

THE EFFECT OF DYNAMIC STRESSES DETERMINED BY ROAD DISLEVELMENTS ON THE STRUCTURE OF TIRE WHEELED LOADERS USED IN CONSTRUCTION SITES

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Abstract: The study of vibrations of a self-propelled vehicle is made with the help of simplified dynamic models with a degree of freedom determined by the allowed degree of freedom existing between various subassemblies. This paper is a commentary of the analysis referring to the measurement of vertical accelerations for non-suspended masses (wheels), frame joint, cabin floor or seat, as well as vehicle translation speeds and accelerations for various types of roads, road excitation being an random stochastic process. Dynamic performance and car comfort are determined by the quality of the drive, breaking, and operation systems. Longitudinal accelerations are dynamic characteristics while speed and the impulse (impact force) developed during work determine the tensile strength of the car. For modern loaders, these values are measured by sensors which are limiting the engine drive operation, as well as the functioning of the drive system, thus ensuring an optimal functioning of the entire vehicle.

I. INTRODUCTION

Transport vehicles on wheel with tires are used in constructions for the transfer of various materials to the construction site for unloading in a vehicle bucket, followed by an idle movement. The study of transport vehicle vibrations is made with the help of simplified dynamic models, with a certain degree of freedom; their choice depends on the purpose of the calculation and on the number of transports as allowed by the existing links between various subassemblies of the vehicle [1].

For the operator's comfort, vibrations are analyzed at the level of the cabin floor and driver's seat, both for the suspended load, and for non-suspended loads, that is, for the wheels with a tire loader DK-2.8D, whose characteristics are almost similar to those of model 950H (or 966H), shown in Figure 1 [5]. Figure 2 shows a Komatsu large capacity loader of 8.7 – 9.4 mc, with a total operation weight of 70,800 kg used for quarries and surface mining running [7].



Fig.1 [5]



Fig.2 [7]

The analysis of measurements made in [3] is fundamental in determining the elements of comfort and of the wheel to road adherence, for different road conditions, transport speeds and buckle loads. The purpose of the analysis is to propose solutions meant to improve the elastic and dissipative characteristics of the axle spring, cabin and driver seat. For a tire wheel loader of the DK-2.8D type similar in dimensions with a 950H (or 966H) vehicle, as in Figure 1, the load mass supported by the experimentally determined wheel is of 4,600 kg for a transport speed of 7.5 km/h. An analysis of the loading factor e_f was made during the working process, with a load of 30-40% per transport cycle which included one transport with the load and one without a load.

Vehicle type	Wheels	Load factor, load square average and tire load amplitude - Table 1 [3]					
		Narrow Road coefficient 1000 daN	Load square average on ordinate	Load amplitude x1000 daN			
				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	1,31	2,76	1,97	3,46	1,85	1,81

2. CHARACTERISTIC ELEMENTS OF RANDOM STRESSES [2]

The notion of stochastic process is used as a mathematical – hypothetical – model for describing excitation processes, the response of mechanical systems or the stresses taking place in resistance elements, just as a simple sinusoidal is used for the general modeling of these phenomena.

Thus, if a unique frequency component is predominant, it is useful to apply the model of a simple sinusoidal; if the number of components is too high and their separate identification is not possible or convenient, the most frequently used model is that of the stochastic process, whereby average values are used instead of individual data, the most important being correlation functions and Fourier transforms – spectral power densities. The importance of these average values results from the fact that, for an oscillating system with constant coefficients, it is possible to calculate the appropriate average directly, once the excitation average is known.

