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IDENTIFICATION OF THE INTERFACE OF LATERAL SLIDE BEARINGS

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ABSTRACT:
In the paper, the model of slide bearing of piston engine is presented, together with methodology of research on this type of bearings and its results. The measurement standpoint consists of main elements of Briggs & Stratton 550 serie 10 T 802 model piston engine, driven by three-phase electric motor. On the aforementioned measurement standpoint specific measurements were done, needed for specification of proper performance of crank slide bearings, operating on smooth friction. The bearing was stressed with variable dynamic force of constant value and changeable direction. The force caused a change in dislocation of a plug versus the bearing, and so the change in thickness of oil film and alternation of lubrication process. The aim of the paper was to conduct preliminary research, in order to confirm the fact that fixing the moment of transition from smooth friction to boundary friction in crank bearing, responsible for deterioration of bearing's performance, is possible.

Keywords: *vibrations, performance, energy impulse, crank bearing*

1. INTRODUCTION

During the perfomance of piston engine, part of thermal energy generated during combustion of fuel is converted to mechanical energy - in the form of work. In this energy transformation process, crank bearings take active part. Performance of any crank slide bearing [5] can be interpreted as such transfer of energy in specific time, which enables the shift of sliding - reversible movement of the piston to rotating movement of crank shaft in specifed conditions and time t, when the work $L_{eŁ}$ is being done by the bearing. Therefore, the above mentioned can be interpreted as [4,6]:

$$D_{Ł}(t)=L_{eŁ}(t)\cdot t \tag{1}$$

In case of $L_{eŁ}$=idem in time t, below relation is correct:

$$D_{Ł}(t)=L_{eŁ}\cdot t \tag{2}$$

where:
$L_{eŁ}$– work done by the bearing;

t – time of bearing's performance.

Crank slide bearing is the friction node consisting of:

– crankpin as the element stressed with variable dynamic forces;
– bearing, as the element in relation to which the plug is moving;
– lubrication oil, separating the bearing and the crankpin.

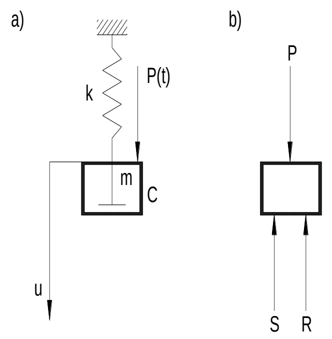
Between the mentioned elements, specifically determined relations can be observed, which will be subjected to distinct changes together with f.i. changes of friction coefficient, depending on the mode and lubrication process. Such changes are visible during the transition from smooth friction to surface friction, what causes the change in quality of crank slide bearing's performance (1,2).

The paper covers the possibility of diagnosing the bearing's performance by registering the changes in friction process in slide bearing, caused f.i. by the changes in lubrication oil flow and not balancing the bearing as friction node.

2. PHYSICAL MODEL OF CRANK SLIDE BEARINGS

For the purposes of analysis of crank slide bearings, possibly simplest physical model was built. In the model, the conditions were created for bearing's performance in smooth friction, so that reliable information on momentary transitions of lubrications from smooth to boundary one in possesed.

The simplest physical model of crank slide bearing of piston engine is the system of one degree of freedom (pic.1) that can be perceived as adequate for entire class of crank slide bearings, whose attributes can be defined by four paramaters: *m* – mass, *k* – elasticity coefficent, *c*– damping factor , *u* – temporary dislocation of the crankpin.



