Aspects about the influence of the lubricant from a rectilinear pair above the work accuracy of the elastic elements from the high precision mechanisms

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Abstract: - The paper presents the determination by experiment of the vibration of a long elastic cinematic element with a mobile rectilinear pair, lubricated by two kinds of oil with low and high values for lubrication properties and cinematic viscosity. The lubricant pressures' field is analyzed by using the finite element method (COSMOSM program), starting with the datum obtained by experiment in the analysis of the elastic cinematic element vibration. The vibration acceleration for the elastic element was measured and the spectrum analysis was attached for each case separately. The efficacy acceleration and the power spectral density for different kinds of oil were comparatively presented.

Key-Words: - Lubricated rectilinear pair, elastic element, pressures field, cavitations, spectral analysis

1 Introduction

As a result of several stringent requirements concerning higher work speeds and position accuracy of some points of the cinematic elements, it is more necessary to take into consideration the influence of the lubricant from the cinematic pairs. The pressure fields from the lubricant have a big influence over the vibrations that occur during the motion.

This work tries to prevent and control the apparition of the lubricant film-breaking phenomenon, which has consequences in the gripping and the working accuracy for the high precision mechanisms, especially for the robotic parts.

2 Problem Formulation

The considered mechanism is a slider-crank and connecting-rod assembly presented in figure 1. The vibrations of one long elastic element subjected to various spin speed of the leading element are experimentally measured. The rectilinear pair on the elastic element is lubricated with two kinds of oil.

Previously, the solving of the motion's equations of the elastic cinematic elements was made by assumption that these are continuous medium with infinite degrees of freedom, or they are discrete systems (using finite element method), or using the Lagrange's method from elastic-dynamics [3], [7].

This paper solves the equations of motion using Hamilton's principle and combines it with the Reynolds's lubrication equation [6].

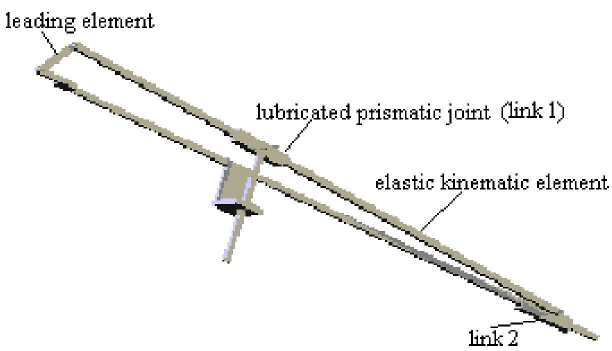

Fig. 1 The slider-crank and connecting-rod assembly

A crank and connecting-rod assembly with slide-bar is considered with rigid cinematic elements, excepting the 1m length element. A lubricated rectilinear pair with 0.100m length slides on this element.

Figure 2 shows the elastic cinematic element with the rectilinear pair.

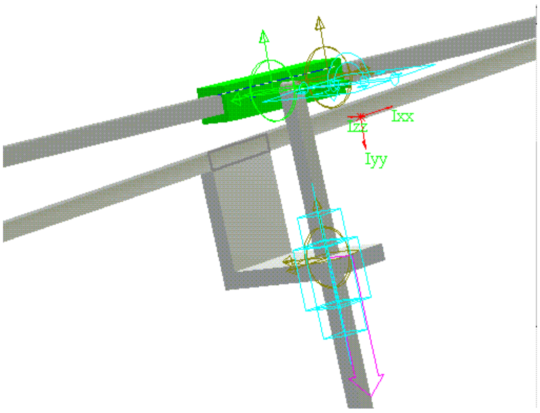

Fig. 2 Elastic cinematic element with mobile rectilinear pair and vertical element technologically loaded.

3 Algorithm, equations of motion and equation of the pressure field

The long cinematic element with the rectilinear pair slides on is considered linear elastic, with plane motion [1]. The equations of motion are obtained by using Hamilton's Principle from elastic-dynamics. The cinematic energy used in applying this principle is given by generalized speeds field theory used for elastic bodies.

