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PHASE TRAJECTORY RUN AS A USEFUL TOOL IN AXIAL-PISTON PUMP WEAR DETECTION

SUMMARY

In this paper the possibility of phase trajectory use in detecting of axial piston pump damage was presented. The wear of main part of pump elements such as: rotor and swash plate have been studied. The Phase trajectories were estimated from vibration signal measured on pumps body in third directions. In order to get quantitative specification of analyzed trajectory At_{pi} *parameter was introduced. At the and the relation between Atpi parameter and the wear of the pumps parts have been set.*

***Keywords**: positive-displacement pump, signal analysis, phase trajectory, diagnostic of machines*

WYKORZYSTANIE PRZEBIEGU TRAJEKTORII FAZOWEJ DO WYKRYWANIA ZUŻYCIA POMPY TŁOKOWEJ

W artykule przedstawiono możliwość zastosowania metody trajektorii fazowych w procesie oceny stanu zużycia pompy wyporowej, na przykładzie wypracowania wybranych elementów składowych pompy: tarczy wychylnej i zespołu wirnika. Wykorzystano do tego celu przebiegi trajektorii fazowych, które obliczono z pomierzonych na korpusie pompy sygnałów wibracji w przypadku wyposażenia jej w uszkodzone elementy składowe. Dokonując parametryzacji wyliczonych trajektorii otrzymano ich ilościowe miary w postaci bezwymiarowych współczynników At_{pi}*, które powiązano z wypracowaniem zespołu pompy.*

***Słowa kluczowe**: pompa wyporowa, trajektoria fazowa, analiza sygnału, diagnostyka maszyn*

1. INTRODUTION

In hydraulic drive systems the positive-displacement pumps are one of the important elements. Proper work of these elements causes proper work of whole hydraulic system. The wear of pump elements causes pump's pressure lost and increase of volumetric losses, which leads to decreasing delivery of the pump and increasing in vibration and noise of its work. The wear of pump elements is mainly caused by forces between their kinetic pairs (e.g. piston-cylinder, valve plate – rotor, piston shoe – swash plate) which interacts. The next reason of pump's elements wear are improper operating conditions (Murrenhoff and Scharft 2006) such as:

- exceeds of operating pressure level,
- low oil viscosity,
- insufficient oil filtration.

The most common type of wear elements in axial piston pump is their abrasive wear. Elaboration of that type appears in all components of the pump, between which there is relative and contact motion. In this article the use of phase trajectory method as an effective tool in assessing wear of positive displacement pump is considered. Abrasive wear of rotor and swash plate is an example of pumps wear described in this paper.

2. DESCRIPTION OF THE TESTED OBJECT

Multi-piston axial positive-displacement pump of constant efficiency was a subject of the examinations. Operation of such pump (Fig. 1) is based on distribution of the fluid flow to and from rotor (2) cylinders via stable valve plate (6), which is perpendicular to axis of rotation of the pump cylinder. During rotational movement, rotor face co-operates with valve plate (6), sliding along its surface. Two kidney-shaped holes are drilled in valve plate (6). Suction hole is separated from compression hole with transition zone, so called bridge.

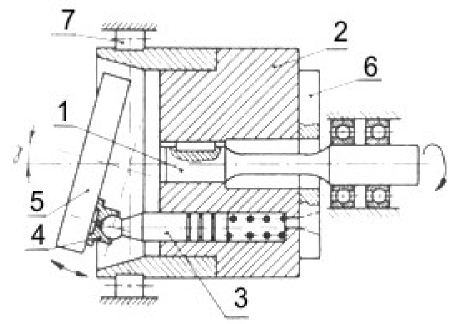

Fig. 1. General scheme of axial multi-piston pump: 1 – shaft, 2 – rotor, 3 – piston, 4 – piston shoe, 5 – swash plate, 6 – valve plate, 7 – bearing (Stryczek 1995)

In axial piston pump the rotor unit is usually their most important item directly decisive of pump outlet flow generation. Existing construction of pumps are differ in rotor unit way location and its drive. The main rotor unit of pump used in tests was composed of seven pistons which were located in rotor cylinders. The whole rotor unit is supported by the roller bearing (Fig. 2). The abrasive wear of rotor is

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mainly due to the forces that occur on surfaces in piston-cylinder kinematic pairs. Piston load is sum of radial forces of swash plate – piston shoe reaction and inertia force of rotor unit which rotates with required speed. Excessive load of rotor leads to abrasive wear on its components and radial clearance enlargement in piston-cylinder kinematic pairs. This increase in volumetric losses and decrease in pumps overall efficiency.

