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DYNAMIC STABILITY OF MACHINES WITH ARTICULATED FRAME STEERING IN AUTOMATIC WARNING SYSTEM ASPECT

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Abstract

The stability of operation of heavy machines is one of the main factors determining their working ability and safety. This problem is of a major importance for articulated frame machines that have worse stability than machines with the conventional steering system. So far the problem has not been solved satisfactorily. In this paper, problems of the dynamic stability of the articulated frame machine will be discussed. Using as an example the bucket loader, the structural form of physical and mathematical models of the machine's dynamics will be outlined. The mathematical model of the articulated frame machine's dynamics which is presented in this paper, is indispensable when one wants to define the control quantities for a system automatically warning the operator about the overturning of the machine. The technical concept of this system based on theoretical considerations in this field is also presented.

1. INTRODUCTION

Mobile earthmoving equipment, particularly with extension arm units, when operating, is exposed to the action of variable static and dynamic loads produced mainly by the movement of the machine on a rough and considerably inclined ground, the loading of the working system, the too high acceleration/delay of the working extension arm or of the driving system. As a result, the machines state of balance may be disturbed or the machine may even overturn.

The problem of stability against overturning is a major one in machines in serial production, despite the fact that these machines have several operational advantages, such as an articulated frame, for example. These machines have relatively worse stability on an incline, when compared to traditional equipment with turn wheels.

Studies conducted in the USA in 1970-1979 showed that 15 % from over 1400 recorded accidents caused by instability of earthmoving equipment resulted in the death of the operator [1]. These data indicate the great importance of ensuring the stability of earthmoving equipment.

The standard approach to the analysis of the stability of mobile articulated frame machines does not cover this problem thoroughly, since the majority of methods deal with the static analysis of the stability, e.g., [2]. The evaluation of analytical results obtained by these methods, i.e. the determination if the machine meets the stability requirements or not, depends mainly on the experience of the person that carries out the analysis, the adopted design principles and the regulations govern the design. The static analysis of stability, despite adopted empirical dynamic coefficients which increase static moments, is always associated with uncertainty as to the fact whether the dynamic phenomena that occur during the operation of mobile construction machines have been properly taken into account.

To protect the operator during the overturning of the machine strengthened ROPS (roll over protective structures) cabs are currently used. The requirements which the producers of such cabs have to fulfill are regulated by appropriate codes and standards, e.g., ISO 3471-1980. In some countries, e.g., Germany, for additional protection of the operator, special seats are used, with safety belt fastened at two points.

The above mentioned protection is of the passive type and it protects the operator only partially during the overturning of the construction machine but unfortunately it does not protect against the loss of stability by the machine and it does not eliminate the danger for the people who happen to be in the vicinity of the machine.

In response to the needs revealed by the above analysis, a research project, financed by the Committee on Scientific Research in Poland (Komitet Badań Naukowych w Polsce), aimed at a complex theoretical-experimental analysis of the mobile earthmoving machines dynamic stability and the construction of an active safety system for such machines is being carried out. The first results of this research project will be presented below.

2. SYSTEM OF AUTOMATIC STABILITY AND HOISTING CAPACITY CONTROL

The active system protecting the articulated frame machine against overturning developed in the Institute of Machinery Design and Operation is shown in fig. 1 [3, 4, 5].

The proposed device will provide perpetual control of the machines road wheels. When a machine with such a device installed, comes close to a stability-loss position during its operation, the controlling device will signal that a minimum permissible value of the normal reaction has been reached on one of the wheels. On the basis of this signal, a decision will be taken to counteract this phenomenon. This automatic control would protect the machine against the loss of stability in different operational situations, and it would be possible to reduce without risk the stability safety factor. As a result, a bucket loader's rated hoisting capacity, for instance, and thus its output could be increased (without changing the machine's weight). Moreover, the system presented in figure 1 makes possible the automatic determination of the reloaded excavated material, and thus of the loader's real output.

A system, similar to the one showed in figure 1 has been installed on a Ł 220 loader and is being tested. At the same time, theoretical analysis and simulation studies of the dynamic stability of mobile articulated frame machines are being conducted.

