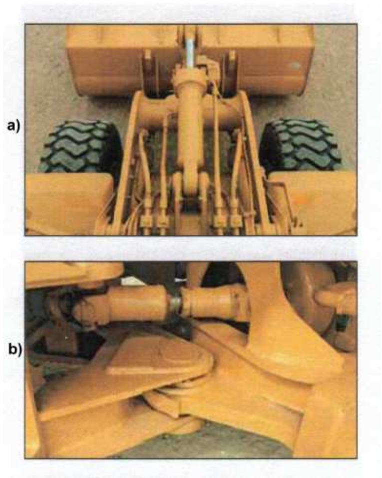

Fig.5, a sib [7]

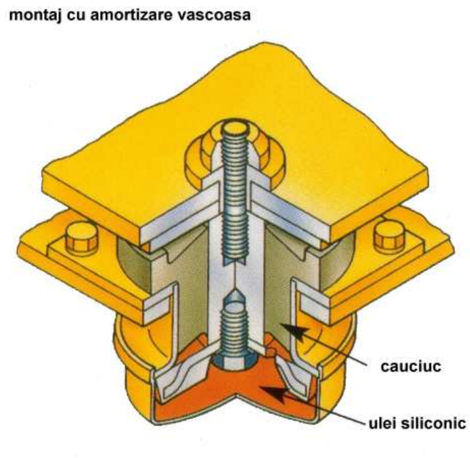

Fig.6 [7]

Structure. The design of jointed frame aimed to obtain enhances durability of the gear box section (Fig 4) and the rigidity of the 4 boards that make the loader tower (Fig 3). The construction is welded by the robots. The robotized welding joint the elements by a deep penetration into the plates making an excellent fusion of the adding material with the base one, such obtaining a maximum resistance and durability.

Back frame. The structure of the gear box is endowed at the front and back ends with plates that ensure a strong resistance and rigidity towards twist, mainly towards the impact during working (see fig 4).

There are mounted on the platform the gear, transmission axes, protection system of the ROPS cabine, and other accessories.

Chassis joint *(Fig 5b).* The distance between the stands and the bottom plates of the linking joint is important for the car performances and components life. This link is designed to ensure an excellent distribution of the load and a longer life to the bearing. (Fig 5b).

Front axis. The end of the front frame ensures a solid mounting base for fore main elements of the car such as front axis, lift arm, arm lift cylinders, shovel sleeping/trimming cylinder. The 4 plates forming the tower, absorb the forces generating the shovel movement and the structure twisting plus the stressed transmitted by the traction forces of the wheels when the shovel penetrates the material.

Joint mechanism of the loading equipment (Fig 5a) is a simple type, designed with Z bars. It generates a big ultimate strenght of material at a good angle dip of the shovel or its coming back.

ROPS protective structure and the cabin

Both ROPS protection and the cabin have a standardized structure according to ISO 3471/1994 and SAEJ 1040C ROPS (Roll Over Protective Structure) as well as ISO 3449/1992 FOPS (Falling Object Protective Structure). To enhance the comfort by diminishing the vibrations, the cabine is mounted on rubber viscous vibration isolators with silicone oil damping (figure 6). The noise level of the cabin is standardized by the standard 2000/14/EC [6,7]

Part 2

Within the category of the joint-type dump transport self-propelled machine for constructions and roads, there are performance equipments, such as loaders for material handling or self-propelled dumpers for construction sites that can transport the material by shovels or forks.

For these equipments there are given the testing conditions as well as the feasibility and durability test of the subassemblies and structures that are built in their construction.

