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Radka JÍROVÁ¹, Lubomír PEŠÍK²

DYNAMIC LOAD OF LINEAR GUIDING SYSTEMS IN HANDLING MACHINES

Summary. Linear guiding systems are used in machines for relative translational motion of components, especially for handling devices, which are the basis of production lines. In these operating conditions, the linear guiding systems are dynamically loaded. Dynamical influences may lead to the decrease and damage of bearings. The aim of this paper was to calculate dynamical influences of a mass inertia on linear guiding systems. This issue was solved by a numerical calculation of differential motion equations in 2D. The motion was calculated on an example of a handling machine for welding car bodies. The result is a percentage representation of the dynamic forces on the total load of the linear guiding systems. The result shows the percentage representation in the solved case as more than 10%.

Keywords: linear guiding system, dynamic load, 2D motion

1. INTRODUCTION

Nowadays, the linear guiding system is an indispensable part of handling machines. They are usually used in serial productions with a high degree of robotisation. The advantage of

¹ Faculty of Mechanical Engineering, Technical University of Liberec, Studenstká 2, 460 01 Liberec, Czech Republic. Email: radka.jirova@tul.cz

² Faculty of Mechanical Engineering, Technical University of Liberec, Studenstká 2, 460 01 Liberec, Czech Republic. Email: lubomir.pesik@tul.cz

linear guiding systems is their versatility, high load capacity and operational reliability. In many cases, they enable the replacement of complex spatial mechanisms. The objects motion is usually electrically controlled by drives with stepper motors [1,3,5,6,7,9].

The design of linear guiding systems are usually based on the evaluation of a static load, the operating velocity and the sufficient stiffness of the assembly. Producers of handling machines usually use this approach in the design of linear guiding systems. However, during their operation, dynamic load occurs due to the incident of the inertial forces of the mass. Therefore, the objective of this paper is the assessment of the operational dynamical influences. These are evaluated as the percentage representation of the dynamic load on the total load.

2. CALCULATION OF DIFFERENTIAL MOTION EQUATIONS

The influence of the dynamic load was solved on an example of a handling machine for welding car bodies. The motion of this handling machine is a linear translation in a horizontal plane. The clamping frames transported were, in this case, hung on the cart, which is assembled to the linear guiding systems. For linear movement, the handling machine uses the rack and pinion gear, onto which power from the stepper motor is transmitted. The stepper motor is situated in the handling machine cart. The clamping frames shows that there is a relatively high inertial mass, which occurs during the acceleration and deceleration of dynamical forces transmitted to the linear guiding systems.

For the calculation of the dynamic forces, the linear guiding systems may be represented as an elastic and damping part [2]. Hence, during linear translation, oscillating motion occurs. The total movement of the handling machine consists of absolute translation and relative rotation around the contact point of the gear. The solution of the dynamic forces is simplified in a 2D planar motion.

2.1. D'Alembert's principle

d'Alembert's principle [4] is used for creating motion equations in 2D. Fig. 1 shows the general planar motion with an elastic and damping member.

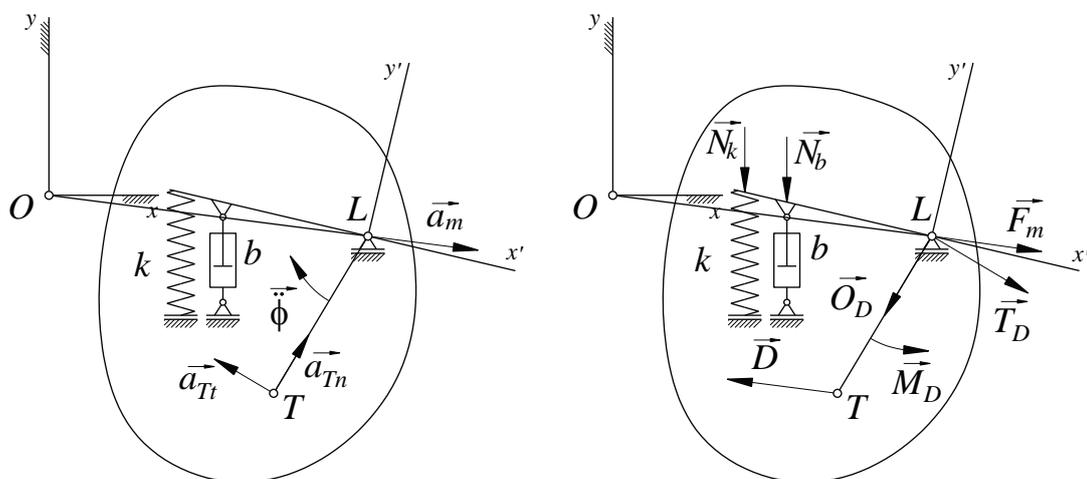


Fig. 1. Kinematic and dynamic schema of the system.