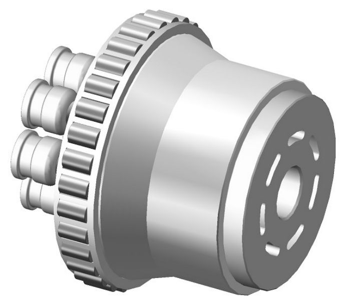

Fig. 2. View of pump's rotor unit

The task of swash plate (which cooperate with the rotor) is ensuring the appropriate setting of the pump performance by changing the angle between axis plate and driving shaft.

Change in swash plate angle has effect on the length of the piston stroke in rotor's cylinder and thus has an impact on volume flow in pump working cycle.

Physically the swash plate works with surfaces of pistons shoes which slides on its surface. Cooperation of these elements is burden with friction losses on their surfaces. Minimizing these losses, (thus increasing the mechanical efficiency of the pump) provides hydrostatic support between swash plate and pistons shoes surfaces.

In the case of that support disappears following progressive wear on surface plate and pistons shoes appears. Finally it lids to complete degradation of kinematic pair (Fig. 3).

In laboratory research conducted by the author wear of examined pumps elements concern to:

- 10 μm radial clearance enlargement in piston-cylinder kinematic pairs,
- rotor bearing race abrasive wear,
- elliptical deepening of swash plate surface.

Measurements of pump wear elements have been done by optical surface measure method (Kowal and Sioma 2009). Measured values of elliptic deep on swash plate surface in seven directions (Fig. 4) ranged from 0.1 mm per direction 1 to 0.5 mm at the direction 7 (Fig. 5).

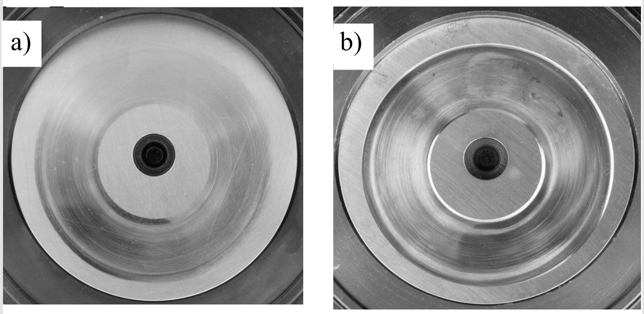

Fig. 3. The view of swash plate surface condition: a) for functional plate – without wear; b) for out of order surface plate

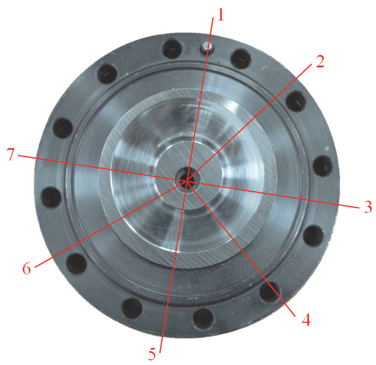

Fig. 4. Adopted guidelines for measuring the deepening on swash plate surface

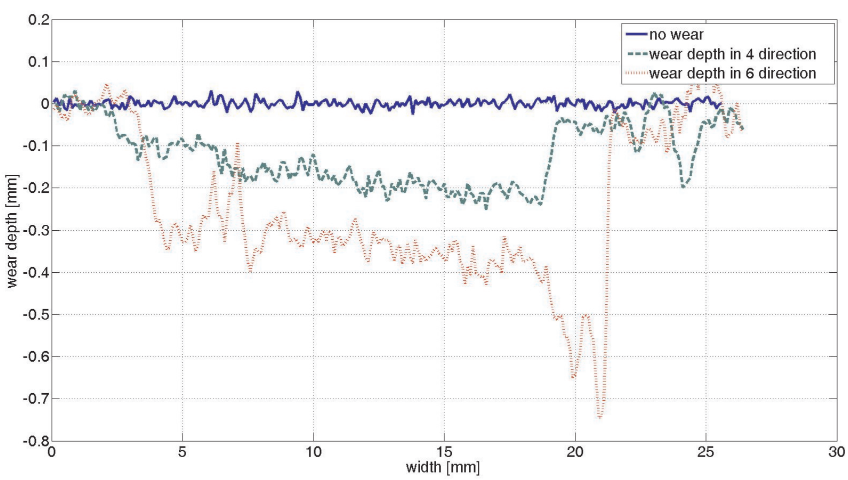

Fig 5. Comparison of swash plate wear depth for selected measuring guidelines

3. TESTING STAND AND MEASUREMENT DESCRIPTION

Scheme of the laboratory stand used for testing is shown in Figure 6. The stand comprise: pump P with worn elements, set of hydraulic elements (servo-valve, throttle valve, cut-off valve and set of filters) with transducers of tested values of: flow rate, static and dynamic pressure, and transducers used for measurement of acceleration pumps body vibration.