5. Motor deck loading

There are two types of problems in loaders; these problems may influence the construction of the motor decks [5]:

- Phenomena and loadings resulted from the car moving as well as the phenomena that accompany the shovel filling. The loaders overcome the following resistances when working:
- Road resistance as both the terrain and the wheel loose shape
- Force – a component of gravity (on slopes)
- Pressure or traction strengthes
- Depending on the work conditions, some resistances may have an ultimate significance, others having no significant importance. Always a certain sum of those ones decides upon the total resistance and engine power. The elements of the acting system for moving such car also suffer, beside the above mentioned resistances, dynamic and variable trials with different amplitudes and frequences, depending on the dynamic parameters and system rigidity. These trials accompany almost all the operations done by the car; starting-up, passing the obstacles at a certain speed, sliding over a soft soil as well as the working process itself. It is shown the results of some trials made at ht e loaders L2 of 3 cm indicated under [5].

The load of the acting system was measured using the tensometer method on the axes that link the main transmission with the wheel reductor. On these axes there were measured the total moments and their distribution on those wheels and were established the moments resulting from the internal tensions that may appear sometimes when acting a 4x4 type.

Figure 7 shows 3 measures registered at 4x4 moving system, during shovel filling on dry earth, asphalt and concrete, at maximum coefficients of adherence. In figure 7,a the loadings of front and back wheels are almost equal; in figure 7,b, the loadings of the front wheels are much bigger; in figure 7,c the back wheels have bigger loadings. These records show that the moment peaks are provoked by the dynamic action (due either to the excavated material non-uniformity, or to non-uniform turns of the engine, etc). These values sometimes exceed the adhesion capacity and so the wheels ………..

The maximum moments on the front wheels appear on the total unload of the back wheels; it may be done when the shovel blade is grasped Mf max=3Momax. The maximum of the back wheels will take place when the front wheels are totally unloaded. M max = 1.5 Mo max, where Mo max corresponds to the equal distribution of the loadings on desks.

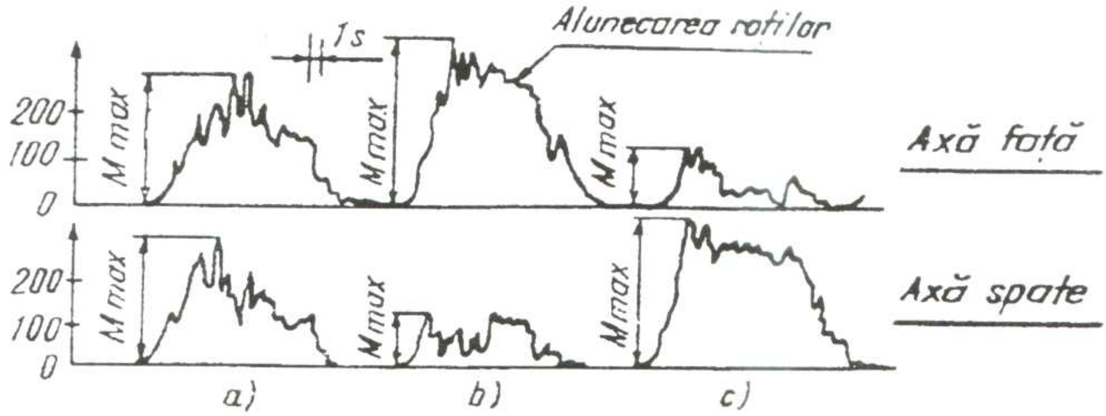

Fig.7 [5]

6. Working potential characteristic and the energy balance of the car

During operation, the dynamic behaviour of the equipment relies on the foundation of the performance criteria operating in real time on working technological cycle. The energy balance of the car (Fig 8) is analyzed for:

- Tractor + equipment that works in a stationary regime with THM or TM to load the shovel
- Tractor + equipment in dynamic regime with THM or TM and loading (tilting) hydraulic mechanism.

The skidding given depends on the loading time:

For 20% time for loading shovel, the skidding varies between 12% and 60%;

For a 20-30% loading time, the skidding reaches 60%

For the technological regime on skidding phase it takes 10-30%. The characteristing of engine functioning in a dynamic regime correspond to the powers consumed by the tractor + equipment: a skidding Pp = f(Fer); to a overcoming of the global resistance Pf =(Fcr,Vt) and the loading hydraulic mechanism Pmh = f(Pe).