The kinematic schema is shown on the left, while on the right is the dynamic schema, wherein k_φ is torsion stiffness of the linear guiding system, b_φ torsion damping coefficient of linear guiding system, \vec{a}_{Tt} tangential acceleration vector, \vec{a}_{Tn} normal acceleration vector, \vec{a}_m acceleration vector of stepper motor, $\vec{\varphi}$ angular position vector, \vec{N}_k elastic force vector, \vec{N}_b damping force vector, \vec{F}_m force vector of stepper motor, \vec{D} dynamic force vector of linear motion, \vec{O}_D centrifugal force vector of rotation motion, \vec{T}_D tangential force vector of rotation motion and \vec{M}_D is dynamic moment [3]. Point O is the origin of coordinate system, L is rotation point and T is centre of mass.

Dynamic force vector \vec{D} can be calculated as:

$$\vec{D} = -\vec{a}_n m \quad (1)$$

where in m is mass of the system.

Tangential force vector \vec{T}_D is:

$$\vec{T}_D = -\vec{a}_{Tt} m \quad (2)$$

centrifugal force vector \vec{O}_D is:

$$\vec{O}_D = -\vec{a}_{Tn} m \quad (3)$$

dynamic moment \vec{M}_D is:

$$\vec{M}_D = -\vec{\varphi} J_L \quad (4)$$

where in J_L is moment of inertia with respect to the L point.

Elastic force vector \vec{N}_k and damping force vector \vec{N}_b are:

$$\vec{N}_k = \vec{\varphi} k_\varphi \quad (5)$$

$$\vec{N}_b = \vec{\dot{\varphi}} b_\varphi \quad (6)$$

where in $\vec{\varphi}$ is angular position vector and $\vec{\dot{\varphi}}$ angular velocity vector.

Dynamic load of linear bearing system can be calculated:

$$\vec{N}_D = \vec{N}_k + \vec{N}_b \quad (7)$$

2.2. Differential motion equations

The cart for transporting clamping frames is connected to the frame of the handling machine by eight linear guiding systems. In the worst-case scenario, it is possible to expect only two fully loaded linear guiding systems. Fig. 2 shows the solution of the system regarding the force balance with respect to d'Alembert's principle.

Differential equations of motion are:

$$F_m(t) - ma_m(t) + me \frac{d^2}{dt^2} \varphi(t) \cos(\beta + \varphi(t)) - me \left(\frac{d}{dt} \varphi(t) \right)^2 \sin(\beta + \varphi(t)) = 0 \quad (8)$$

$$-me \frac{d^2}{dt^2} \varphi(t) \sin(\beta + \varphi(t)) - me \left(\frac{d}{dt} \varphi(t) \right)^2 \cos(\beta + \varphi(t)) - k\varphi(t)(a_1 + a_2) - b \frac{d}{dt} \varphi(t)(a_1 + a_2) = 0 \quad (9)$$

$$J_L \frac{d^2}{dt^2} \varphi(t) - ma_m(t)e \cos(\beta + \varphi(t)) + k\varphi(t)(a_1^2 + a_2^2) + b \frac{d}{dt} \varphi(t)(a_1^2 + a_2^2) = 0 \quad (10)$$

wherein a_1 is distance between point L and linear guiding system 1, a_2 distance between point L and linear guiding system 2, e distance between point L and T , k stiffness of the linear guiding system and b damping coefficient of the linear guiding system [5].

