THE ARCHIVE OF MECHANICAL ENGINEERING

VOL. LVII 2010 Number 4

10.2478/v10180-010-0024-9

Key words: axial piston pump, signal analysis, phase trajectory, machine diagnostics

JERZY STOJEK *

APPLICATION OF PHASE TRAJECTORIES TO THE PROBLEMS OF EXPLOITATION OF AXIAL PISTON PUMPS

In this paper, the author presents the possibility of using phase trajectory for detecting damage in an axial piston pump. The wear on main part of pump elements, such as the rotor and the valve plate, was investigated, and phase trajectories were determined based on vibration signal measured in three directions on the pump's body. In order to obtain a quantitative measure of the analyzed trajectory, the $At_{p,i}$ parameter was introduced, and the relation between this parameter and the wear on the pump's parts was determined.

1. Introduction

Axial piston pumps take particularly important place in the systems of power hydraulics. In many cases, proper functioning of the whole hydraulic system depends on their proper operation. The wear on particular pump elements most often leads to limiting the operational pressure of the pump, increasing volumetric loses; thus in consequence leads to limiting the general pump efficiency, increasing vibrations and the related noise. Vibration-acoustic exploitation diagnostics of the pumps, performed in exploitation period, is aimed at looking for symptoms of a hypothetical damage that might appear in the vibration signal. In the case when the level of disturbances is high, and the mechanical system is complex, uncertainty of these results might be significant. The diagnostics based on the process analysis [7] indicates that there exists a substantial damage, which eliminates the given pump from exploitation. In such a method, the process of damage development is not taken into consideration, thus there is no possibility to predict the damage. The search for new methods of diagnostics of axial piston pumps

* *AGH University of Science and Technology, Department of Process Control al. Mickiewicza 30, 30-059 Cracow; E-mail: stojek@imir.agh.edu.pl*

is based on assumptions which eliminate the drawbacks of the mentioned methods. In the present article, the author presents the possibilities of applying the methods of technical statics theory [4] in the exploitation process of a positive-displacement pump.

2. Desription of investigated object

The subject of the study was an axial multi-piston fixed-displacement pump, whose simplified diagram is shown in Fig. 1. In a pump of this type, the rotor (2) together with the system of pistons (3) and the supporting bearing (7) is mounted coaxially on the drive shaft (1). The shoes of pistons (4) cooperate with an immobile swash plate positioned at an angle γ against the axis of the pump rotor. The pistons, together with the rotor, make a rotational motion. At the same time, their shoes (4), which slide on the surface of the immobile swash plate, force a reciprocating motion in the rotor cylinders. Additionally, the rotor slides on the immobile valve plate (6), where are placed suction and pumping openings of the pump.

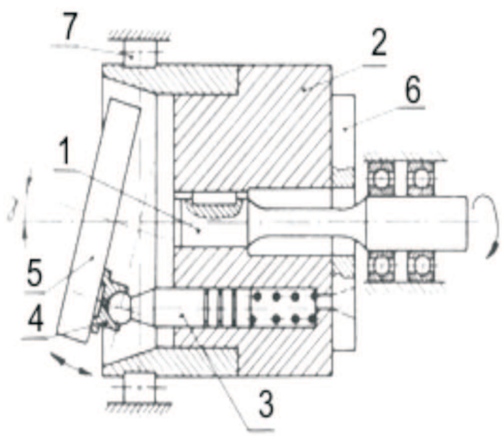


Fig. 1. General scheme of axial multi-piston pump: 1 – shaft, 2 – rotor, 3 – piston, 4 – piston shoe, 5 – swash plate, 6 – valve plate, 7 – bearing [6]

In multi-piston pumps, the rotor system is usually the most important element, which directly decides about the intensity of flow generated in the pistol connection of the pump. The existing solutions of multi-piston pump structures differ one from another with respect to the method of placing the rotor system within the pump structure, as well as the kind of rotor drive. The basic system of the pump rotor, used in the presented experiment, contained a rotor consisting of two component elements and seven pistons placed in its cylinders. The entire rotor system was supported on a roller bearing.

3. Causes for wear in displacemet pump elements

The wear of displacement pump elements is caused by forces which occur during the cooperation of particular components constituting the motion pairs (for example, piston-cylinder, valve plate – rotor, piston shoe – swash plate). The wear also results from improper exploitation conditions of the pump, such as (among other things):

- exceeding the nominal working pressure of the pump,
- operating with too low viscosity of the working fluid,
- lack of the working fluid, or unsatisfactory filtration of the fluid.

The most frequently kind of wear occurring on displacement pump elements is of abrasion type [2]. The wear of this type appears in all these elements of the pump which are in a relative motion, and are in contact with other elements.

The wear of the rotor system is caused mainly by the forces which exist on the surfaces of the piston-cylinder kinematic pair. These forces result from the load exerted on the piston by a normal force, being a sum of the radial reaction of the swash plate acting on the piston shoe and the inertial force of the rotor rotating with the assumed angular speed. Excessive loading of the rotor system leads to (among other things) abrasive wear on its components and to an increase in the radial clearance in the piston-cylinder pair. The result of it is an increase in volumetric losses and a decrease in the overall pump efficiency.