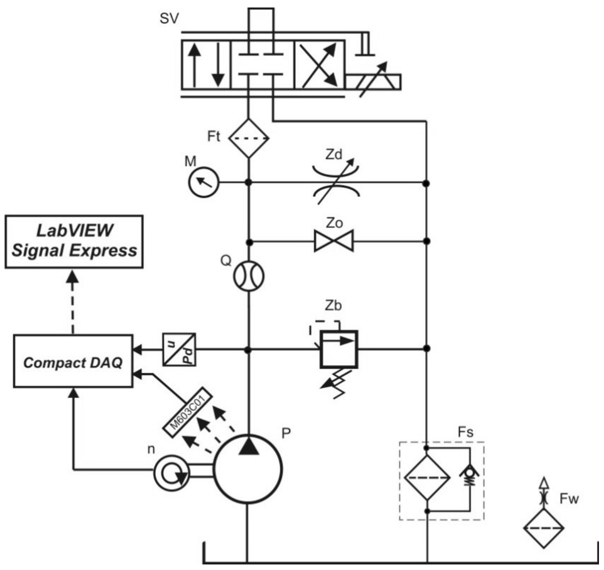

Fig. 6. Hydraulic scheme of experimental stand:
P – pump with worn rotor and swash plate, SV – servo-valve, Zb – maximal valve, Zd – throttle valve, Zo – cut-off valve, Ft – pressure filter, Fs – low-pressure filter, Fw – inlet filter, M – manometer, n – tachometer, Q – flow-meter

Examinations of the influence of worn elements on the positive-displacement pump operation were conducted via changing worn swash plate and rotor unit in the pump body. Additionally for comparative purposes, examinations of new pump with not worn elements have been executed. Measurements of physical values (vibration, flow, static and dynamic pressure) have been executed in the pump operation in following conditions:

- no pressure at the pump outlet,
- loading of the pump outlet with static pressure of 70 bar,
- loading with dynamic, periodically variable pressure.

Measurements of the pump body acceleration were executed for three measurement axes (*X*, *Y*, *Z*). After previous mounting of vibration transducers on the pump body, near rotor unit 16-bit measurement cards, co-operating with conditioning system and programmed by Signal Express in LabView program, have been applied in measurements of physical quantities. Measured courses were stored on computer disc, and then they were tested by numerical analysis with use of Matlab-Simulink program. General view of the pump with mounted vibration transducers is shown below (Fig. 7).

4. APLICATION OF PHASE TRAJECTORIES METHOD

Classical methods of vibroacoustics machinery diagnostic (Cempel 1992; Żółtowski and Cempel 2004) are based on time or time-frequency signal analysis. In order to proper interpret such results, same experience in signal processing is required. This is particularly important for analyzing results of vibroacoustics processes from machinery and equipment with high complexity, where movements of machinery components overlaps. In addition, often correct interpretation received from time or time – frequency distributions is difficult to interpret by parasitical components (disturbances), occurring from items (machines) not related to the examined object (Zieliński 2009).

Example of obtained power spectral density PSD (Lyons 2006) representations of measured acceleration vibration signals for pomp without damage and with wear elements (measured for each axis separately) have been shown below.

As we can see from Figures 8 to 10 the result of pump condition estimated by PSD diagrams is difficult to evaluate (the way how to estimate obtained PSD runs can be find in another authors publications (Stojek 2010a, b)).