Figure 8 shows the potential working characteristic (Per) and energy balance of the standard tractor (P=60kw) on which it may be mounted the loading equipment [5]: a) for stationary loading; b) for dynamic regime; I,II tractor with THM and TM; Pmh – power at the hydraulic mechanism; THM, TM – power at HT (hydrotransformer) and mechanical power.

Other symbols: Pm engine power (at the pump) Pcr = critical power; Pp = power at skidding; p = skidding coefficient; Pf = global resistance power at forwarding; ; Ff = global resistance

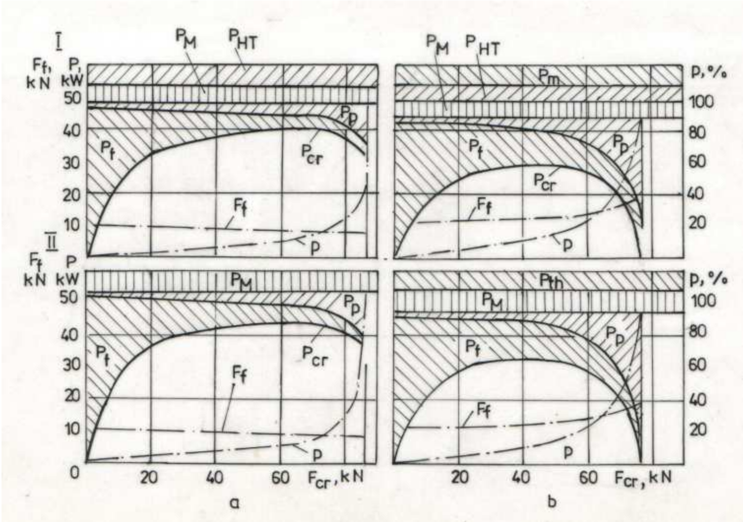

Fig. 8 [5]

7. Testing the main subassemblies of hanlage machine on bench in lab conditions [5.6]

If feasibility and durability trials are made, the regime is chased on the basis of the spectra of stress recorded during functioning or on the tracks of some testing fields for the operating conditions specific to the given type of the equipment. The hanlage characteristics are provided by the industrial tractor that is on the basis of the machine construction. The standard industrial tractor to performe the haulage tasting works together with various equipments such as loader, dumper, pis tio or scraper. Figure 8 shows the working potential characteristic and energy balance of the standard tractor with a power of 60kw.

To obtain the experimental haulage characteristics of the industrial tractors one must have into account the followings:

- For the industrial tractor it is characteristic the skidding area increased from 7% to 100%, but in this area it is quite different to obtain a stable loading;
- When tasting the industrial tractors one cannot uplift the characteristics on all steps as when the time is modified the terrain conditions alter; so the experimentators' performing intervals are changed. The hauling testing are according to Gost 23734-74 and General rules for testing methods STAS 9624/1-1990. The methodology of accelerated hauling testing of the industrial tractors is based on the using of engine bench testing characteristics STAS 665-1987 (mechanical transmission – TM) or on thermal engine-hydraulic transformer aggregate (hydromechanical transmission – THM), on the experimental curve of skidding to determine the check points and further calculus of the hauling characteristics.

The methodology contains the following steps:

I. Uplifting the characteristics on the bench. For the tractor with TM the turn characteristic of the engine is uplifted; and for THM the output characteristic of hydraulic transformer turbine wheel. The loaders also used TM+THM.