This fact supposes the directly inclusion of inertial terms in the equations of motion [2]. The pressures distribution is computed by using Reynolds equations of lubrication for a viscous and incompressible fluid. An averaging on the transverse direction (z) is applied to this fluid and it is computed depending on the external force practiced on the cinematic element by the fluid [6]:

$$\xi^3 \frac{\partial^2 P}{\partial X^2} + 3\xi^2 \frac{\partial \xi}{\partial X}\frac{\partial P}{\partial X} = 6\mu V_r(t)\frac{\partial \xi}{\partial X}, \qquad (1)$$

where: P is the distributed pressure from the film, $v_r(t)$ is the relative speed between the elastic element and the mobile rectilinear pair, ξ is the width of the film, μ is the dynamic viscosity of the film.

Moreover, it is verified the next relation by using geometry:

$$\xi = \frac{H}{2} - u_2\left(x^* + X, t\right). \qquad (2)$$

The boundary conditions for p(X,t) are given as:

$X = 0 \Rightarrow P(0,t) = p_{atm}$; $\quad X = l \Rightarrow P(l,t) = p_{atm}$,

where p is the atmospheric pressure.

The general solution for the equation (1) is:

$$P(X,t) = 6\mu V_r(t) I_1(X,t) + C I_2(X,t) + C_1, \qquad (3)$$

where:

$$I_1(X,t) = \int dX \Big/ \left[H/2 - u_2\left(X + x^*\right)\right]^2 \qquad (4)$$

$$I_2(X,t) = \int dX \Big/ \left[H/2 - u_2\left(X + x^*\right)\right]^3. \qquad (5)$$

The constants C and C1 can be computed from the boundary conditions. The integrals I1 and I2 are computed later in this article by using Fourier Transform method.

Cavitation phenomenon occurs when the pressure computed with equation (3) decreases below the ambient pressure. In this case, Reynolds equation can't be applied inside the cavitation region.

The cavitation occurring position can be established by changing the boundary conditions (the pressures curve is easily translated in the cavitation region).

The boundary conditions are:

$$X = 0 \Rightarrow P(0,t) = p_{atm}; \quad P(\overline{X},t) = p_{atm} \quad \text{iar} \quad \left.\frac{\partial p}{\partial X}\right|_{X=\overline{X}} = 0 \qquad (7)$$

where: X is the position of the point where cavitation occurs.

The rectilinear pair is lubricated by oil with average lubricating properties and cinematic viscosity; its specific characteristics are shown in table 1.

Table 1 The characteristics of the types of oil used as lubricants in the rectilinear pear

Oil	Density [kg/m^3]	Dynamic Viscosity [Pa.s]
TB32E without additives STAS 742/81	890	0.02848
SHELL TONNA T STAS 871/68	894	0.19668

Algorithm:

The solving algorithm steps are as follows:

- the cinematic and kinetics-static analysis of the mechanism is made considering the hypothesis of the rigidity of all its cinematic elements;
- the equations of motion are solved by introducing an external force R/(B*L) on the region occupied by the rectilinear pair and then, the pressures field is determined in the first approximation p1(X, t);
- the distributed force p1(X, t)/B, where B is the rectilinear pair width, is introduced in the new external force; then the equations of motion are solved again and so it is determined the pressures field in the second approximation.

This proceeding may continue by successively improving the mechanism dynamic reply, depending on the wished accuracy of the calculus [4].

It was achieved a MAPLE Program for computing this algorithm and it was determined the necessary

datum by two iterations, with enough accuracy, compared afterwards with the experimental datum.
An important average used for simplifying the program was no considering the pressures field depending on time after the first iteration, but considering its effective value. Otherwise the model is difficult and it couldn't be solved anymore.
Furthermore, the pressure field from the rectilinear pair was modeled by finite element method with COSMOSM program. The specific heights of the oil film to the entrance and the exit of the pair were taken from the theoretical results of the deformations after the MAPLE program calculation of the deformations field.

4 Problem Solution

4.1 The theoretical results

The theoretical results for the deformation field are presented in figure 3 for the inferior oil in condition of the speed of 387 rpm at the leading element; figure 4 presents the same values but for the superior oil in the same work conditions.