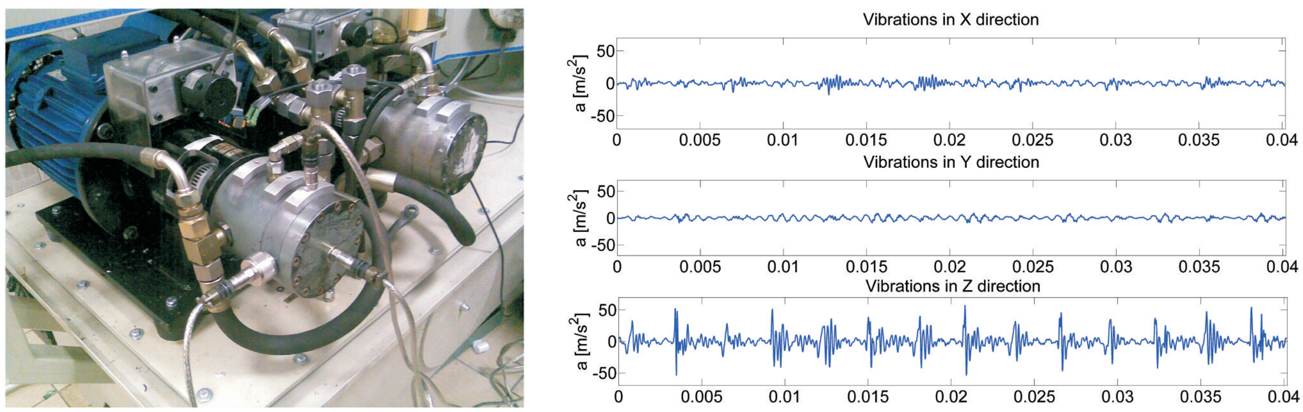

Fig. 7. Distribution of measurement transducers on the tested pump body, and example of acceleration vibration courses for three measurement axes

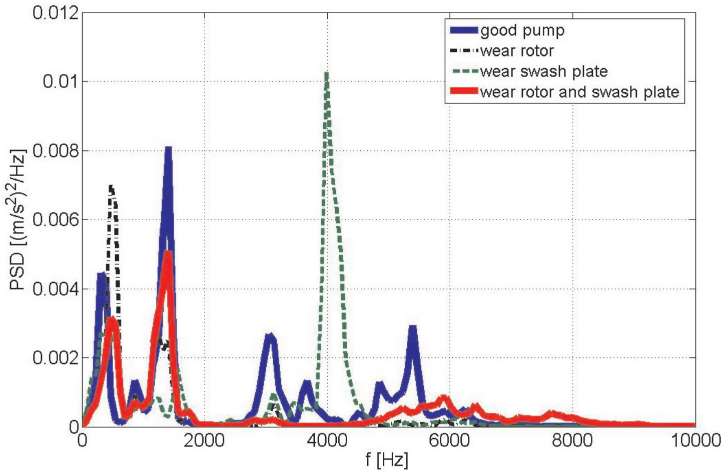

Fig. 8. Power spectral density PSD representation of signals measured on pump body along *X* axis

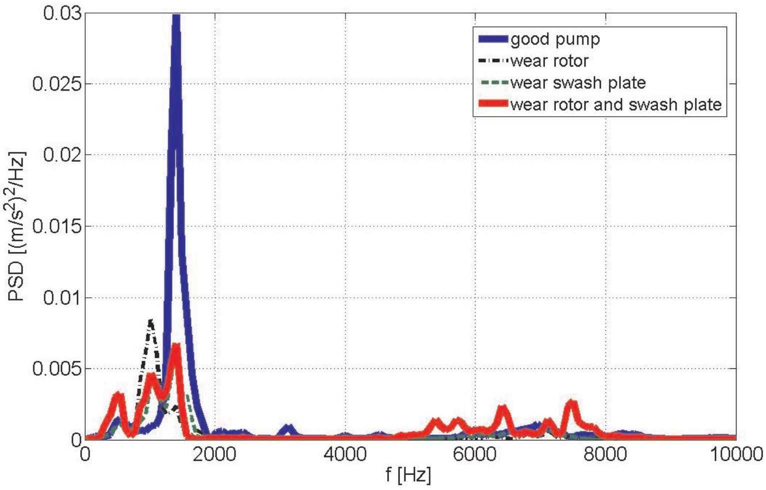

Fig. 9. Power spectral density PSD representation of signals measured on pump body along *Y* axis