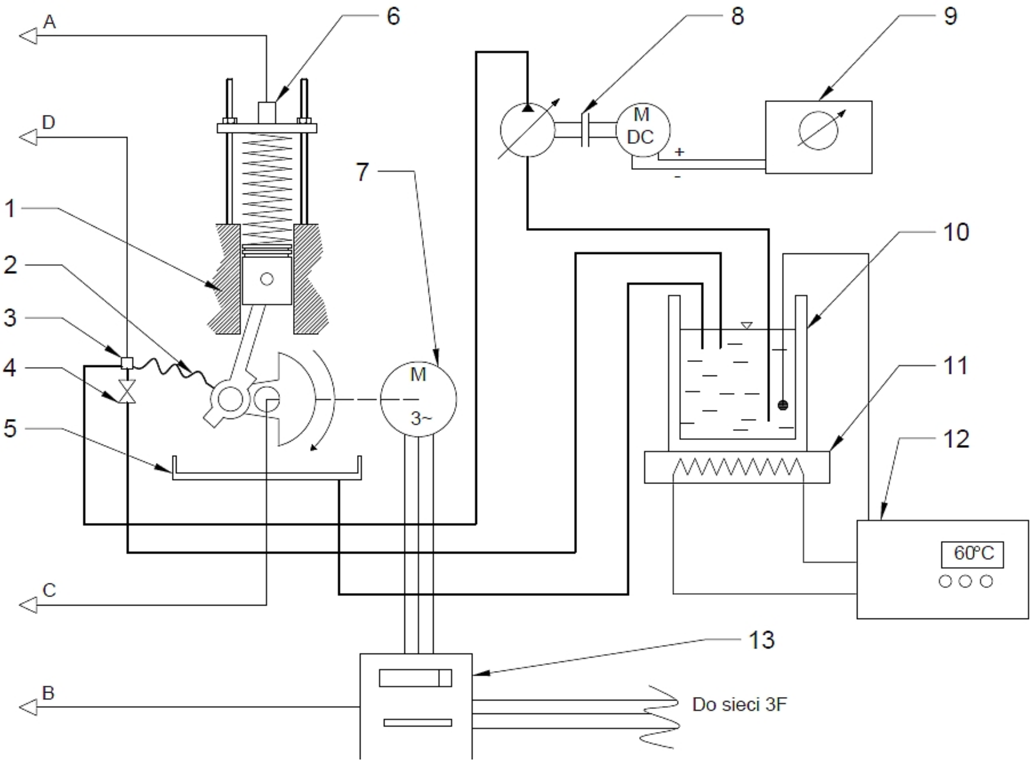
Pic.1 Physical model of crank slide bearings
m – mass of crank bearing, k – elasticity factor, c – oil damping factor, P(t) – connecting rod force,
S – damping force, R – the reaction of elastic ties, u – displacement

Dislocation of vibration of the crankpin u(t) versus the bearing was measured during the bearing's operation in the area of smooth friction or mixed one, in which also the boundary friction appears. During the measurement of vibrations, the crank bearing, except for the forces presented on pic. 1B, was subjected to: load force, inertial force and interferencing forces.

3. RESEARCH METHODOLOGY

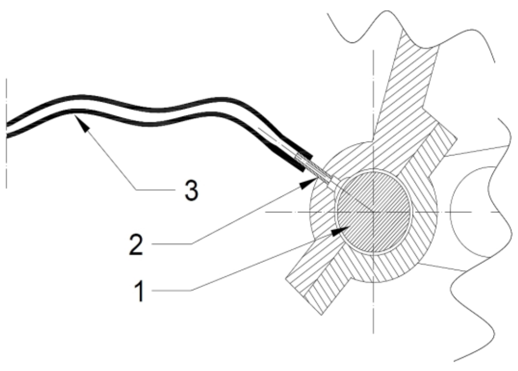
The measurement standpoint (pic.2) was built on the bed of turning lathe TSA 16. For the purpose of the research, the turning lathe was revoked from the support and its drive.

Regulation of angular velocity of piston engine's crankshaft is realized by the change of gear ratio, transferring the drive from the electric motor.
The measurement standpoint is equipped with asynchronic motor of 0,4 kW, enabling the change of rotations' direction.



Pic.2. Scheme of measurement standpoint
1 - engine housing; 2 – elastic oil hose; 3 - temperature sensor; 4 – three-way oil valve; 5 – oil pan; 6 - accelerometer; 7 – three phase induction electric motor; 8 - electric oil pump; 9 - DC PWM controller; 10 - oil tank; 11 - electric heater; 12 - electronic thermostat; 13 - three-phase electromechanical induction meter. Signals: A - vibrations; B - electricity consumption; C - speed / TDC; D - oil temperature

The object for the measurements is crank bearing of piston motor engine Briggs&Stratton series 550 model 10T 802, in which the mode of lubrication named Orlen Platinum Mineral 15 W-40 has been modified. Splash lubrication forseen by the producer has been replaced by permanent supply of oil to the bearing by the elastic hydraulic hose of diameter ϕ4mm. In the lubricating opening placed in the crankshaft, the connection terminal was fit in, connected to the above mentioned crankshaft (pic.3). Such solution resulted in the necessity of cutting out the opening in the frame of the motor, to install the pipe supplying the lubricating oil from the outiside.