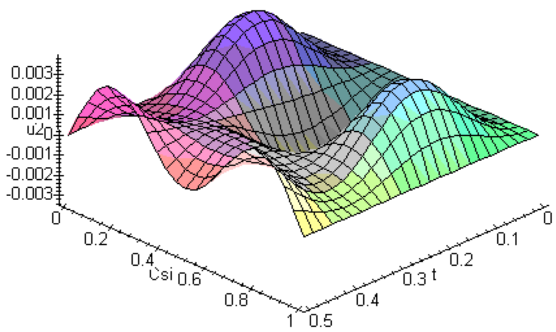

Fig. 3 The deformations field for the 387 rpm inferior oil

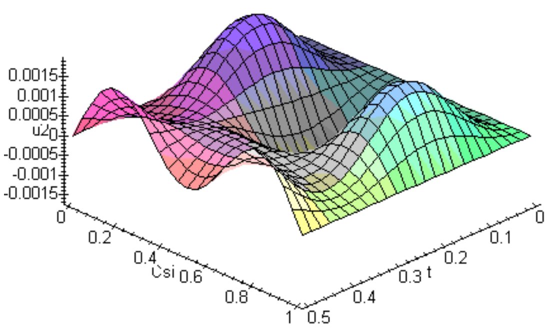

Fig. 4 The deformations field for the 387 rpm superior oil

The theoretical pressure fields computed by the mathematical model for the two types of oil and the 387 rpm are represented in figure 5, respectively 6. The maximal distributed pressure is p= 0.016 N/m, reading the pressures map for rotation value 387 rpm, at the moment t=0.0625 s and x= 0.1m for the inferior oil, and the maximal distributed pressure is p=0.11N/m for 387 rpm rotation value at the moment t=0.0379s and x=0.1m for the inferior oil.

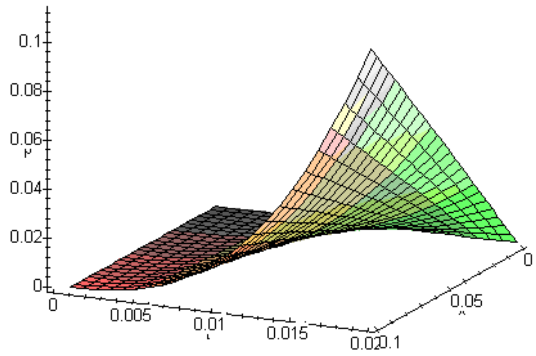

Fig. 5 Pressures field in lubricant at 387 rpm - superior oil

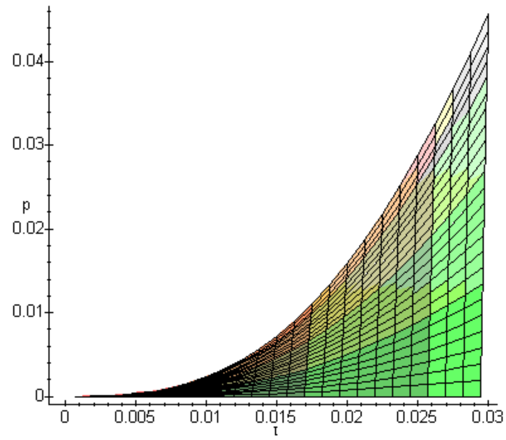

Fig. 6 Pressures field in lubricant at 387 rpm - inferior oil

4.2 The experimental results

The experimental results using the accelerometer device give the effective values of the vibration acceleration and the spectral analysis for every kind of oil and the 387 rpm speed of the leading element.
Figures 7 and 8 present the same experimental determinations for the inferior and, respectively, for the superior oil. The effective value of the vibration acceleration is 106.56 m/s^2 for the inferior oil and 80.37 m/s^2 for the superior oil.