II. On terrain conditions there are determined the qualities of the propulsion adherence and some checking point according to the methods used in haulage testing. The tractor is tested on the track by braking action, by means of traducer element, by using a loading device or another tractor. The transducer indications are set by the recording apparatus sincronously with the turn transducer indication of the engine crank axle, tractor driving wheels and the measuring wheel of the road run. The continous loading by decelerating the braking tractor (□ pis□ tion trials of the vehicles equipped with Diesel engine SR ISO 7644-1998)

III. The haulage characteristic is calculated using the results of the field and bench testing. On the bans of the field trials one determines both the engine and the turbine wheel turns, the theoretical and real speed, the skidding coefficient and haulage force at the hook, then the diagram p (Fcr) is drawn.

$$F_{cr} = F_{crT} - m_g \cdot j \tag{4.1}$$

Where: Fcrt- the force measured with detecting element, in N;
m_g- weight power unit, in kg;
j – accelerate, in m/sp;
Slow up tractor with wheel, m/□ pis:

$$j = \left(V_{ti} - V_{ti-1}\right) / \Delta t \tag{4.2}$$

The founded values Mm or M_T determine the conventional tangential force without taking into account the transmission yield on the car driving wheels with TM or THM:

$$F_{Rt} = \frac{M_m \cdot i}{r_R}; \quad F_{Rt} = \frac{M_T \cdot i}{r_R} \tag{4.3}$$

And motor wheel radius:

$$r_R = \frac{S}{2 \cdot \pi \cdot n_R} \tag{4.4}$$

Where S = the road run by the tractor, in m; n_R = turns number of the tractor driving wheel on the road S.

The obtained results are written in a table, than the relations F_{Rt} (F_{cr} and n_T /n_R) are drawn for all the haulage force interval. Then, the graphics are used to determine Fcr by points used the graphic analytical method.

When breking a tractor one must take into account the inertia forces. The conventional haulage force is:

$$F_{Rt} = \frac{M_m \cdot i}{r_R} + j\left(m_g + \frac{J_T i^2}{r_R^2}\eta_{cr}\right) \quad (4.5)$$

Where Jt = inertia moment of the rotating masses of the engine or of the turbine wheel

The difference between F and Fcr is the sum of the haulage forces to move the tractor Fr and of the frictions Ff in the mechanical side of the transmission.

This difference is called the lost of the haulage force under stress:

$$F_{pts} = F_{Rt} - F_{cr} = F_r + F_f \quad (4.6)$$

Forwardly, the characteristic drawing is done:
- It is given the turns of the crank axle on the turns characteristic of the engine on the bench or on the output engine-hydraulic transformer characteristic. In the given point, using the graphic-analytical method, it is determined for: mechanical transmission TM – engine power Pm, moment Mm, hour fuel consumption Ch and fuel specific consumption Ce. For the hydraulic transmission THM, it is determined the engine turns n, hour fuel consumption Ch and specific fuel consumption Ce
- The conventional haulage force is determined by the relations (4.1 and 4.2). Using the graphic F (Fcr) by graphic-analytical method it is determined Fcr.
It is determined the skidding p corresponding to Fcr in graphic F (Fcr), according to the calculus methodology shown under the point [5].

8. Test of the durability on the bench and their acceleration

The acceleration of the tests may be done by the following methods[6]:

- By intensifying the loading regimes by increasing the numbers of trials in time unit;

- By intesifying the environment influence (temperatures, humidity, etc);

- By using some abrasive media artificially created (when testing the industrial tractors on different testing fields).

To reduce the working time and volume, it may be used the dependence of the wear against the time or journey, using the wear curves of the researched parts.

When using the methods of the accelerated trial it is determined the coefficient of passing from the bench trial to the normal working conditions:

$$K_d = \frac{K}{h} \quad (5.1)$$

Where: h – is the durability of the part or the assembly on the stail determined in hours or after the numbers of the working cycles; K is the durability of this part according to the kilometers run.