Pic.3. Detailed lubrication system of crank bearing.
1– craknpin; 2–hose barb; 3–elastic oil hose

Possibility of control of bearing's workload was obtained by removing the head of the motor and replacing it by the steel plate, being the support for the helical spring leaning against the bottom of the piston and transmitting the vibrations to vibrations' sensor. The spring leaning against the bottom of the piston moves the force acting on the piston. The force acting on the piston can be regulated by straining the spring with the use of the steel plate, placed in the four threaded rods located in the frame of the motor. One rotation of crankshaft falls on the maximum workload of the piston, what reflects the operation of the two-stroke motor. Main deviation from original lubrication mode of crank slide bearing is installing of the elastic crankshaft (pic.3), what resulted in other lubrication mode ensuring smooth friction during the study. Operation of crank bearing, represented by dependencies (1), was diagnosed on the research standpoint consisting of:

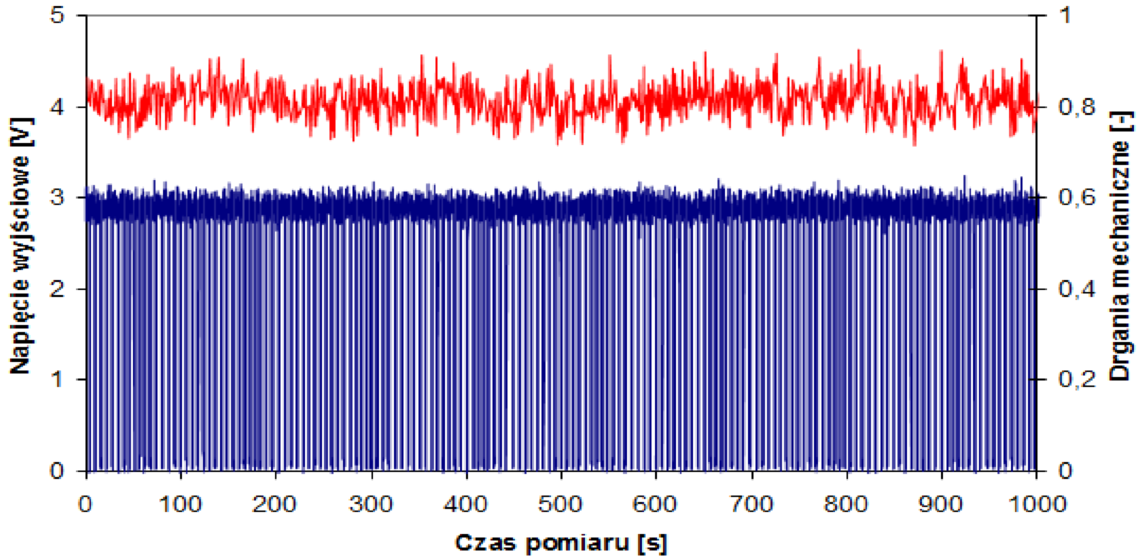
1. Vibrations sensor Bruel&Kjaer 4370V, analysing the action of friction forces in the form of random, succeeding impulses of unpredictable values, low intensity and durability. Measurements were done according to recommendations of producer of the apparatus T-03 [9].
2. Three-phase electric energy meter of type 4C52Z, which was modified for automatization of the measurements. Modification was based on dividing the face of the meter into 20 equal parts (without interfering in the face structure) by sticking the optical markers. Counting of the revolutions was realised by reflective transoptor located above the surface of the face. The signal from the transoptor was formed to TTL level and transmitted to measurement card in the form of impulses. Construction of the meter ensures 120 rev/1kWh. At set scale of the face, the measurement resolution of 0,000417 kWh/impulse was obtained.
3. Electric oil pump of type SL-001, of 12 V voltage and maximum power of 0,2kW. Oil circulation in the measurement standpoint was closed and forced by the operation of the pump. Taking into account too high capacity of the pump, it was necessary to connect it to the regulated power supply in the form of PWM regulator, what enabled the reduction of pump's consumption and its localization.
4. Temperature sensor of type LM35 located in the three-way valve casing, which controls temperature of the oil supplied to the bearing. Voltage output of the temperature sensor was connected to measurement card.
5. Data Translation measurement card of the type DT9816, to register the energy consumption, oil temperature and rotational velocity (simultaneously GMP signal) at sampling frequency of 60 Hz.

