SOME ASPECTS REGARDING THE DURABILITY OF SELF-PROPELLED TIRE-WHEELED VEHICLE COMPONENTS FOR BUCKET TRANSPORT FOR CONSTRUCTIONS

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Abstract: Self-propelled transport vehicles of articulate bucket type for constructions include loaders or automated dumpers for sites, with buckets for ground transport. This paper presents the calculation of the fatigue interval for the moving element of the self-propelled vehicle with tires, considered as an in line vibration system, and the working functions are represented by the micro-profile of the road (stationary ergodic processes). A relationship is stressed out between dynamic processes and structural stresses determined by these processes during the work. The spectrum vibration theory allows, once the mechanism working character is known, a complete arrangement of dynamic tension distribution places. The calculation method for the fatigue interval, in the case of vehicle parts, is based on the cumulative fatigue defects theory. With these details in mind, the "narrow band" type process is determined.

1. INTRODUCTION

This paper is aimed to calculate 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].

2. MATHEMATICAL 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]

During a working cycle, the tire-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 tire-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 β_{ef}, 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.

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 random experiments (unexpected deviations from nominal parameters).

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

Fig.1 [6]

Fig.2 [8]

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 random 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.

Weibull Law: associated with large classes of phenomena, from metallic material breaking, to durability and environment pollution.

In the case of digging and transporting devices the probability density of transport distances S repartition is described by Weibull'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 Weibull's repartition law ; the earth transport distance is S = 30 – 50 m but it can also reach 100 m. For tire-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 TIRE-WHEELED SELF-PROPELLED VEHICLES FOR CONSTRUCTIONS (STVMB).

Self-propelled tire-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):

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

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.

MATBC	**Load mass x 1000 kg**	**Working speed, in km/h - Table 1 [4]**
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

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

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

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.

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

Vehicle type	Wheels	Narrow band	Square average of loads on	Load amplitude x1000 daN Average	Square average

		coefficient	ordinate 1000 daN	Experiment.	Calculated	Experiment.	Calculated
DK-2,8D	front bridge	1,34	3,94	2,87	4,94	2,71	2,58
(950H)	back bridge	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]:

$$\begin{aligned}
F_1(t) &= h_1 - \frac{1}{c_R}\left\{\frac{1}{2BL}\sum_{j=1}^{n} P_{z_j}\cdot\left[-x_j L + 2y_j B + \left(1-\beta_s\right)BL\right] + \frac{1}{2B}\sum_{j=1}^{n} P_{x_j}\left(Z_i + R_R\right) - \right\};\\
F_2(t) &= h_2 - \tfrac{1}{c_R}\left\{\frac{1}{2BL}\sum_{j=1}^{n} P_{z_j}\left[-x_i L - 2y_j B + \left(1-\beta_s\right)BL\right] + \frac{1}{2B}\sum_{j=1}^{n} P_{x_j}\left(z_j + R_R\right)\right\};\\
F_3(t) &= h_3 - \frac{1}{c_R}\left\{\frac{1}{2B}\left[\sum_{j=1}^{n} P_{z_j}\left(x_i + \beta_s B\right) - \sum_{j=1}^{n} P_{x_j}\left(z_j + R_R\right)\right]\right\}\\
F_4(t) &= h_4 - \frac{1}{c_R}\left\{\frac{1}{2B}\left[\sum_{j=1}^{n} P_{z_j}\left(x_i + \beta_s\cdot B\right) - \sum_{j=1}^{n} P_{x_1}\left(z_j + R_R\right)\right]\right\}
\end{aligned} \tag{3.7}$$

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:

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

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:

$$T_g \gamma = T_g \cdot S_\gamma (\nu) \tag{3.9}$$

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. CALCULUS OF THE TRANSPORT SPEED [8]

The selection of the working stage of the drive according to the resistance to road movement and to the road angle, expressed in a percentage of the vehicle weight, for articulated dumpers (for instance, Terex TA 27) is shown in Figure 3. The calculus of the transport speed is made during ramp climbing or when the slowing down brake is applied. Transport speeds for the articulate damper range manufactured by Terex are shown in Tables 3 [8].

From the intersection of the vertical line (on the right side of Figure 3) representing the vehicle mass with or without a load, with a reclining line representing the resistance to road movement and slope, expressed in %, according to the characteristics of the road where the damper works, a straight horizontal line is drawn towards the left side, to the vehicle drive characteristics, indicating the speed level at which the vehicle may move; then from this point a vertical line is drawn down and the transport speed is calculated in km/hr. With the speed values established in the diagram, and the transport times per cycle (with and without a load) over the road sections, the vehicle productivity and durability are calculated.

Transport speed values (in Km/h), on each speed level, for Terex articulates trucks - Table 3 [8]

Dumper type		**TA25**	**TA27/TA30**	**TA35**		**TA40**	
Speed types				**slow**	**fast**	**slow**	**fast**
Forward (backwards)	**I**	5,9 (5,9)	5,6 (5,6)	5,2 (4,6)	7,9 (7)	5,5 (4,8)	8,4 (7,4)
at each level	**II**	9,1 (13,6)	8,7 (14,2)	11,0	16,7	11,7	17,8
in km/hr	**III**	14,2 (31)	13,6 (32,4)	15,9	24,3	16,9	25,8
	IV	22,1	21,6	24,3	37,1	25,8	39,5
	V	32,4	31,0	31,0	47,7	33,0	50,4
	VI	52,0	51,0	36,2	53,9	37,5	60,0

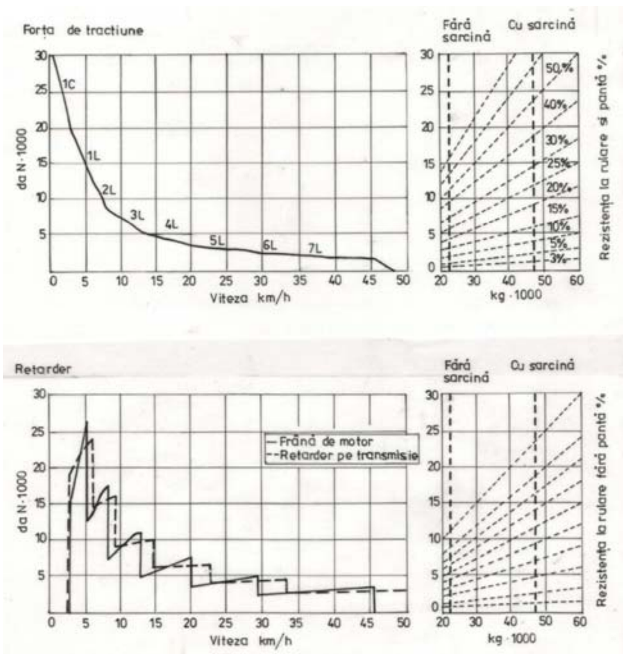

Fig.3 [8]

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[7] **x x x** – *Cat 966H Whell Loader: Cat C11 Diesel Engine with ACERT Technology*, Power 211kW, Bucket Capacity 3,5-4,8 mc, Operating Weight 23800 to 27300 kg, Caterpillar, 2005.

[8] **x x x** - *Articulated Trucks TA25, TA27, TA30 New TA35 new TA40, Terex Building on Technology*, Terex Equipment Limited, Scotland,2006, 24 p.