When testing the durability on the bench three typer of trial programmes may be used:

a) Constant loading (cyclic loading with constant amplitude or constant twisting moment)
b) Variable stepped loading (cyclic loading with stepped variations of the amplitude or the variation of the twisting moment) in a set order of stress acceleration;
c) Trial with random variation (more often by reproducing the loading regime during working)

While in case of road tests it is usually determined simultaneously the durability of many subassemblies, on the bench tests this is done only for one subassambly or part so being possible to choose the optimum programme for each case bottle at stress level and its application succession.

9. Statistic methods to establish the stress level

From researching the stress regime of the subassemblies on loaders or dumpers it may be obtained data to set the programmes of bench loading and to compune the parts durability. The research data may appear as bar charts, functions of the probability density (for wheel skidding, adherence, transport distance, etc), correlation tables, magnetic records directly used in programming the stressed on the bench[6].

For each car assembly one sets the basic parameters of its stress regime.

To determine the stress regime of the motor desks one need the distribution functions of the twist moment on the main transmission shaft and on the planetary shaft as well as the relations between angular speeds of the planetary wheels, the forces and moments that act upon the motor desk beam.

For the planetary shafts and the motor desk beam one also need the number of stress cycles on km run.

For all the transmission assemblies it is measured the turns of the driving shaft or the car speed. The stress spectrum of the unitary efforts at a planetary shaft of a car is shown in Fig 9 [6]

For the equipment frame and cabin, the unitary efforts are measured in a series of points of the vertical and horizontal accelerations; for the bumpers the distribution of the relation moments of the cylinder and bars are measured.

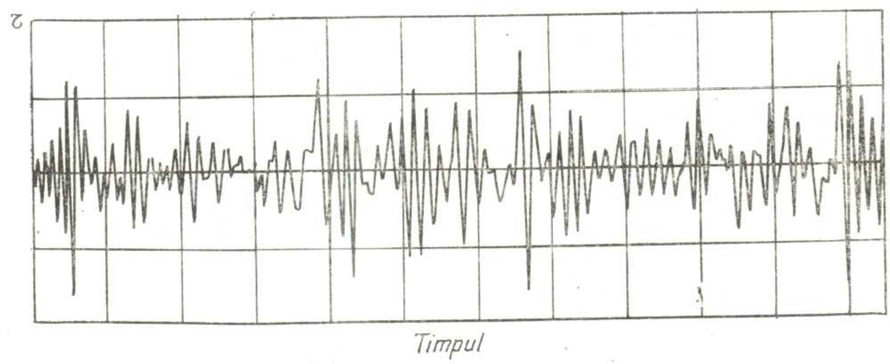

Fig.9 [5]

For the breaking action it is measured the distribution functions of the forces applied on the break clutch (or to the pressure inside the brake cylinders and chambers),

The friction mechanical work from the brake and the temperatures of the brake drums or disks. For braking STAS 11960-1989 is used.

10. Statistic processing of the stress regime recordings.

The stresses that appear inside the elements of the car aggregates when working usually are random time functions. A significant example is the oscillogram of variation in time of the unitary efforts within a car planetary shaft. Fig 9.

This processing of the stress spectrum may be done using two groups of methods [6]:

d) Direct sistematization method of recording the random processes using the set criteria and
e) Methods of the random function theories

In the first case, if there is special classifying apparatus, the methods under group a) allow the rapid gesting of the results (the trials made during the run included) under the accessible form of fatique calculus, adequate to set up the trial programmes determined on the bench.

The methods under the group b) are more complex but mainly offers the possibility of determining the distribution functions of the stress amplitudes, function of the known static properties of the input parameters (for instance the road microprofile) and the dynamic properties of the researched functional system. From the 20 direct sistematization methods some are the very direct object of DIM 45667

11. Methods of elaboration the testing programme.

The programme of testing on the bench the car assemblies is achieved using the histogram or the distribution function of the stress spectrum recorded during the car movement in typical site conditions[6].
The typical working conditions were made for each group of cars, that is industrial tractor working together with other equipments such as: bulldozer, loader, dumper, symbolized here by MATBC.