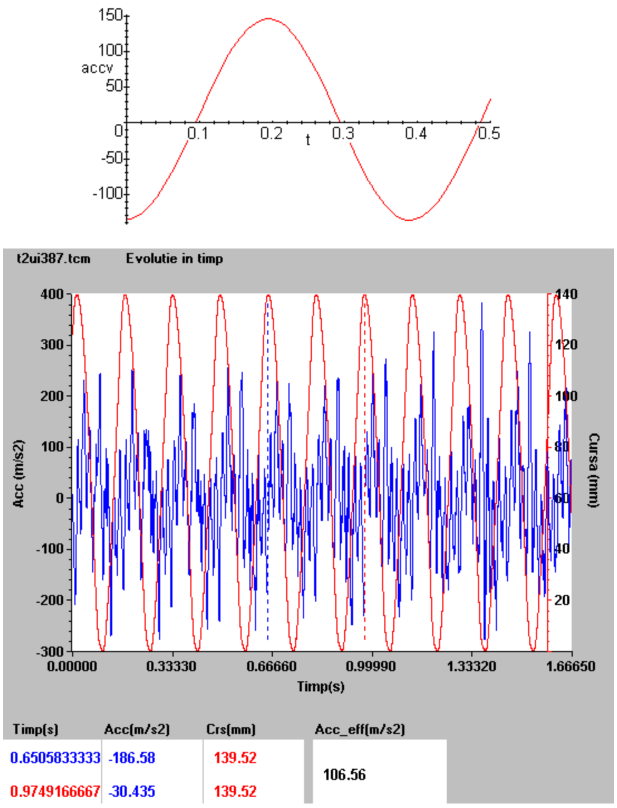

Fig. 7 The vibration acceleration for the inferior oil at 387 rpm

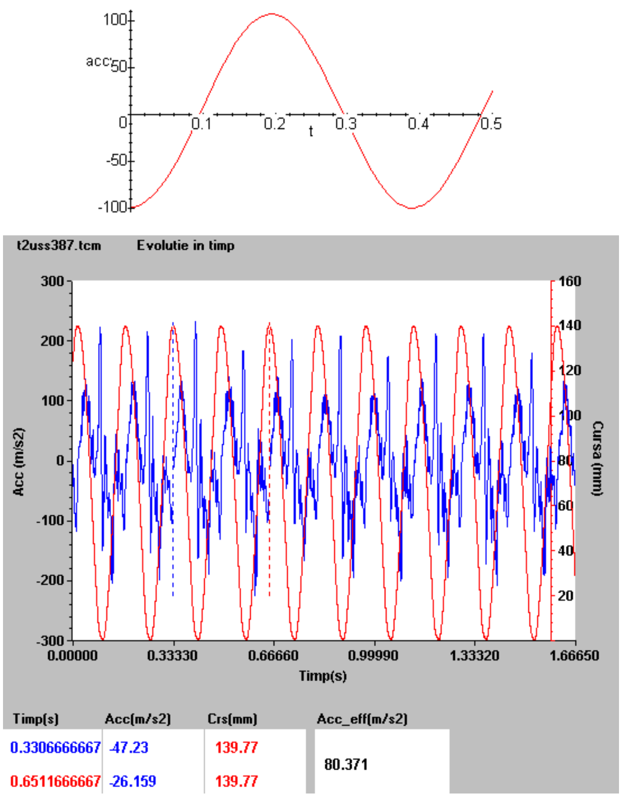

Fig. 8 The vibration acceleration for the superior oil at 387 rpm

The spectral analysis for the inferior oil and 387 rpm is presented in fig. 9 and fig. 10 presents the spectral analysis for the superior oil.