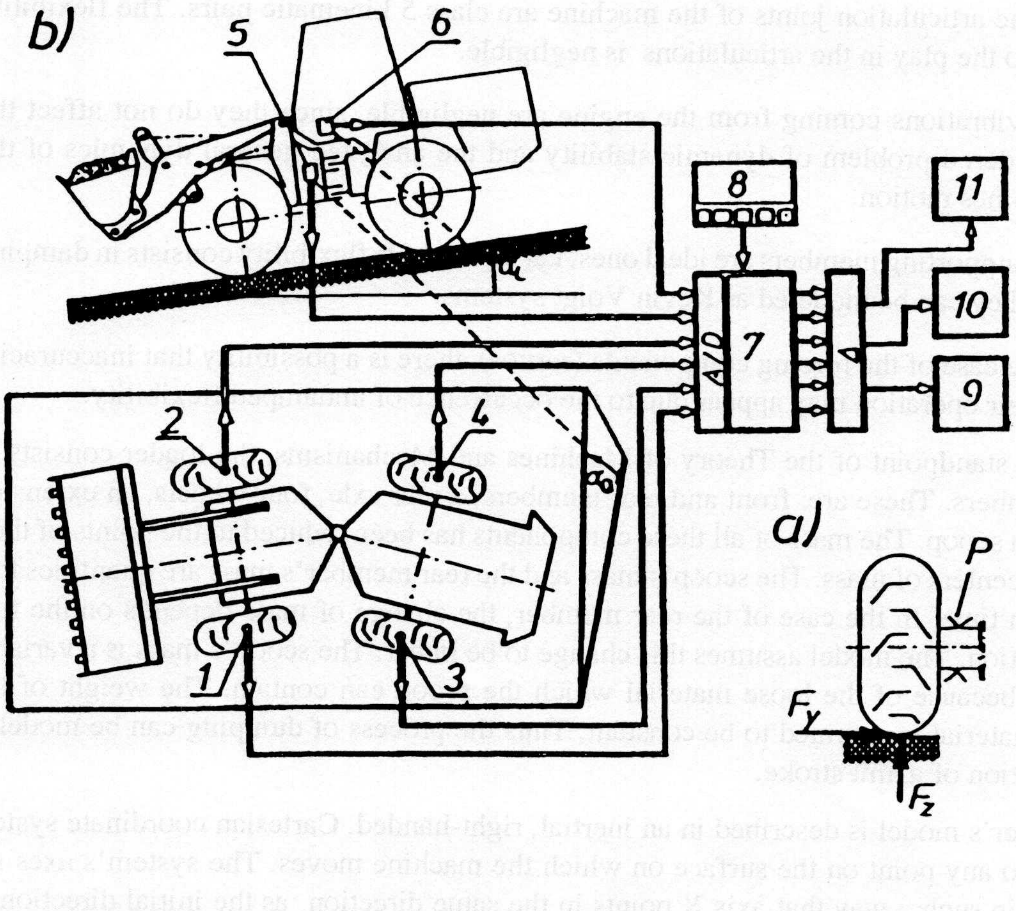

Figure 1. The system of automatic control of the stability, the capacity and the real efficiency of wheeled bucket loaders:
a) the arrangement of measuring transducers "P" for the current control of the machine's wheels normal reaction;
b) the system's diagram: 1,2,3,4-sensors of normal loading of machines wheels, 5,6-base inclination converters, 7-microprocessor, 8-set of nominal capacity switches, 9-warning device, 10-numerical indicator, 11-printer

3. MULTI-MASS DYNAMIC SPATIAL MODEL OF ARTICULATED FRAME MACHINE

3.1. Physical model of articulated frame loader

The articulated frame loader was modeled under the following simplifying assumptions:

- The loader is a multi-mass system of stiff links with imposed constraints depending on time and holonomic, bilateral and ideal constraints.
- All the articulation joints of the machine are class 5 kinematic pairs. The flexibility due to the play in the articulations is negligible.
- The vibrations coming from the engine are negligible, since they do not affect the considered problem of dynamic stability and the analyzed general dynamics of the machines motion.
- The supporting members are ideal ones, i.e. their whole flexibility consists in damping and they can be modeled as Kevin Voigt systems.
- In the case of the forcing components (servos), there is a possibility that inaccuracies in their operation may appear due to the occurrence of undamped flexibility.

From the standpoint of the Theory of Machines and Mechanisms, the loader consists of nine members. These are: front and rear members, a rear axle, four wheels, an extension arm and a scoop. The mass of all these components has been reduced to the points of their physical centers of mass. The scoop's mass and the rear member's mass are quantities that change in time. In the case of the rear member, the change of mass depends on the fuel consumption. The model assumes this change to be linear. The scoop's mass is a variable quantity because of the loose material which the scoop can contain. The weight of the loaded material is assumed to be constant. Thus the process of dumping can be modeled as a function of a unit stroke.