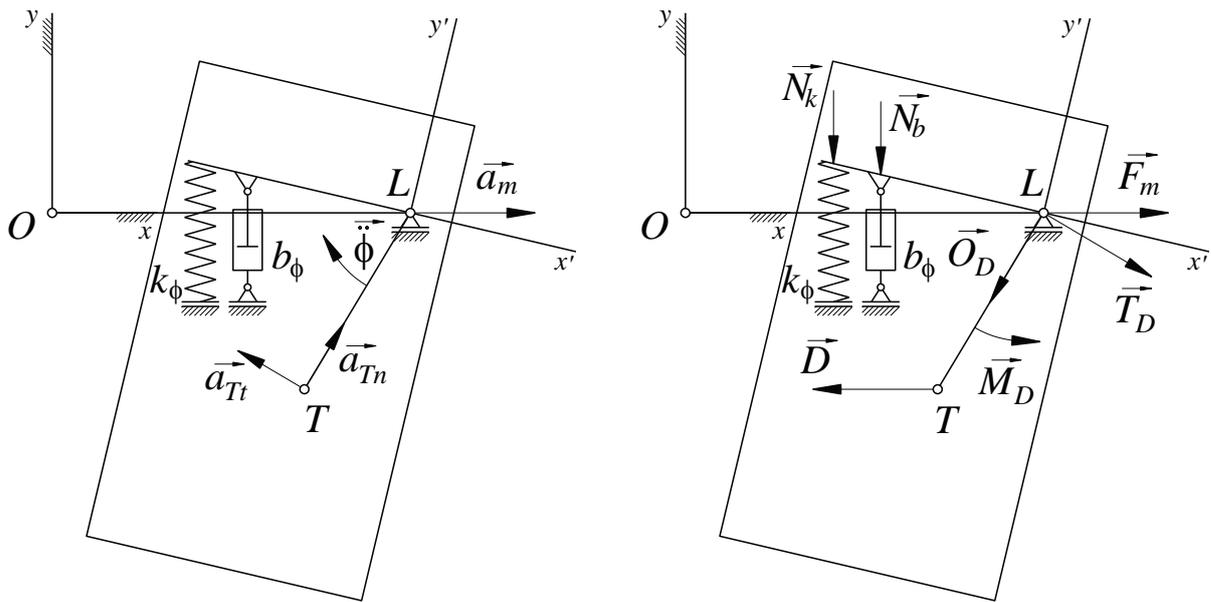


Fig. 2. Kinematic and dynamic schema of the handling machine cart.

2.3. Static force of linear guiding systems

The linear guiding systems are loaded by static force, cart mass and mass of the clamping frames. In Fig. 3 this is shown in the dynamic schema in the xy and yz plane.

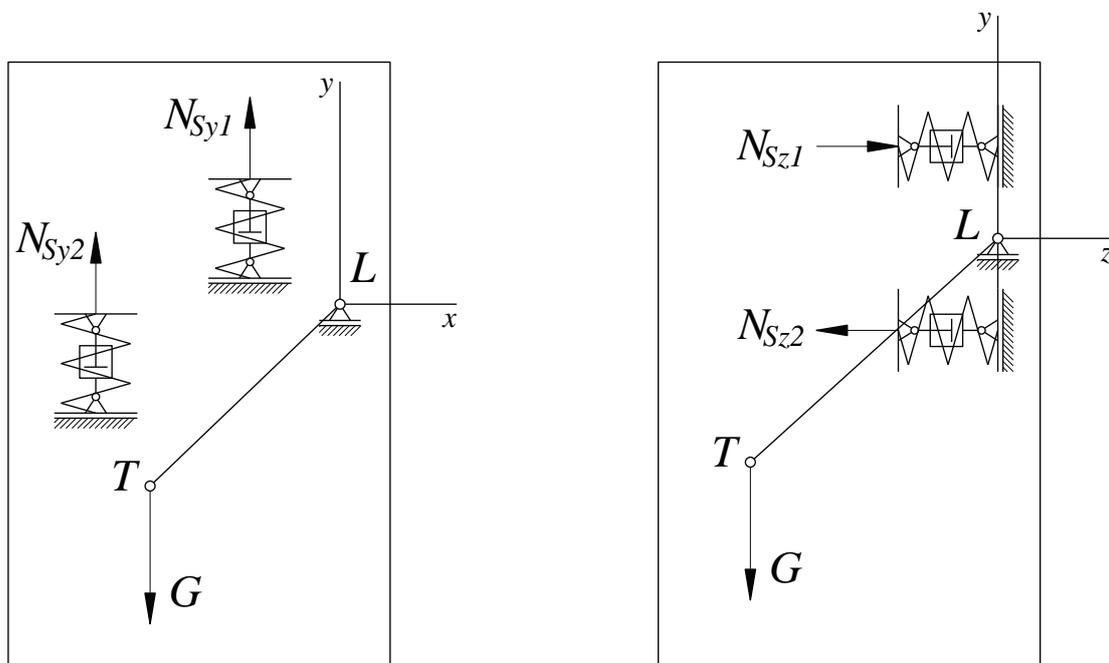


Fig. 3. Static load in axis y and z direction.