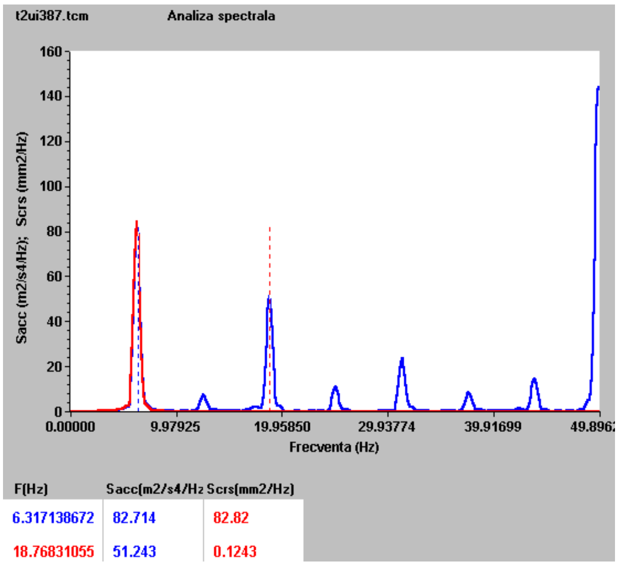

Fig. 9 The spectral analysis for the inferior oil at 387 rpm

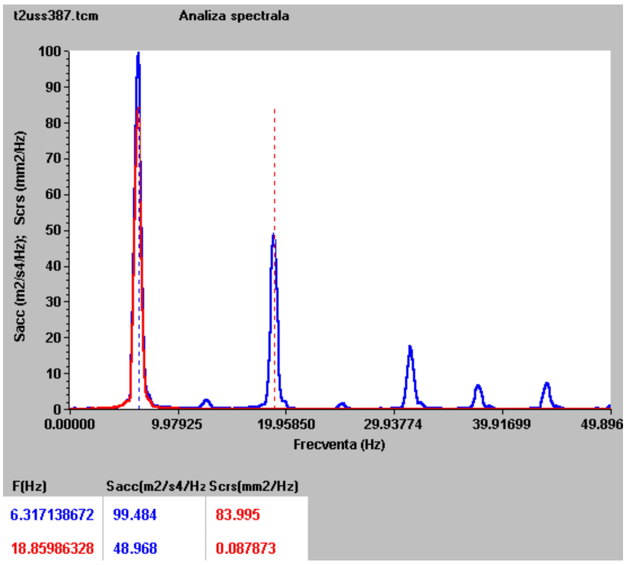

Fig. 10 The spectral analysis for the inferior oil at 387 rpm

The experimental results were compared to the ones given by the mathematical model solving. The compared characteristic quantity is the vibration acceleration and its values are shown in Table 2 for the inferior oil (TB32E).

Table 2 Comparison between the theoretical vibration acceleration for the inferior oil at 387 rpm

Rotation [rpm]	120	200	300	387
Theoretical Vibration Acceleration [m/s]	7.23	13.47	62.41	106.06
Experimental Vibration Acceleration [m/s]	7.66	13.301	63.61	106.56

The compared characteristic quantity is the vibration acceleration and its values are shown in Table 3 for the superior oil (SHELL TONNA T).

Table 3 Comparison between the theoretical vibration acceleration for the superior oil at 387 rpm

Rotation [rpm]	120	200	300	387
Theoretical Vibration Acceleration [m/s]	5.39	12.41	51.06	79.43
Experimental Vibration Acceleration [m/s]	5.59	12.732	51.44	80.37

The lubricant film thicknesses at the entrance, respectively at the exit of the rectilinear pair, are computed with the datum obtained by solving the mathematical model.

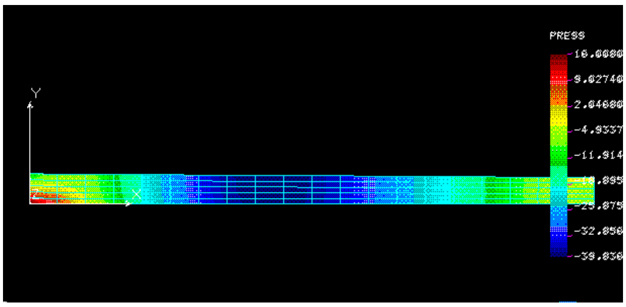

Fig. 11 Pressures distribution in inferior lubricant for 387 rpm rotation value

It was achieved a finite element routine COSMOS with this datum, considering the lubricant film as a plane laminar boundary layer [4].
In figure 11, it can be noticed that the pressure value decreases below 0; this means that cavitation phenomenon appears.
The comparison between the theoretical and experimental results for the values of the pressures field is revealed in the table 5.