The loader's model is described in an inertial, right-handed, Cartesian coordinate system [1] tied to any point on the surface on which the machine moves. The system's axes are oriented in such a way that axis X points in the same direction as the initial direction in which the rear member was positioned. Axis Z points perpendicularly upwards along the action of gravitational force vector g.

Each mass of the loader has been assigned a local, right-handed, Cartesian coordinate system placed on the axis of a kinematic pair. This system is invariably bound with this axis, i.e. it does not move relative to the member this axis is tied to. Axis Z of local coordinate systems is always directed along the axis of an articulated joint. The coordinate systems, being stationary to each other, are bound with the rear member and the front member. In the case of the rear member, these systems are positioned on the axes of the wheels rotation and on the axis of the machine's turning articulated joint, whereas in the case of the front member–systems are associated with the axles of the wheels and the articulated joint of the scoop's extension arm.

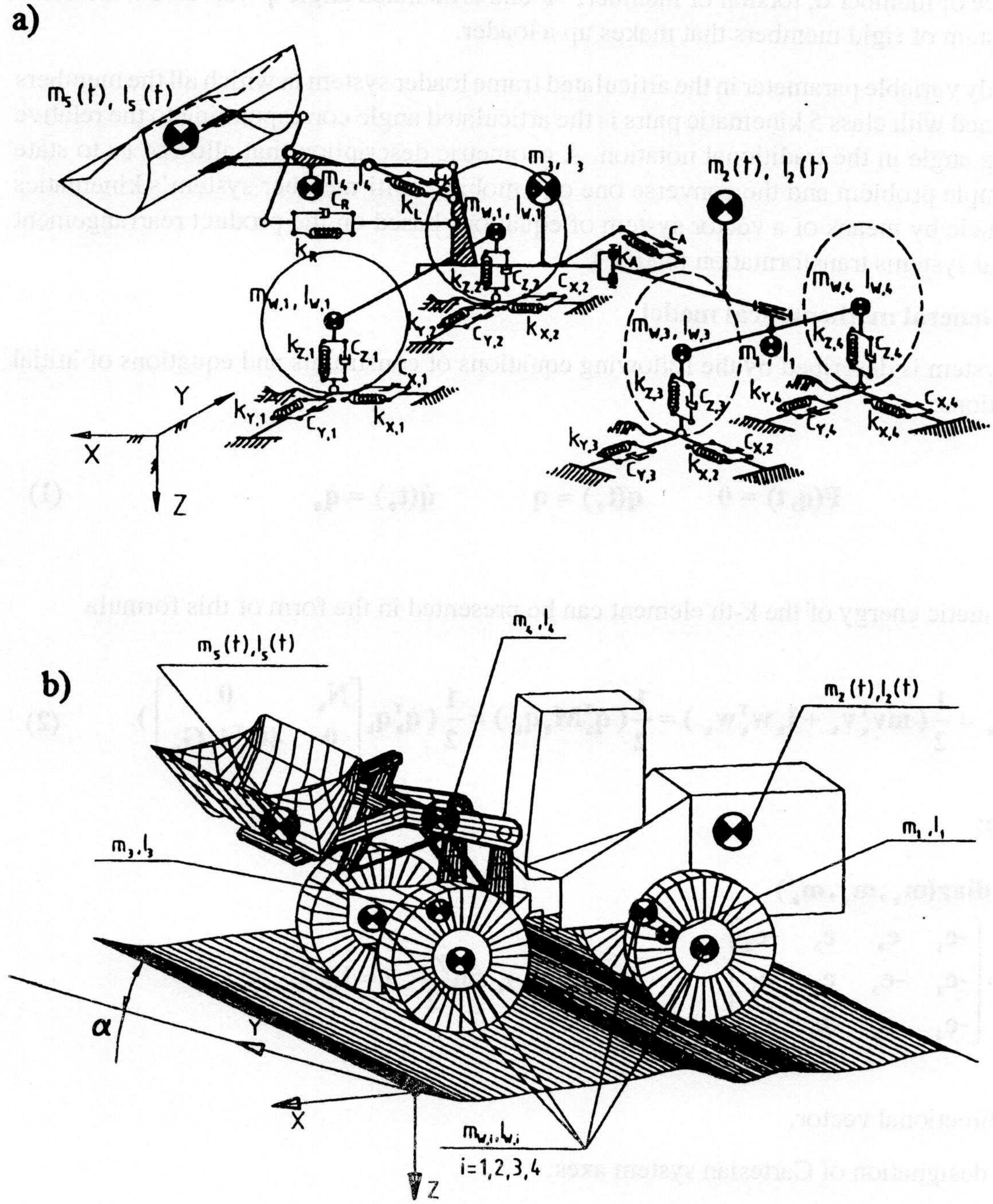

Figure 2. The physical model of the articulated frame loader
a) physical model,
b) computers visualization

Since the body's and the scoop's centers of mass are points that wander in space relative to the position of the members then the values of the tensors of these members moments of inertia also change in time.