If, at the system input, a stochastic excitation process $X(t)$ is applied, the structure behaves as a filter at this input signal, so that the system output results in another random process $Y(t)$ with different statistical characteristics. The purpose is to establish the correlations between the excitation and response signals, when certain characteristics of the mechanical system are known or not. Thus, the fundamental elements, specific for the study of stress structures dynamics may be defined, as follows: *analysis elements*, in fact criteria for checking and determining probability properties of the answer, when the excitation (in a certain form) and system characteristics are known, either in a real time field, or in the imaginary field of pulsation; *synthesis elements*, which actually represent the core for a an optimal design of a strength structure, by determining the characteristics of a certain system if the excitation and response (given or fixed) are previously known. According to their properties, dynamic systems can be divided into three main categories: linear, non-linear and parametric systems. Topics presented herein are specific for linear systems for which the statistical methods are best conceived. These systems have an important advantage, that of allowing the application of the effect overlapping principle. If the statistical parameters of exterior actions are known, then the analysis or synthesis of such systems avoids major difficulties. The dynamic system is stationary when its reaction to any interference does not depend on the moment when the interfering action begins, but only on the periodicity of the considered moments. Linear systems with constant parameters are stationary systems. For such systems two distinct aspects of the movement are usually studied, that is: the stabilized regime and the transitory regime, durability research being especially important in this case.

Model chosen for the study of vehicle vibrations

In the study of vibrations taking place in the case of site vehicles, the first problem to be solved is rendering them through a mechanical equivalent model, made of rigid masses elastically linked with springs and dampers with no mass. The simpler the chosen model, the more useful it is.

Figures 3, 4 and 5, *a* and *b* show the components of the vehicle structure and their linking elements. Figure 3 presents the front tower of the metallic structure to which the bucket handling equipment is attached (see Figure 5, *a*) and the front axis, while Figure 4 shows the back frame onto which the engine and hydraulic power system are attached. The two semi-frames are articulated and operated with the help of two hydraulic cylinders (Figure 5, *b*).



Fig.3 [6]



Fig.4 [6]

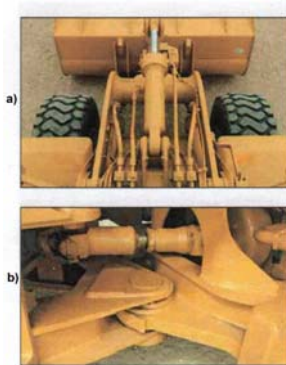


Fig.5, a and b [6]

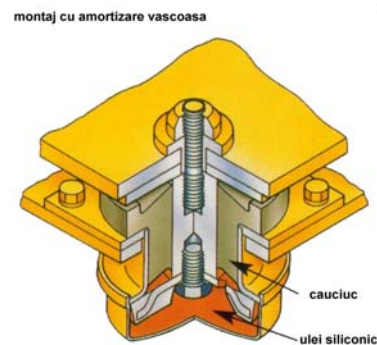


Fig.6[7]

3. EXPERIMENTAL CALCULATION OF THE DYNAMIC STRESS EFFECT OF ROAD DISLEVELMENTS ON THE MECHANICAL ELEMENTS AND VEHICLE DRIVER

Measurements in 7 points were made for the handling of DK-2.8D site and quarry vehicles, similar to the more recent 966H model; acceleration values were thus determined during work, according to [3]. Vertical movements, speeds and accelerations were measured for the four vehicle wheels, marked $Z_1, \dots, Z_4$; at the chassis turning articulation Z_5 , at the level of the driver seat Z_6 and cabin floor Z_7 . Paper [3] mentions the oscillation measurement results of vertical acceleration, for the four vehicle wheels $\ddot{Z}_1, \dots, \ddot{Z}_4$, the acceleration of the coupling device $\ddot{Z}_5$, the acceleration at the level of the driver's seat $\ddot{Z}_6$ and the vertical accelerations of the cabin floor $\ddot{Z}_7$. The discrete acceleration time is of 0.04 sec.