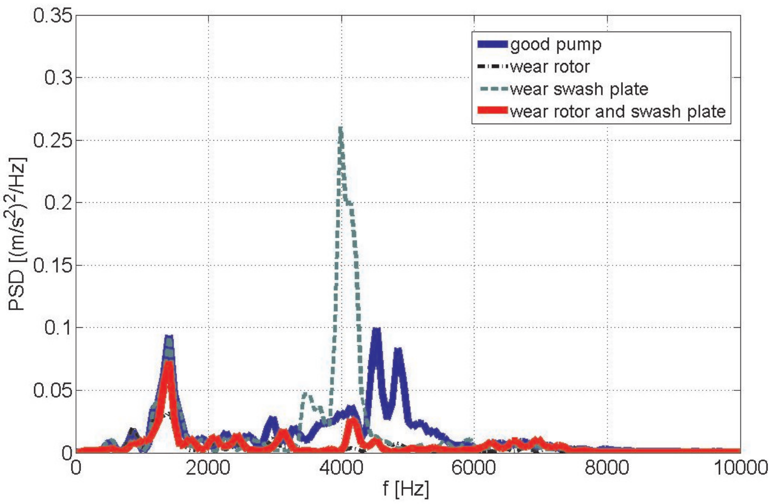

Fig. 10. Power spectral density PSD representation of signals measured on pump body along *Z* axis

Searching for methods, which results will allow to faster and easier interpretation of the pump condition it has been proposed the phase trajectory runs estimation. The concept of phase trajectory is related in automatics with the concept of stability in terms of Lapunov and technical stability for test object, device or process (Bogusz 1972).

Let's assume that certain element of the pump is working on forces which come from other elements or object.

Motion equation of testing element can be formulated as:

$$\ddot{x} = F\left(\dot{x}, x, t\right) \tag{1}$$

where:

- $\ddot{x}$ – acceleration of tested element,
- $\dot{x}$ – velocity of tested element,
- x – displacement of tested element,
- t – time.

This equation has unique solution determined by initial conditions. Taking into account environment reaction of testing element written as disturbance equation „R" constantly operated:

$$\ddot{x} = F\left(\dot{x}, x, t\right) + R\left(\dot{x}, x, t\right) \tag{2}$$

The solution of this equation is carry out by substitution and reduction of the order of equation which lids to:

$$\dot{x} = f\left(x, t\right) + r\left(x, t\right) \tag{3}$$

Taking into consideration that $R(x, t)$ function consider permissible deviations from steady-state conditions, the change in initial conditions and predicted external and inner disturbance which operate on system in random character or periodic character, dynamic conditions of object thanks to technical stability conception can be determined (Bogusz 1972).

Proposed approach doesn't require full identification of system structure that is strict determination of function $f(x, t)$ and we may concentrate only on equation (3) solutions. Effective tool for research solution of system of differential equations is trajectory analysis in phase space. From definition of technical stability system arise, that for initial condition included in ω area of phase space the solutions of system (3) remain in Ω area. So system is technically static (Fig. 11).

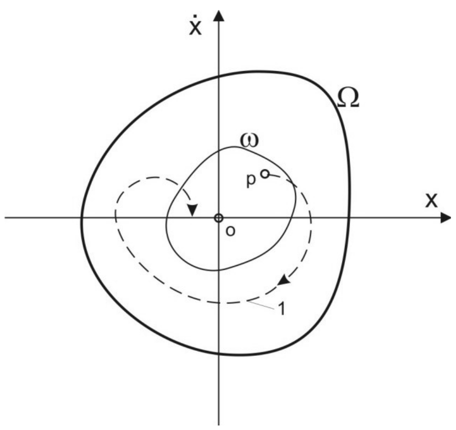

Fig. 11. The concept of technical stability: 1 – phase trajectory, Ω – area of permissible deviation of equilibrium state, ω – area of initial conditions, p – initial state of system after out of balance „o"

Testing real object such as hydraulic axial piston pump we deal with many elements which interact each other. Stability of that system can be determined by motion parameter measured from particular element. Physically we may only base on elements which are connected with housing of pump.

Vibration parameters of the housing are connected with motion of pump's components. After selecting proper place on housing of the pump which is geometrically connected with testing elements. The displacement and velocity of vibration measured in selected point related to phase space will characterize a class of solution of partial equation connected with particular elements of pump. The problem how to define Ω area can be solving in many way. In case of axial-piston pump it will be determination of Ω on the base of dynamics analysis of pump which define as good pump. Testing of phase trajectory for good pump with consideration of outer noise will allow trajectory determination.