Before the start of the measurement, the oil was heated by laminated electric heater of 1,5 kW power, to the temperature of 85ºC. The temperature was mantained by electronic thermostat. After reaching the required temperature, the oil pump was activated and simultaneously the three-way valve was open and placed on the casing of the motor. Opening of the valve triggered the return (reversion) of heated oil to the container, what enabled further mixing and heating up of the oil. During the measurements, the valve is not completely closed, so that part of the oil is not supplied to the bearing, but returns back to the container. Such”short” circulation enables the exchange of the oil several times, maintaining constant temperature, what is especially important in case of low intensity of flow for the bearing itself. Taking into account the modification of motor's frame (cutting out considerable opening for the hydraulic pipe), it was necessary to dismantle the outer dry oil pan, from which all leaks of lubrication oil were flowing back to the container.

4. RESULTS OF STUDY

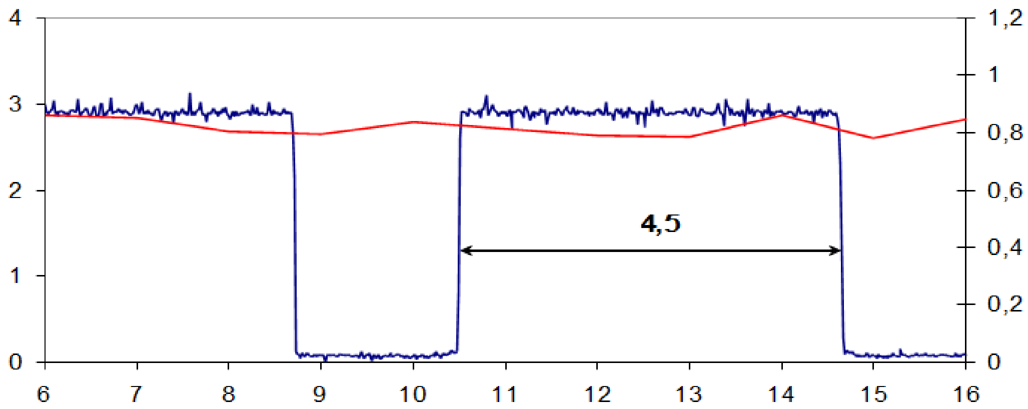
As parameters identyfing the moment of disappearance of smooth friction in the bearing and appearance of boundary friction, the change in vibrations amplitude and the time of measurement impulse reflecting the energy level were assumed. Measurements consisted of four stages, lasting 1000 seconds each, if referential level f vibrations informing on the appearance of boundary friction was exceeded. The stages were defned as A,B,C,D, and E.

A. Right rotational direction of crankshaft with the crank bearing supplied with lubricating oil of 2,6l/min capacity.
The course of the operation lasting 1000s (several hundred energy impulses) is presented in the pic.4.