Wherein N_{Sy1} and N_{Sy2} are static loads of linear guiding systems 1 and 2 in axis y direction, N_{Sz1} and N_{Sz2} are static loads of linear guiding systems 1 and 2 in axis z direction and G gravity force.

Static load of linear guiding systems 1 and 2 in plane xy is:

$$N_{Sy1} = mg \left(1 - \frac{x_T - a_1}{a_2 - a_1} \right) \quad (11)$$

$$N_{Sy2} = mg \frac{x_T - a_1}{a_2 - a_1} \quad (12)$$

where in g is acceleration of gravity and x_T position of mass center.

Static load of the linear guiding systems 1 and 2 in plane yz is:

$$N_{Sz} = N_{Sz1} = N_{Sz2} = mg \frac{z_T}{2a_3} \quad (13)$$

where in a_3 is distance between point L and force N_{Sz} and z_T position of mass centre.

Static load of linear bearing systems 1 and 2 can then be calculated thus:

$$N_{S1} = \sqrt{N_{Sz}^2 + N_{Sy1}^2} \quad (14)$$

$$N_{S2} = \sqrt{N_{Sz}^2 + N_{Sy2}^2} \quad (15)$$

2.4. Linear motion of stepper motor

Fig. 4 – Fig. 6 show position x_m , velocity v_m and acceleration a_m of cart translation.

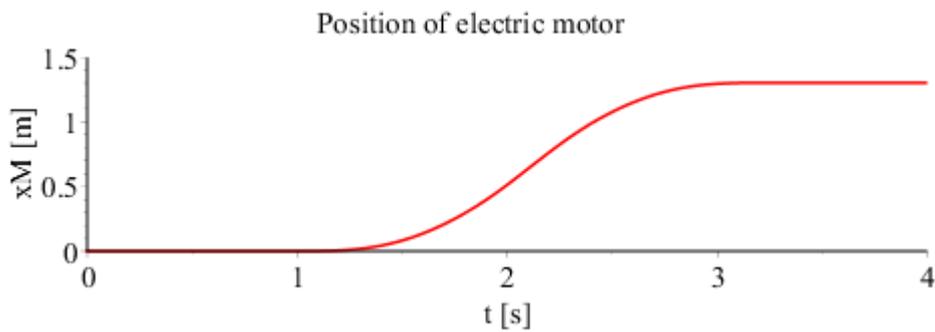


Fig. 4. Position of the system.

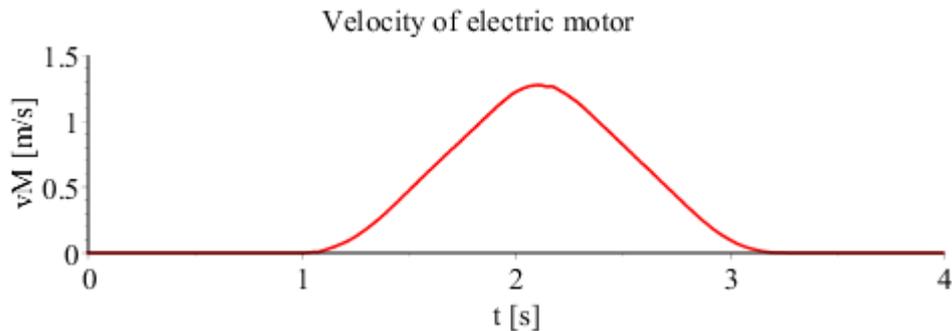


Fig. 5. Velocity of the system.