Theoretical consideration of this problem lids to conclusion that there is no knowledge how the area Ω (which is technically static) will change yourself as a result of degradation of testing element or his surroundings. Phase-space includes information about kinetics and potential energy. From machine model made as energy processor comes that in the case of total energy increase of testing element, the surface of area Ω will increase. This statement enables energy structure identification of testing object. Observation of elements which are connected with technological process (pumping) in case of energetic efficiency decreases it is possible to expect the area Ω decrease. For elements, where destructive coupling of dissipation energy occurred, the area Ω should increase.

4.1. Analysis of received phase trajectories

Measured vibrations signals from pumps body have been put on numerical analysis. The next step was determination of phase trajectories. The methodology of phase trajectories estimation used so as to detect the wear of pump elements (rotor unit and swash plate) was based on integration of numerical acceleration's runs, measured in assumed points of pump's body in assumed direction.

Real shape of phase trajectories received for complex mechanical systems (such as axial piston pump) contains information on object energy distributed in a wide band of frequencies. This often leads to receive trajectories with complex shapes, difficult to further interpretation. In order to limit complexity of estimated trajectories, measured vibration signal have been filtered by moving average filter. This led to narrow down frequency band of analyzed pumps energy distribution. Next, the phase trajectories (called by author real phase trajectories) were estimated and approximation of real shape trajectories by substitute curves (so called substitute trajectories) have been proposed. It was assumed elliptical form of substitute trajectories Figures 12, 13 and 14.

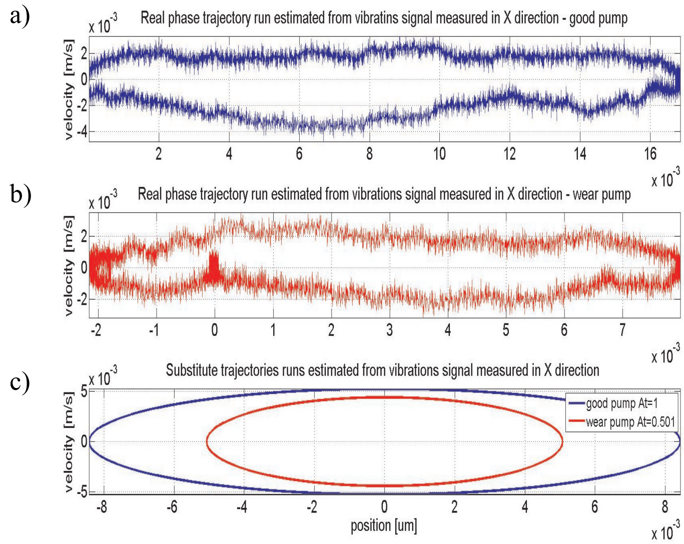

Fig. 12. Comparison of real and substitute phase trajectories runs estimated from vibration signal measured in *X* direction: a) real phase trajectory for good pump; b) real phase trajectory for wear pump; c) substitute trajectories

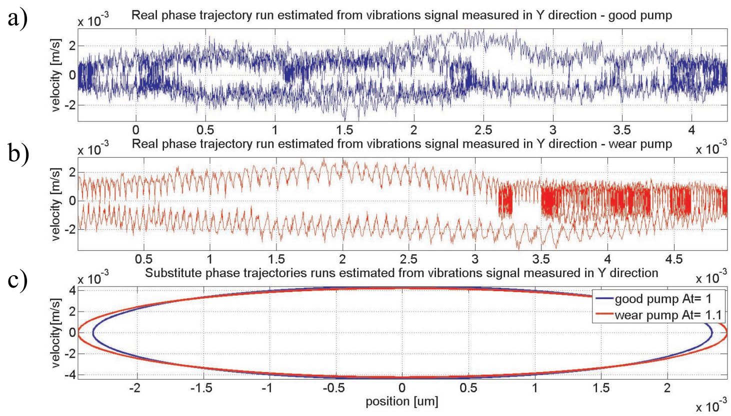

Fig. 13. Comparison of real and substitute phase trajectories runs estimated from vibration signal measured in *Y* direction: a) real phase trajectory for good pump; b) real phase trajectory for wear pump; c) substitute trajectories