Pic.4 Change of course of voltage of measure impulses and vibration's level charasteristic for operation of side bearing for stage A

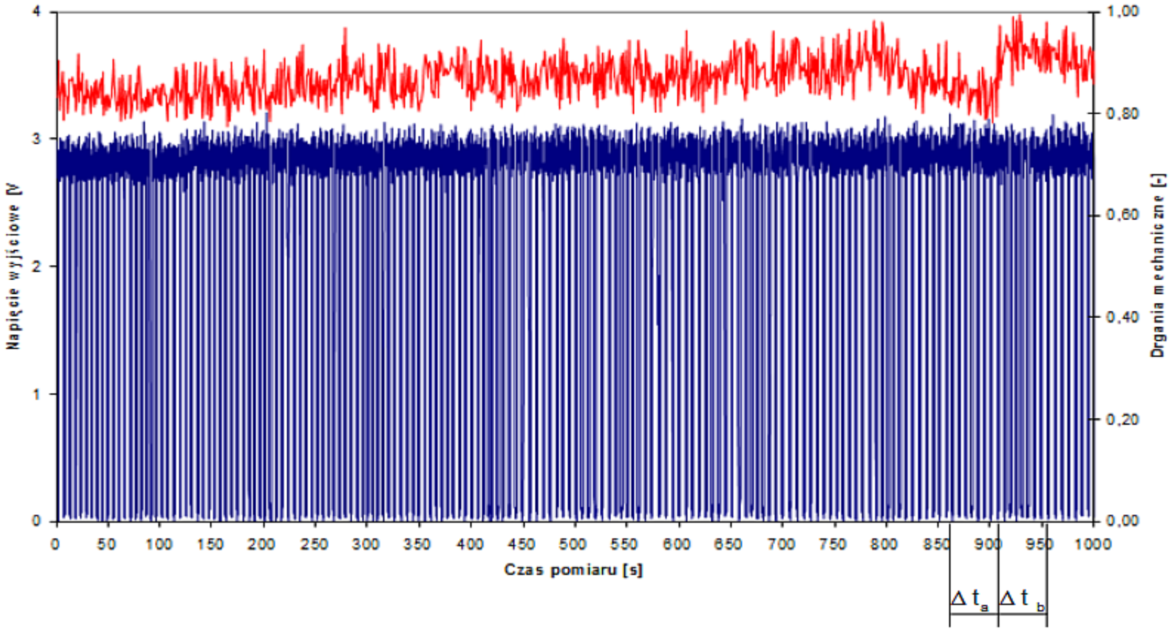
As the change of duration of impulses and vibrations' level is almost the same, it was assumed that duration of the impulse is representative and sufficient for diagnostic purposes for stage A – pic. 4A.



Pic.4A Duration of one energy impulse

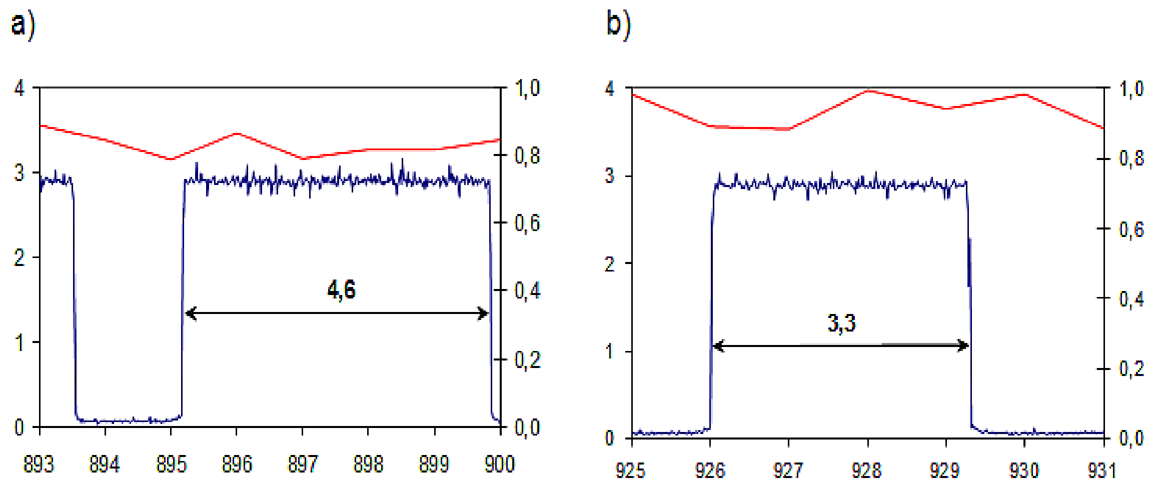
B. Right rotational direction of crankshaft with crank bearing supplied with lubricating oil of 1,3l/min capacity.