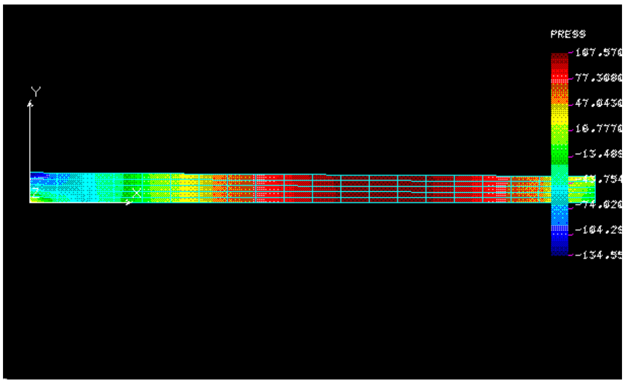

Fig. 12 Pressures distribution in superior lubricant for 387 rpm rotation value

Table 5 The comparison between the theoretical and experimental results for the values of the pressures field

Type of the lubricant N=387 rpm	Theoretical value for the pressure in lubricant [N/m^2]	The COSMOSM value for the pressure in lubricant [N/m^2]
Inferior	0.0160	0.016008
Superior	0.1100	0.10757

5 Conclusion

The influence of the lubricant from the rectilinear pair on the elastic cinematic element vibration was determined in the experimental way: the vibration amplitude varied from 0.005 m to 0.006 m for the oil with inferior characteristics and made decreasing mechanism working accuracy. Besides, phenomena like cavitation and lubricant film breaking appeared in the pressure field from the lubricant film and they may bring about gripping and deteriorating the accuracy of mechanism's elements movements.

ISBN: 978-960-6766-88-6

ISSN 1790-2769

The vibration amplitude varied from 0.0019 m to 0.003 m for the superior oil and the cavitation phenomenon didn't appear for the used range of speeds.

MAPLE routine has a great level of generality. It may be applied to any plane-moving element with a mobile lubricated rectilinear pair [5]. This routine computes the pressure field in lubricant and the maximal deformations of an elastic element with the accuracy less than 8%. It may be used as entrance variable quantity: material properties of the elastic elements, its dimensions, the lubricant viscosity characteristics and the length of the rectilinear pair, as well as the driving element rotation.

The behavior of the mechanism and its precision in work is so better so the oil is better as quality: the properties of lubrication and viscosity. The same situation is obtained when the speed is low. Knowing all these characteristics the deterioration of the elements of the mechanism can be avoided.

In the same time, the differences between the types of oil are low when the speed is low. When the speed of the leader element is high, the differences become higher.

The future directions of research include the material changing of the elastic element, using the larger and continuous interval for angular velocity on the leading element of mechanism for determination the exact moment of cavitations phenomena appearance.

References:

[1] Brown, H. – *507 Mechanical Movements: Mechanisms and Devices,* Dover Publications (August 15, 2005)

[2] Călbureanu, M., Malciu, R., Călbureanu, D., - *The Experimental Measurements Of The Vibrations For A Long Elastic Cinematic Element Influenced By The Operating Conditions And The Lubricant From The Mobile Rectilinear Pair*, International Conference "Advanced Concepts in Mechanical Engineering" ACME IASI 2004; Buletinul Ştiinţific de Iaşi, pag. 107-112

[3] Homer, Eckhardt - *Kinematic Design of Machines and Mechanisms,* McGraw-Hill Professional; 1 edition, 1998

[4] Erdman, J. Holte, T. Chase - *Approximate Velocities in Mixed Exact-Approximate Position Synthesis of Planar Mechanisms,"* ASME Journal of Mechanical Design, Vol. 123, pp. 388-394, Sept. 2001.

[5] Wang Z., Ji S., Wan Y., Sun J., Ou C., Pan Y. - *Kinematic Analysis and Dexterity of a New STPM* - Proceedings of the 7th WSEAS International Conference on Robotics, Control & Manufacturing Technology, Hangzhou, China, April 15-17, 2007, p. 316

[6] White, M. Fr. - *Viscous Fluid Flo*w, McGraw Hill, 3 edition, 2005

[7] Zienkiewicz, O., C., Taylor, R.L., Nithiarasu P. - *The Finite Element Method for Fluid Dynamics,* Sixth Edition, McGraw Hill, 2005