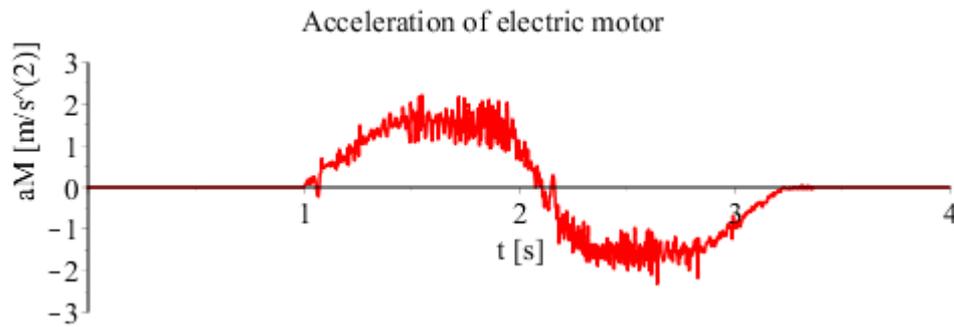


Fig. 6. Acceleration of the system.

It is appropriate to approximate the acceleration a_m by a smooth function for the differential equations calculation. The suitable smooth function may be the hyperbolic tangent (Fig. 7). Fig. 7 shows a comparison between the measured values of acceleration and the approximation function.

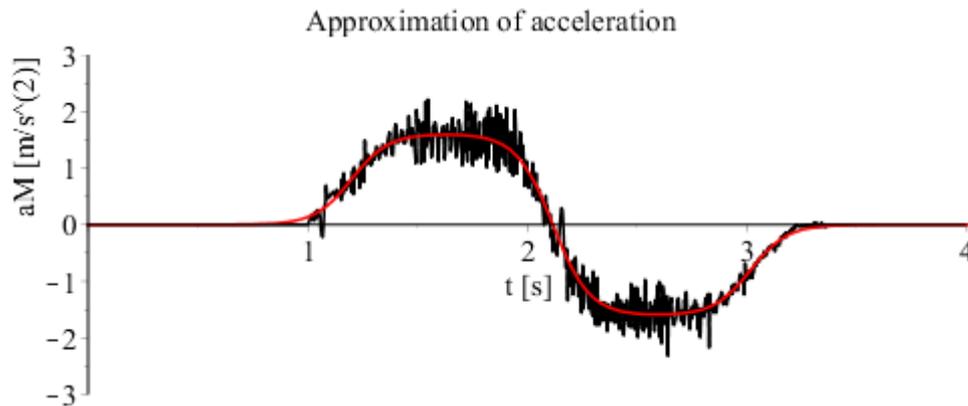


Fig. 7. Acceleration of the system.

3. RESULTS

Fig. 8 and Fig. 9 show kinematics of the relative rotation.

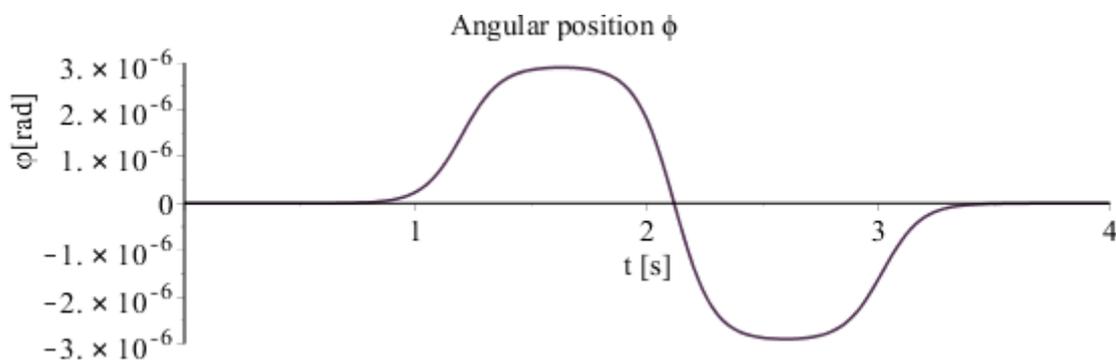


Fig. 8. Angular position of the system.

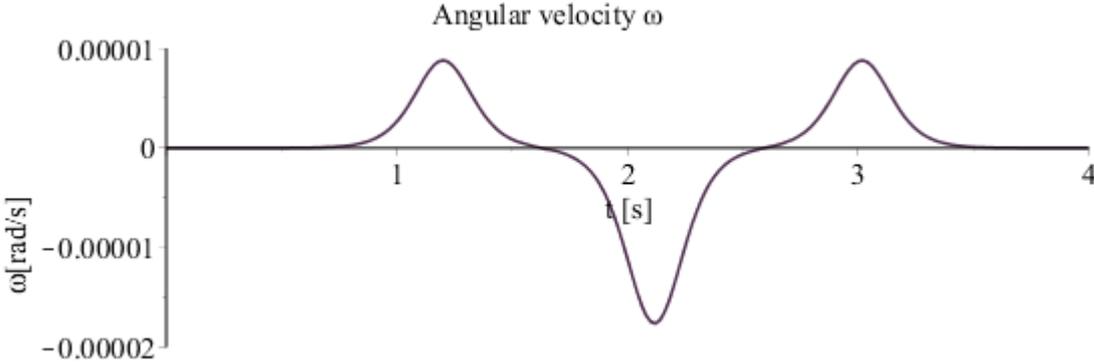


Fig. 9. Angular velocity of the system.