The course of operation lasting for 1000s (several hundred impulses of energy) is presented in the pic.5



Pic.5 Change of course of voltage of measure impulses and vibration's level charasteristic for operation of side bearing for stage B

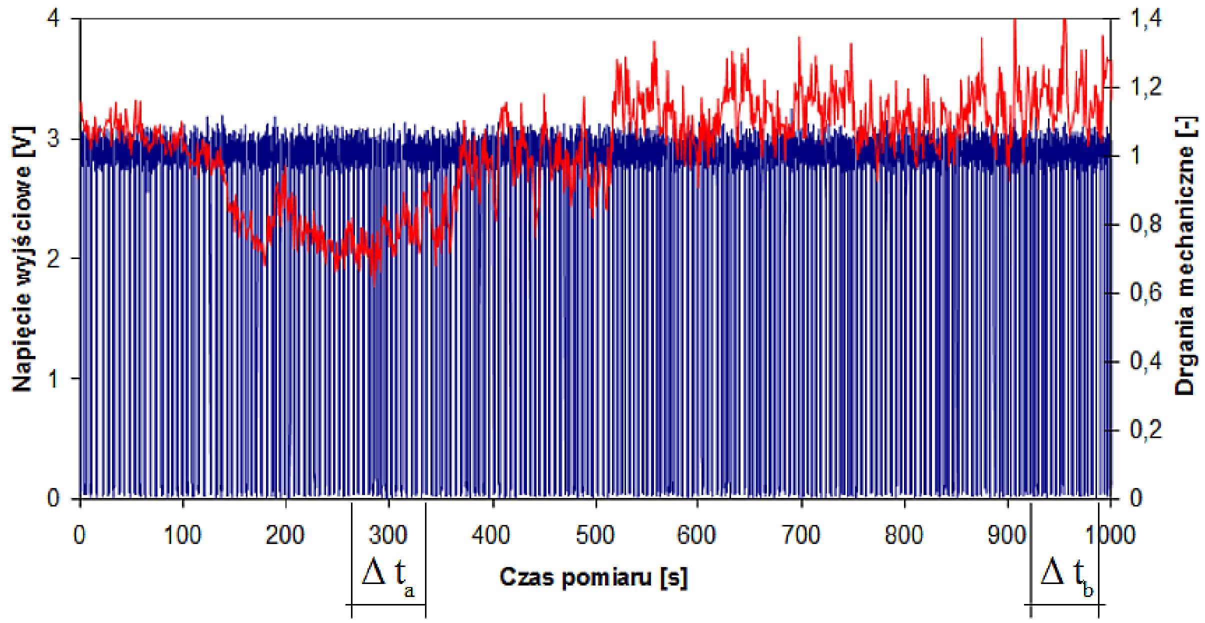
From the above graph, it results that in time range $\Delta t_a \in (870 \div 905)$ a decrease in vibrations level occured and in time range $\Delta t_b \in (910 \div 930)$ maximum increase of vibrations level could be detected. Duration of energy impulses in these time ranges are presented in the pic. 6.



Pic.6a)Duration of energy impulse at low vibrations level b)Duration of energy impulse at high vibrations level

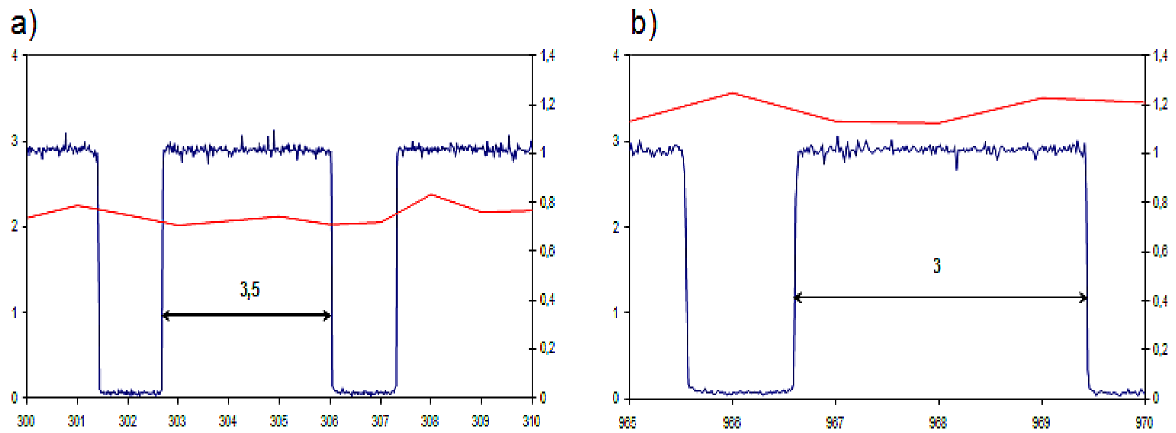
C. Left rotational direction of crankshaft with crank bearing supplied with lubrication oil of 2,6l/min capacity.