Denavit-Hartenberg notation which introduces four parametric values: length of member **a**, distance of member **d,** torsion of member, Φ and articulated angle γ was used to describe the system of rigid members that makes up a loader.

The only variable parameter in the articulated frame loader system in which all the members are joined with class 5 kinematic pairs is the articulated angle corresponding to the relative turning angle in the traditional notation. A geometric description that allowed us to state the simple problem and the converse one of a mobile multi-member system's kinematics was made by means of a vector system of equations based on the product rearrangement of local systems transformation matrix **T**.

3.2. General mathematical model

The system is described by the following equations of constraints and equations of initial conditions:

$$\mathbf{F(q,t)=0} \qquad \mathbf{q(t_o)=q} \qquad \mathbf{\dot{q}(t_o)=\dot{q}_o} \qquad (1)$$

The kinetic energy of the k-th element can be presented in the form of this formula

$$\mathbf{E_k=\frac{1}{2}(mv_k^Tv_k+I_kw_k^Tw_k)=\frac{1}{2}(\dot{q}_k^TM_k\dot{q}_k)=\frac{1}{2}(\dot{q}_k^T\dot{q}_k\begin{bmatrix}N_k & 0\\ 0 & 4G_k^TI_kG_k\end{bmatrix})}. \qquad (2)$$

where:

$$\mathbf{N_k=diag(m_k,m_k,m_k)}$$

$$\mathbf{G_k=\begin{bmatrix}-e_1 & e_0 & e_3 & -e_2\\ -e_2 & -e_3 & e_0 & e_1\\ -e_3 & -e_2 & -e_1 & e_0\end{bmatrix}_k} \qquad \mathbf{(\hat{e_i})_k=(u_i)_k\sin\frac{\gamma_k}{2}};\ i=1,2,3$$

u–a directional vector,

i–the designation of Cartesian system axes,

k–a members number.

The total energy of the articulated loader can be expressed as the sum of the kinematic energy of its individual members.

By considering the problem after generalized forces have been applied at points $\mathbf{A_k}$ on each of the members one can determine the values of the forces from the equation of virtual work $\delta \mathbf{L_k}$ on virtual displacement $\delta \mathbf{r_k}$. The equation of virtual forces has this form:

$$\mathbf{Q}_k = \begin{bmatrix} \mathbf{t}_k \\ 2\mathbf{B}_k^T(\mathbf{s}_k \times \mathbf{t}_k) \end{bmatrix} + \begin{bmatrix} \mathbf{0} \\ 2\mathbf{B}_k^T \mathbf{m}_k \end{bmatrix} \quad (3)$$

where:

$$\mathbf{B}_k^T = \begin{bmatrix} -e_1 & e_0 & -e_3 & e_2 \\ -e_2 & e_3 & e_0 & -e_1 \\ -e_3 & -e_2 & e_1 & e_0 \end{bmatrix}$$

$\mathbf{m_k}$–virtual moments,

$\mathbf{t_k}$–virtual forces.

By putting the above into the Lagrange general equation of motion and performing the operations, the following system of equations correlated with the initial conditions will be obtained:

$$\begin{cases} \mathbf{M}\ddot{\mathbf{q}} = \mathbf{g} - \mathbf{F}_q^T \Lambda \\ \mathbf{F}(\mathbf{q}, t) = 0 \end{cases} \quad (4)$$

where:

$$\mathbf{g}_k = \left[\mathbf{g}_1^T ; \mathbf{g}_2^T \cdots \mathbf{g}_c^T\right]^T = \mathbf{Q}_k - \begin{bmatrix} \mathbf{0} \\ 8\overset{\circ}{\mathbf{G}}{}_k^T \mathbf{I}_k \mathbf{G}_k \dot{\mathbf{p}}_k \end{bmatrix}$$

E–the sum of the energy of the members,

Q–the sum of the generalized forces,

Λ–Lagrange multipliers.

The equations of motion obtained by the Lagrange method or the Kane method shown in this paper, allow after the assignment of properly interpreted equations of constraints reflecting the pairing of low-pressure tires with any base and the addition of a proper description of the generalized internal and external forces acting on the machine during its operation, to simulate the motion of the loader in the cases which are currently under investigation. Solutions can be found by means of ready-made software packages. In the above case, it was the Applied Motion software by RASNA.

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