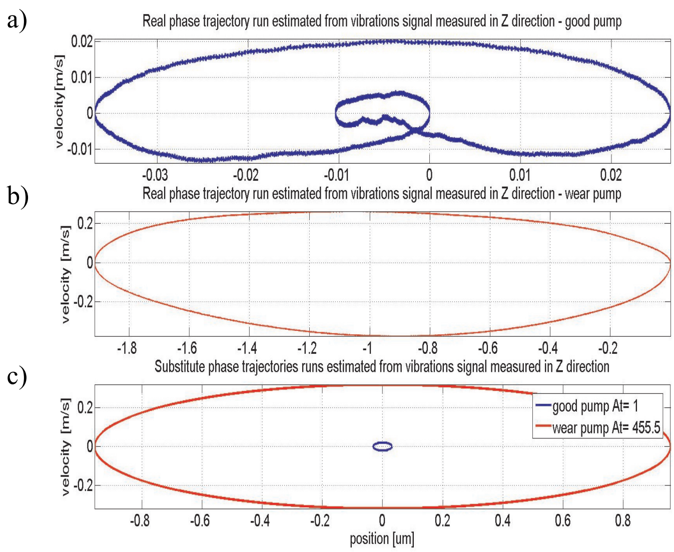

Fig. 14. Comparison of real and substitute phase trajectories runs estimated from vibration signal measured in *Z* direction: a) real phase trajectory for good pump; b) real phase trajectory for wear pump; c) substitute trajectories

So obtained substitute phase trajectories have been given a quantitative measure, specified as dimensionless coefficient $At_{p,i}$ (4) of normalized surface area covered by phase trajectory. This coefficient was determined for p point of pump's body measured in i direction (axis: X, Y or Z). But, as the value of $At_{p,i}$ = 1 was assumed normalized surface area covered by phase trajectory assigned for pump without wear.

$$At_{p,i} = \frac{\dot{k}_{p,i,_{wear}} \cdot k_{p,i,_{wear}}}{\dot{k}_{p,i,_{efficient}} \cdot k_{p,i,_{efficient}}} \tag{4}$$

where:

$At_{p,i}$ – normalized surface area covered by phase trajectory determined for p point of pump body in i direction,

$\dot{k}_{p,i,_{wear}}, k_{p,i,_{wear}}$ – respectively: average values of velocity and displacement for p point on pumps body in i direction for pump with worn out element,

$\dot{k}_{p,i,_{efficient}}, k_{p,i,_{efficient}}$ – respectively: average values of velocity and displacement for p point on pumps body in i direction for pump without wear.

Obtained charts for substitute phase trajectories with specified dimensionless coefficients $At_{p,i}$ (for x, y and z direction) were presented on Figures 15, 16 and 17. Additionally dimensionless coefficients $At_{p,i}$ were put together in Table 1.

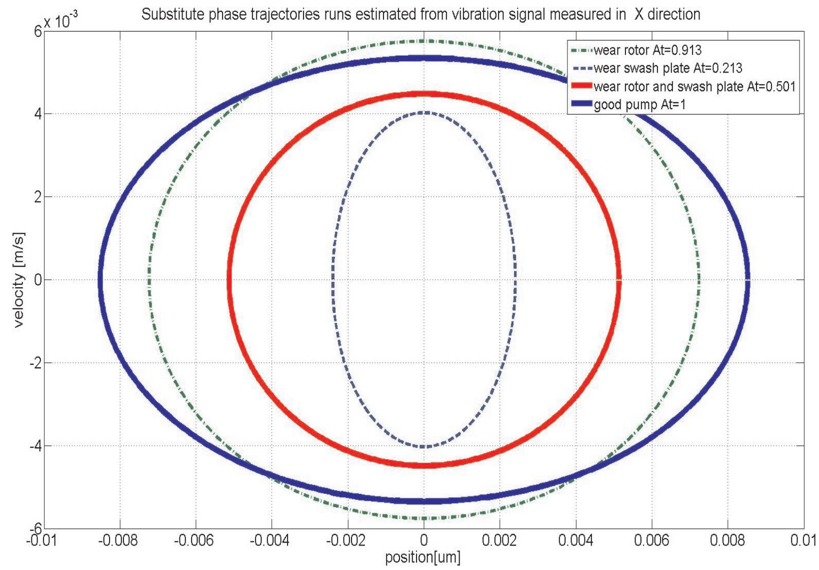

Fig. 15. Substitute phase trajectories runs estimated from vibration signals in X direction