The course of operation lasting for 1000s (several hundred energy impulses) is presented below in the pic.7.



Pic.7. Change of course of voltage of measure impulses and vibration's level charasteristic for operation of side bearing for stage C

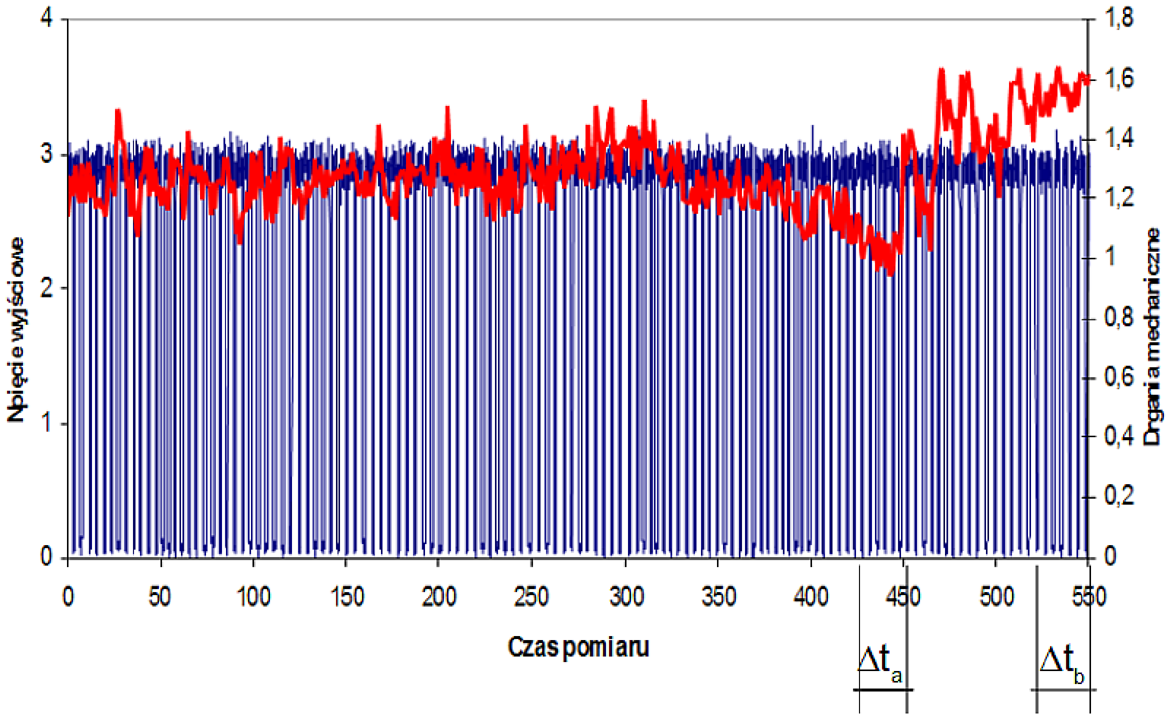
From the above graph, it results that in time range $\Delta t_a \in (280 \div 310)$ a decrease in vibrations level occured, and in time range $\Delta ta \in (920 \div 980)$ maximum vibrations level could be detected. Duration of energy impulse in these time ranges is presented in pic. 8.