In Fig. 10 shows the stepper motor force transmitted to the rack.

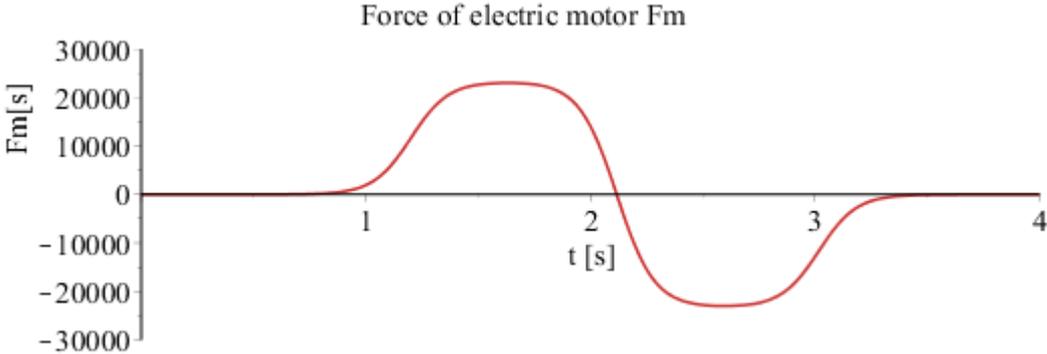


Fig. 10. Force of the stepper motor.

Fig. 11 – Fig. 15 summarises the solution results of the differential motion equations and shows dynamic forces loading linear guiding systems 1 and 2.