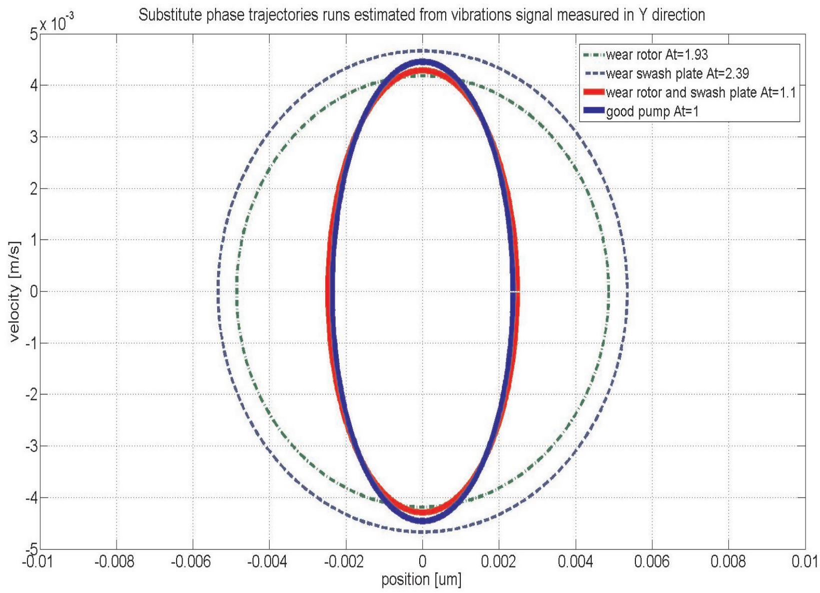

Fig. 16. Substitute phase trajectories runs estimated from vibration signals in Y direction

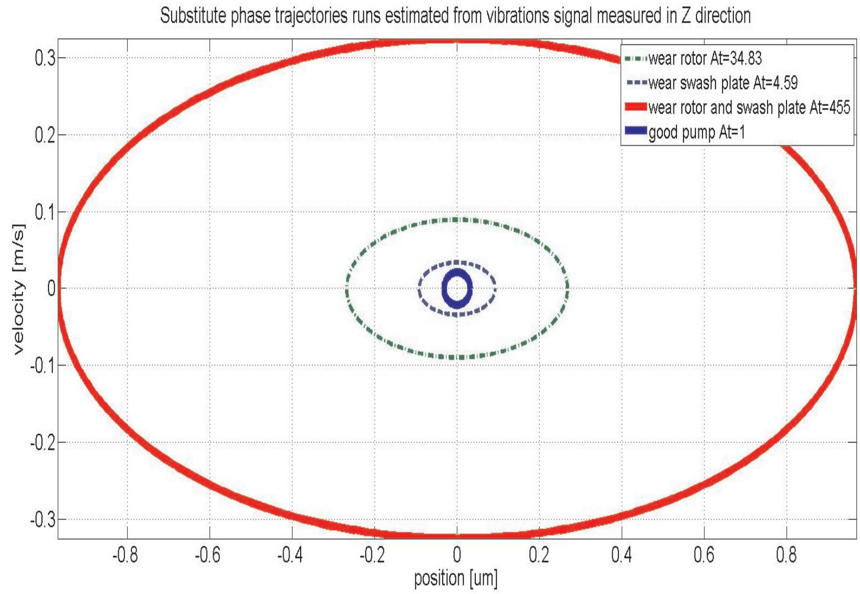

Fig. 17. Substitute phase trajectories runs estimated from vibration signals in Z direction

Table 1

Comparison of $At_{p,i}$ coefficients

i measurement direction	Pump in good state	Wear swash plate	Wear rotor	Wear swash plate and rotor
X	1	0.213	0.913	0.501
Y	1	2.39	1.93	1.1
Z	1	4.59	34.83	455

5. SUMMARY

Received phase trajectories for the positive displacement pump in a good state and pump equipped with damaged elements (rotor unit and swash plate), have shown high sensitivity on investigated damage elements in all measured directions (*X*, *Y*, *Z*). From machine model made as energy processor comes that under vibration measurement near rotor unit, for transducer mounted in *Y* and *Z* direction (Figs. 16 and 17) it can be seen the growth of pump's body energy. It is due to dissipation of total energy in piston-cylinder and piston shoe – swash plate kinematic pairs. In the case of measurements in *X* direction received phase trajectories (Fig. 15) depict decrees of total energy with progressive degradation of investigated pump elements.

Outlined above results indicates that the use of phase trajectory could be a useful tool for axial-piston pump state monitoring. The quantitative measure of phase trajectories specified as dimensionless coefficient $At_{p,i}$ (4), allows to link them with the condition of analysis pump component.

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