Pic.8a)Duration of energy impulse at low vibrations level b)Duration of energy impulse at high vibrations level

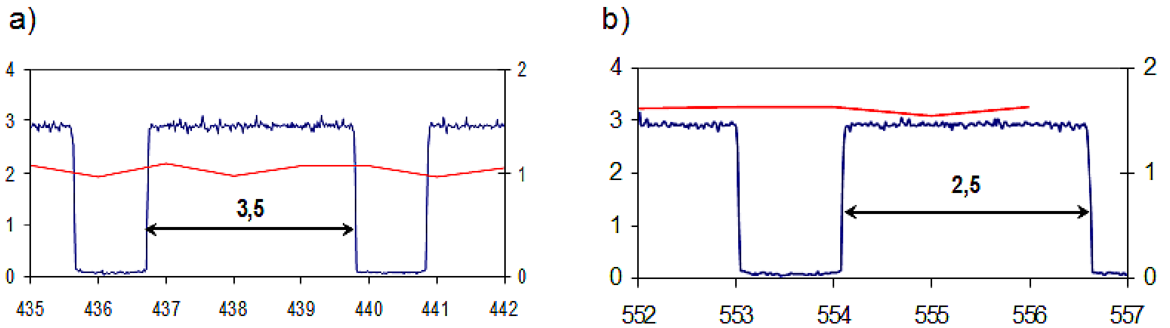
D. Left rotational direction of crankshaft with crank bearing supplied with lubrication oil of 1,3l/min capacity.

The course of operation lasting for 550s, as the referential level had been exceeded, was presented below (pic.9).



Pic.9. Change of course of voltage of measure impulses and vibration's level charasteristic for operation of side bearing for stage D

From the above graph, it results that in time range $\Delta t_a \in (430\div450)$ a decrease in vibrations level ocured, and in time range $\Delta t_b \in (530\div550)$ a referential level of vibrations was exceeded. Durations of energy impulses in these ranges are presented in the pic. 10.



Pic.10 a)Duration of energy impulse at low vibrations level b)Duration of energy impulse at high vibrations level

From the above graphs, it results that thickness of oil film is related to vibration level and duration of energy impulse. From the pic 5a,5b,6a.6b,7a and 7b, it results that with the decrease of oil film vibrations level increases and duration of energy impulse decreases. Thus, it is possible to identify boundary friction that is accompanied by increase of vibrations level abve the referential level and shortnen duration of energy impulse. Operation of slide bearing is then considered as described by formula(1) and can be understood as duration of energy impulse. The longer the impulse, the higher the amplitude level of the bearing, what means that possible amplitude of cross slide bearing is higher than required.

5. FINAL REMARKS AND CONCLUSIONS

All measurements were done for two different capacities of lubrication oil. From theoretical calculations, not taking into consideration lubrication mode described in[8], it resulted that smooth friction was mantained. During the measurements, intensive leakage of lubrication oil at left rotational direction of crankshaft could be observed. It was caused by inappropriate mode of lubrication and thus most probably smooth friction did not occur. The analysis of obtained results leads to the following conclusions:

- Increase of vibroactivity of slide bearing contributes to increase of number of energy impulses in time, so the value of operation increases (1). If this value exceeds permissible value, the bearing should be considered as non - operational.
- Described experimental identification of phenomena, occurances and vibroacoustic processes can be used for identification of the moment of appearance of boundary friction in slide bearing. It is necessary though to elaborate more adequate physical model of analysed slide bearing, so that the results could be adequate for real crank slide bearings.
- Single-cylinder motor used in measurement has too loose elements of crank-piston system, what results in visible and considerable leakage of lubrication oil. The leakage overvalues experimentally fixed flow of oil to the bearing.
- Research methodology proposed by the authors should be treated as introduction to further research of considered object.

For further considerations, it is necessary to improve the measurement standpoint and fixing the required referential level for vibrations' sensor and boundary duration of energy impulses.

- The research is preliminary one, and it is planned to elaborate extended methodology of experiment, which should contribute to elaboration of formulas enabling to set the considered operation as a formula.
- Results obtained are promising regarding the possibility of fixing the time t (1).

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