The static characteristics calculated for the recorded measurements are as follows:

$$\text{Average : } \quad x = \frac{\sum_{i=1}^n x_i}{n} \quad (3.1)$$

$$\text{Dispersion} \quad D_x = \frac{\sum_{i=1}^n \left(x_i - \bar{x} \right)^2}{n} \quad (3.2)$$

$$\text{Square average} \quad \sigma_x = \sqrt{D_x} \quad (3.3)$$

$$\text{Variation coefficient} \quad V_x = \frac{\sigma_x}{\bar{x}} \quad (3.4)$$

Correlation function:

$$R(\mu\Delta t) = \frac{1}{n-\mu} \sum_{\lambda=1}^{n-\mu-1} x_{\lambda}(\lambda\Delta t) \cdot \bar{x}(\lambda\Delta t + \mu\Delta t) \quad (3.5)$$

where $\mu = 0, 1, 2, \dots, m$ ($m < n$)

$\mu\Delta t$ - time lapse ;

$\lambda\Delta t$ - order of time discrete moment

$\Delta t(n-1) = t$ - entire time of achievement according to which $R(t)$ is calculated

The normal correlation function:

$$\rho(\mu(\Delta t)) = \frac{R(\mu\Delta t)}{D_x} \quad (3.6)$$

For $\mu\Delta t = 0$, the correlation function is equivalent to the process dispersion.

The spectral density

$$S = \Delta t R(t=0) + 2\Delta t \sum_{\mu=1}^m R(\mu\Delta t) \cos(\omega\mu\Delta t) \quad (3.7)$$

The normalized spectrum density :

$$s = \frac{S}{D_x} \quad (3.8)$$

The histogram for the calculation x of the maximum value $X_{\max}$ on an Δx interval, and of the number of intervals k

$$X = X_{\max} / k, \quad (3.9)$$

The coincidence of value x_i in the chosen interval

$$P_k = \frac{m_k}{n} \quad (3.10)$$

where: m_k - is the number of values in the k interval, and n - the number of discrete process choice.

Normality criterion

$$X^2 = \sum_{k=1}^n \frac{(m_k - m_k \cdot P_k)^2}{m_k P_k} \quad (3.11)$$

Figure 7 shows the variation diagram of the square average of stationary accelerations influencing the vehicle wheels.

The variation diagram of the square average of wheel accelerations $\ddot{Z}_1$ has an average value of 2.343 m/s^2 . The stationary working process is considered under the hypothesis of $\alpha = 0.05$, as can be seen in Figure 8, representing the normal correlation function. Oscillations shown in Figure 9 are recorded at the level of: a) vehicle wheel; b) cabin floor; c) turning articulation; d) driver seat.

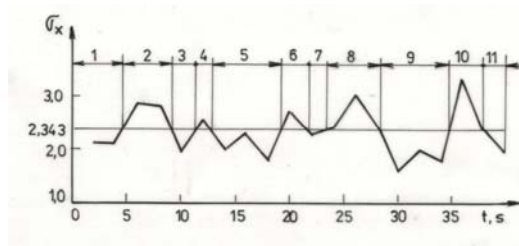


Fig.7 [3]

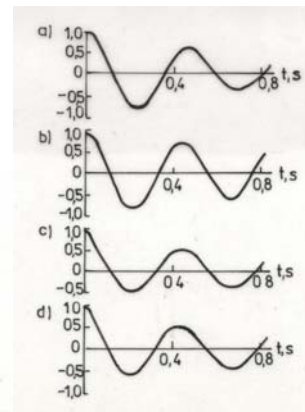


Fig.8 [3]

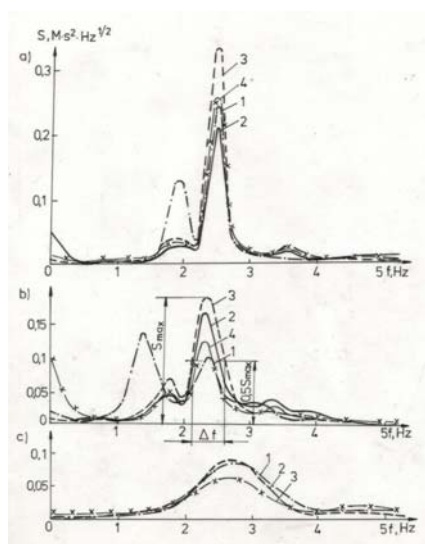


Figure 9 [3]