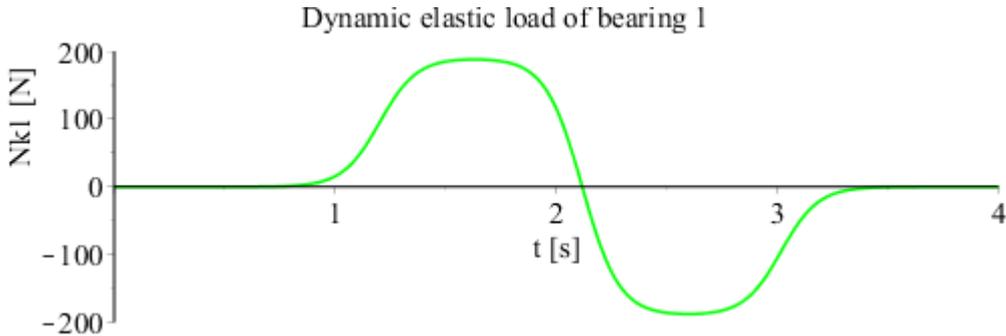


Fig. 11. Elastic force of linear guiding system 1

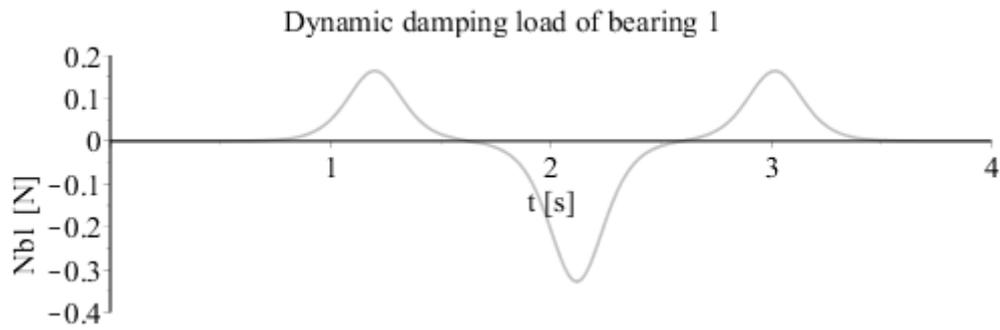


Fig. 12. Damping force of linear guiding system 1

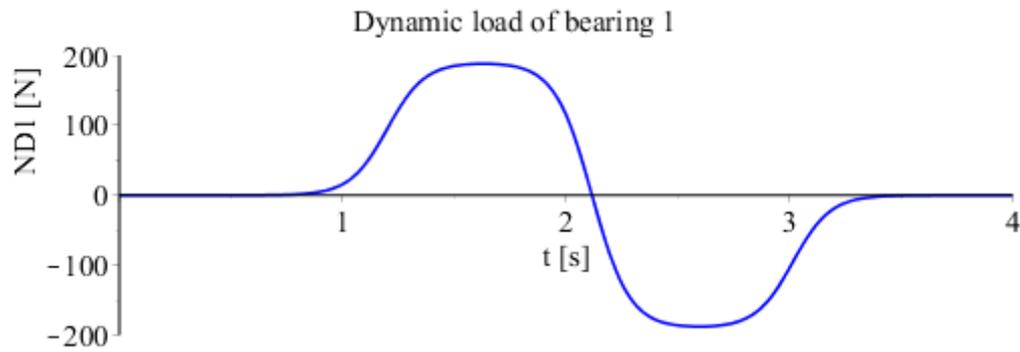


Fig. 13. Dynamic force of linear guiding system 1

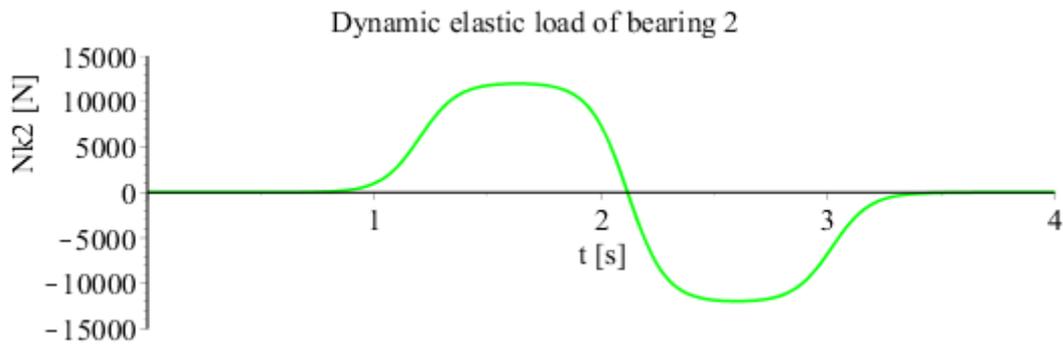


Fig. 14. Elastic force of linear guiding system 2

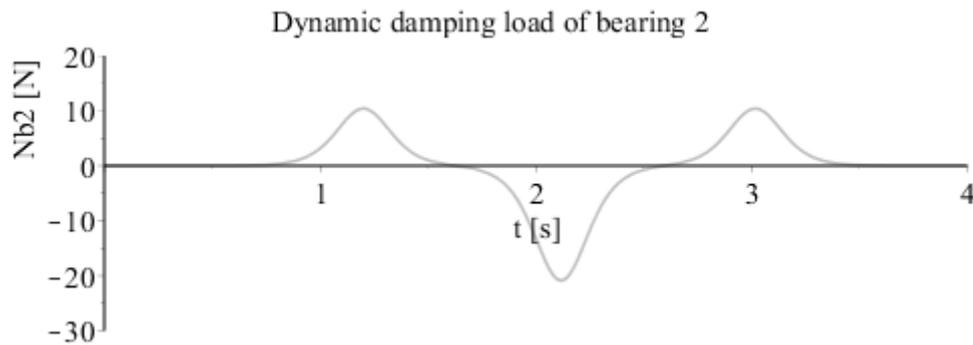


Fig. 15. Damping force of linear guiding system 2

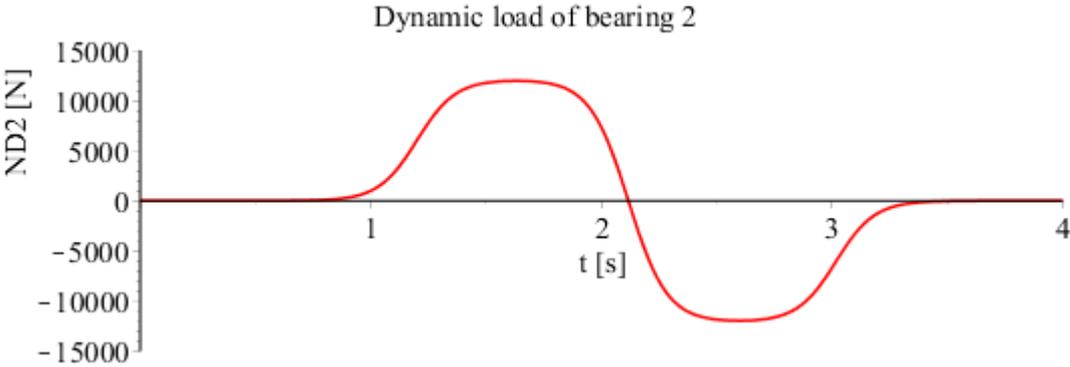


Fig. 16. Dynamic force of linear guiding system 2

Fig. 17 shows the static load of linear guiding systems 1 and 2.

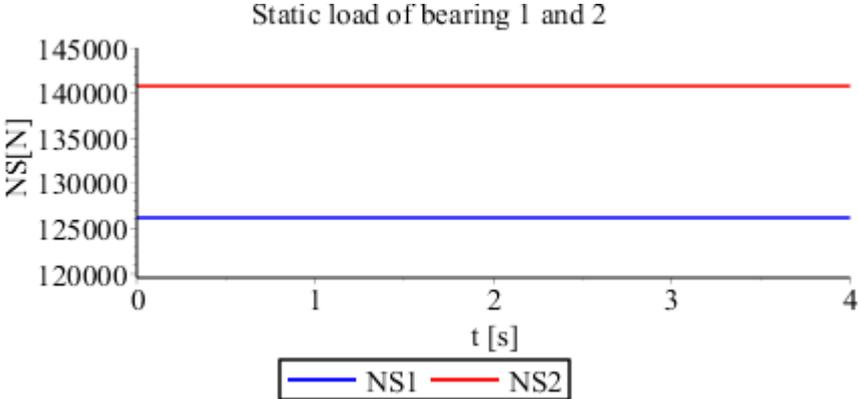


Fig. 17. Static force of linear guiding systems 1 and 2

In Fig. 18, the total load of the linear guiding system is shown. Fig. 16 and Fig. 18 shows that the percentage representation of the dynamic load on the total load is around 10%.

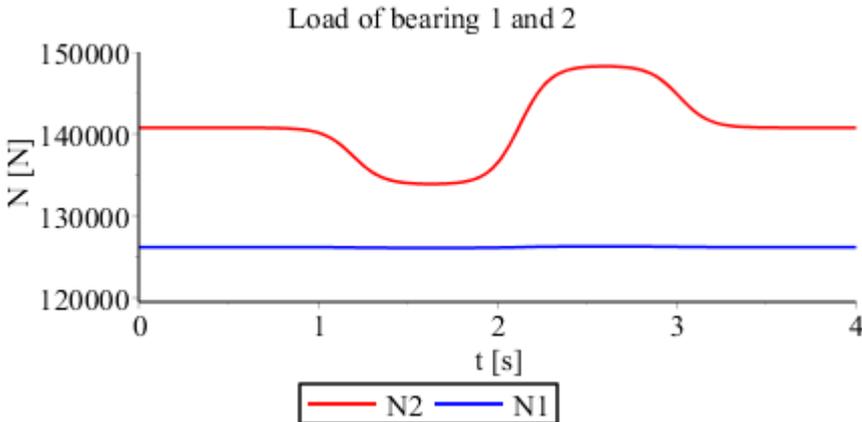


Fig. 18. Load of linear guiding systems 1 and 2

4. CONCLUSIONS

Producers of handling machines usually do not know the operating conditions their machines undergo. Therefore, the linear guiding systems are designed for static strength. However, during operations, dynamical inertial influences appear which are in turn transmitted to the linear guiding systems. A dynamic load may be superposed on the static load. It leads to the total load increase of the linear guiding systems that directly cause a decrease in the bearing life. Although the producers recommend replacement of the linear guiding systems near the end of the supposed bearing life, the dynamical influences may have caused their damage before then. It may lead to significant production losses. Therefore, it is only appropriate to evaluate the dynamic load with the expectation of a decrease in the bearing life.

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