To constant cyclic stress trials their formation is possible by rising the stress level and its application frequences.

The stress level may be increased up to the moment when the part fault character tied on the bench correspond to the same faults in working regime. To determine the optimum value of the stress it is necessary, by preliminary research to obtain the distribution functions of the working stress $f(\sigma)$ and the fatique curve for the investigated part.

The part durability is working conditions (in working cycles) may be determined by the formula:

$$N_e = \frac{a_k \cdot \sigma_r^m \cdot N_e}{\int_{\sigma_{\min}}^{\sigma_{\max}} f(\sigma)\sigma^m d\sigma} \qquad (8.1)$$

Where:

a_k – correction coefficient of the damage linear summing up (a_k=1)

Ne – basic number of the strain cycles (Ne = $10^6 \ldots 10^7$)

m – slope index

using the fatique curves it is established the unitary efforts corresponding to the cycle number chosen to determine the stress level on the bench.

To force the loading level the stress is increased taken as basic number of the cycles Ns<Ne, Nx being chosen according to the tried part type.

For instance, for the leaf spring Nx = $(1\ldots3)10^5$ is taken for the clutch detaching system Nx = (5….10)10. By the help of the fatique curve the corresponding unitary effort may be obtained:

$$\sigma_x = \sigma_c \sqrt[m]{\frac{N_e}{N_x}} \qquad (8.2)$$

σ_c must be smaller then the flow limit of the respective material. The coefficient of the loading forcing in time:

$$K_f = N_e \cdot Z_o / (N_x Z_e) \qquad (8.3)$$

Where: Ze, Zo – represents the stress cycle load of the part in the time unit in conditions of loading on the bench and respectively the mean obtained during working. The coefficient of passing from the trials on the bench at the normal working conditions, that is by return the durability obtained in working conditions on that one obtained on the bench; the relation (5.1). Using the experimental analytical way, the coefficient of passing from one loading form to another being equal to the ratio of the durability obtained by calculus for working condition and of the durability determined by experiment [6].

The attention is paid to the stepped (block) stress programmes that really reflect the stress regime form working the trying bench. The small stressed that do not influence the parts resistance to the fatique are skipped. Their value do not exceed 10-20% from the maximum obtained values. On working conditions, under their influence it results unitary efforts that do not exceed 50% from the fatique limit of the investigated part. At every step of the programme, the unitary effort it is taken equaly with its value in the centre of each interval. Loading cycle number in each step:

$$Z_i = Z_s \int_{\sigma_i}^{\sigma_{i+1}} f(\sigma) d\sigma \qquad (8.4)$$

Where Zs is the possible mean value of the loading cycle number up to damaging the part on the bench: Zs value may be approximately determined by B.1 relation. It is recomandable that the loading programme be divided in Kc blocks, usually the limit being Kc>10. The loading cycle number at every stress step in a certain block.

$$Z_{ib} = \frac{Z_i}{K_x} \qquad (8.5)$$

The statistic characteristics of the working regime of the programme part used for testing must correspond to the statistical characteristics of the whole working regime (this was demonstrate in the first part of this paper)

12. Conclusions

In some cases it is used the process random modeling elaborated after the known statistic characteristics of the real working regime as for instance: the distribution functia, correlation function, special density, etc. This is done, for instance by the programme acting on the bench by the servo-hydraulic devices, controlled with the electronic equipment (for instance the Hydropuls bench) [6]. The car electronic command-control block provide now the optimizing of its working with a reduced consumption of fuel, working regime adjustment depending on the hydraulic installation, minimum pollution and service warning by GPS for the car maintenance (oil change, air, fuel, oil filters, etc) depending on the impact load degree of the car during the whole period of exploitation on different location on the sites [7,8]. Standards: diagnosis systems SR ISO 4092-95, SR ISO 8093-1998 and Digital information change SR ISO 9141-2000.

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