Figure 9 shows the calibrated spectrum densities of the acceleration during the vehicle movement under the speed of 2.9 m/s according to [3]: a) with no bucket load on a bumping road; b) with a bucket load of 6.800 kg , a total mass of $27,000 \text{ kg}$ on a bumping road; c) without load, on a concrete-covered road, for: 1 – the vehicle wheel; 2 – the cabin floor; 3 – the turning joint; 4 – the driver's seat.

The analysis of the normal correlation of the road profile with the operation of the loaded vehicle during transport with a speed of 2.8 m/s , for a 0.55 MN/m^2 load is a periodical harmonic process. The diagram of the normal spectrum of the load variation process (Figure 9, a) corresponds to a frequency of 2.5 Hz . Sanitary requirements give horizontal accelerations of 0.25 m/s^2 and vertical accelerations of 0.76 m/s^2 .

Table 2 shows the maximum values of accelerations during working, according to sanitary requirements.

The acceleration at the level of the wheel run, according to sanitary requirements, must have a frequency of 16 Hz, a speed of 1.4 m/s for a phase of 1.5 and the square value with a phase of 1.5 – 3.0.

Table 3 shows that the maximum value of continuous accelerations is higher than sanitary requirements. Maximum vertical acceleration values on the driver's seat are also higher than the appropriate requirements. The acceleration value at the level of the axle turning joint is according to sanitary norms but, at the same time its value at a frequency of at least 16 Hz does not comply with the requirements [3].

Decreasing the vehicle running speed up to 1.4 m/s determines a decrease of the acceleration values: the maximum ones by 1.5 times, the square average ones – by 1.5 – 3 times. But they are still higher than the sanitary norms. If wheel pressure is changed from 0.55 to 0.45 MN/m² the oscillation type does not change. The pressure reduction inside the wheels by 0.1 MN/m² leads to a decrease of maximum accelerations by 1.1 – 1.7 times, and of the square average ones by 1.2 – 3.5 times. When the vehicle moves with a speed of 2.8 m/s with the bucket loaded, the oscillation power decreases within the frequency range of 2 – 3 Hz by 1.5 times (see Figure 9, b) in comparison to the analogous running regime without a load (see Figure 9, c).

Statistical characteristics during the loader run without load, on site roads, with a speed of 2.8 m/s -
Table 2 [3]

Acceleration	$\bar{X}$	Xmin.	Xmax.	D_x	σ_x	V	X^2
$\ddot{X}$	0,924	-4,01	5,38	4,16	2,04	2,21	608
$\ddot{Z}_1$	0,325	-7,16	5,57	6,3	2,51	7,72	99
$\ddot{Z}_2$	-0,502	-5,2	4,33	3,5	1,87	-3,99	214
$\ddot{Z}_3$	-0,169	-6,37	5,84	5,52	2,35	-14	298
$\ddot{Z}_4$	0,929	-2,27	5,14	2,31	1,52	1,64	187
$\ddot{Z}_5$	-0,176	-8,49	9,09	5,52	2,35	-13	131
$\ddot{Z}_6$	0,414	-5,76	7,33	5,15	2,27	5,49	196
$\ddot{Z}_7$	0,091	-5,69	6,13	4,51	2,12	-23	153

4. CONCLUSIONS

The deformation taking place when the operator's seat load is 120 daN is of 0.007 m, and its rigidity characteristic is 25,100 +/- 8.1 daNs/m. The working operator mass is calculated at 75 kg and has an oscillation frequency of 11.2 +/- 0.97 Hz. During working, vehicles are submitted to vibrations, the seat suspension rigidity is 6,000 daN/m with a viscous damping coefficient of 8 daN/m, for a working frequency of 5 Hz. The deformation varies from 0 to 0.1 m.

The comfort of the operator [5, 6] is given by the possibility of adjusting the pneumatic seat suspension and head support during the vehicle operation. The result is an increased degree of comfort, which contributes to an increased productivity for handling over longer periods of time. Moreover, the height and position of the vehicle drive column can also be adjusted, according to the operator's own built.

Optionally the digital control of the vehicle may ensure a smooth operation of the automatic drive. In this case, the optimal working speed can easily be chosen according to the working conditions, while the coupling of the automatic shaft is smooth and without shocks, thus increasing the comfort level. When the vehicle climbs up or down a slope, or during load operation, the automatic drive may be shifted on manual control, with the help of a manual switch.

ROPS and the cabin have a standardized structure according to ISO 3471/1994 and SAE J 1040c ROPS (Roll Over Protective Structure), as well as ISO 3449/1992 FOPS (Falling Object Protective Structure). In order to obtain a significant vibration reduction at the level of the cabin floor and to increase the degree of comfort during

work, the cabin is mounted on viscous rubber puffers and silicone oil dampers (Figure 6). The cabin noise level is defined through standard 2000/14/EC [5, 6].

The vehicle stability and its operation mass may be affected by: the balance weight or balast, the tire dimension, as well as by other types of equipment which may be attached for the working purpose. The dynamic performance and vehicle comfort are determined by the quality of its control systems, transmission, braking, drive system, and suspension. Longitudinal accelerations are specific for dynamic characteristics, while the speed is specific for the impulse (impact force) developed during work, according to which the tensile strength of the vehicle is set. For modern loaders, these dimensions are measured by sensors which are limiting the engine operation control and the drive system control, thus ensuring an optimal operation of the entire vehicle.

Vibration monitoring [5]:

- the vehicle vibrations are monitored, thus ensuring the operator's efficiency and an increased productivity;
- from the ground level upwards, the design of the vibration reduction system of Cat 950H includes many standardized and optional characteristics;
- the oscillations of the back axle adapt to the road profile, thus allowing the cabin to stay firm on its position;
- the cabin is attached to the front frame with the help of a suspension, thus reducing shocks and load drops onto the ground;
- the joint link is equipped with two special valves allowing movement control from one axle frame to the other;
- during transport the cylinders are smoothly balancing the bucket, so as to limit the load placement and thus preventing vibration transfer onto the vehicle;

The Ride Control system is optional and it adjusts the functioning of the drive shaft, reducing bumps and jolts during loading and transport. The **Ride Control** ensures the engine *rpm* leveling to the load performance regime, when the vehicle runs over a bumping road. By handling the lever, the operator indicates a fast movement to high *rpm* with load, leading to a decrease of the cycle operation times and to an increased productivity.

With a digitally automated drive, the optimal working speed can easily be chosen, for vehicle transport conditions, while the coupling of the automatic shaft is handled smoothly and without shocks, thus increasing the degree of comfort. When the vehicle climbs up or down a slope, or during load operation, the automatic drive may be shifted on manual control, with the help of a manual switch.

- The digital automatic control of the system prevents the shaft piston hitting the cylinder, which would lead to bumps and jolts;
- The air suspension mounted at the seat level implements the decrease control of the vertical vibrations which are transferred upwards from the cabin floor.

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- [5] **x x x** – *Cat 950H Wheel Loader: Cat C7 Diesel Engine with ACERT Technology*, Power 161 kW, Bucket Capacity 2,7 to 4 mc, Operating Weight 18400 to 19500 Kg.
- [6] **x x x** – *Cat 966H Wheel Loader: Cat C11 Diesel Engine with ACERT Technology*, Power 211 kW, Bucket Capacity 3,5-4,8 mc, Operating Weight 23800 to 27300 kg
- [7] **x x x** – *Wheel Loader WA 700-3 Komatsu, Engine Power 478 kW, Bucket Capacities 8,7-9,4 mc, Operating Weight 70800 kg.*