The task performed by the swash plate cooperating with the rotor mainly consists in ensuring proper adjustment of the pump efficiency by changing the inclination angle of the shield axis with respect to the axis of the driving shaft. The change in the inclination angle of the shield translates into the length of stroke of the piston in the rotor cylinder and, at the same time, translates into the capacity of the pumped medium in the working pump cycle. Physically, the swash plate cooperates with the surfaces of the piston shoes of the rotor, which, while making the rotational motion, slide on its surface. The cooperation between these elements is burdened with some loses that result from friction on the sliding surfaces. Minimization of the losses, and therefore, an increase in mechanical and hydraulic efficiency of the pump is achieved by hydrostatic support of the shoes on the shield surface. In the case when such a support disappears, there appears wear on the shield surface (and piston shoes), which gradually increases and leads to creating an elliptical valley on its surface and eventually to wearing it out totally.

In turn, wear on the valve plate is caused, among other things, by disappearance of the lubricant layer between the shield surface and the surface of the rotor face. Consequently, mixed friction (or dry friction) arises between cooperating parts and causes attrition on the surfaces. The result of the wear is the appearance of flow micro-channels on the surface of the shield bridge. The working medium flows through these channels between the suction zone and the pumping zone of the pump, and the result of it is the lack of leak tightness, a decrease in exploitation pressure and deterioration of volumetric efficiency of the pump. It should be also remembered that, in the case of exploitation of a pump with the valve plate with the so called negative overlap, the surface of the shield bridge is subjected to harmful phenomena of cavitations.

4. The course of investigations

The investigation on the wear in multi-piston pump elements was conducted on a special experimental stand, constructed specially for this purpose. Its principal element was a brand-new multi-piston axis pump of constant delivery, together with hydraulic accessories and installed measuring transducers. One of the aims of the study was to cause wear on the pump components in a natural way, so that the study consisted of long-lasting tests, during which realistic conditions of pump exploitation were maintained. In the investigation on the development of wear in the pump, it was assumed that the pump elements would be loaded with a constant pressure of 70 bar; such a value was set by proper adjustment of the area of the flow gap in the throttle valve.

In order to fasten the process of wear in the pump elements, and to model work of the pump in heavy industrial conditions (with unsatisfactory filtration of the working fluid), after carrying out the investigation for a month, we added a portion of grinded rock crystal powder (15 g in 60 liters of oil) to the working fluid, and we continued the investigations afterwards. During the whole experiment, we recorded diagnostic signals from the transducers installed on the pump. Flow of the working fluid, static and dynamic pressure, and vibration acceleration of the pump body were continuously registered.

The measurements of vibration acceleration of the body pump were carried out for three measurement axes (X, Y, and Z), by means of transducers installed on the pump body near the valve plate, near the rotor and the swash plate. In the measurements of the physical quantities, we used 16- bit data acquisition cards, which cooperated with a conditioning system and were programmed with the use of the LabView software package [9]. The measured courses of signals were stored on a computer disk, and then subjected to off-line numerical analysis using the Matlab-Simulink software package.

A simplified diagram of the system for measurement of the pump body vibrations is presented below (Fig. 2). Allocation of vibration transducers is also shown.

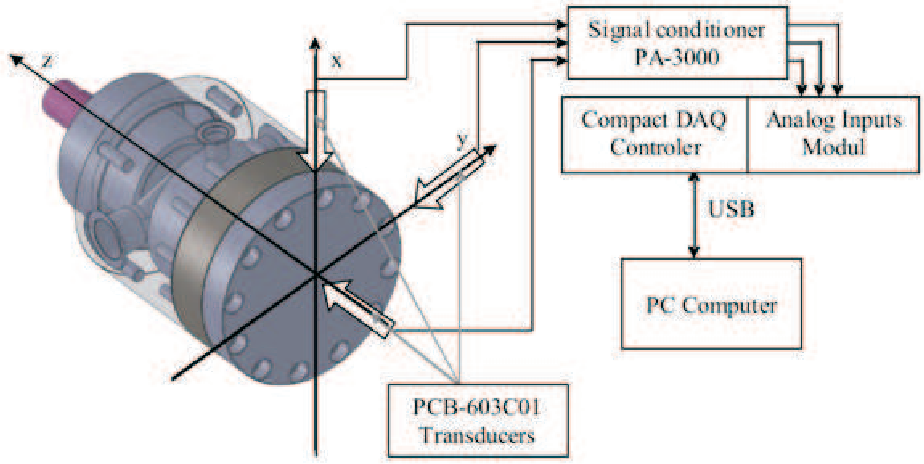


Fig. 2. Simplified block diagram of the system for measurement of pump body vibration on the experimental stand

After laboratory investigations were completed, we evaluated wear in the pump's components. Visual inspection of the pump components revealed wear in the rotor system and on the valve plate. We did not observe wear on the swash plate surface, neither on the surface of the piston shoes. The lack of wear in this kinematic pair was due to the fact that, in static conditions of the pump work, the lubrication layer between the cooperating surfaces of the swash plate and the piston pad did not disappear.

The observed wear in the rotor system resulted in increasing the radial clearance in every piston-cylinder pair, on average by about 10 μm. Additionally, we found that the rotor face, which cooperated with the surface of the valve plate, was worn out. The degree of degradation of the rotor face was measured by means of a contact profilometer [8] in three measurement directions chosen as shown in Fig. 3. Having compared the three profiles, we found that the wear on rotor face was practically uniform, with an average wear deepness of about 50 μm (Fig. 4).

On the other hand, the wear on the valve plate surface caused that flow microchannels appeared on the surface of the transient zones (the so-called bridges) between the suction channel and the pumping channel. Having measured the profiles of the transient zone *A* (the pathway from the suction side to the pumping side) and the transient zone *B* (the pathway from the pumping side to the sucking side), we found that the two zones were not equally worn

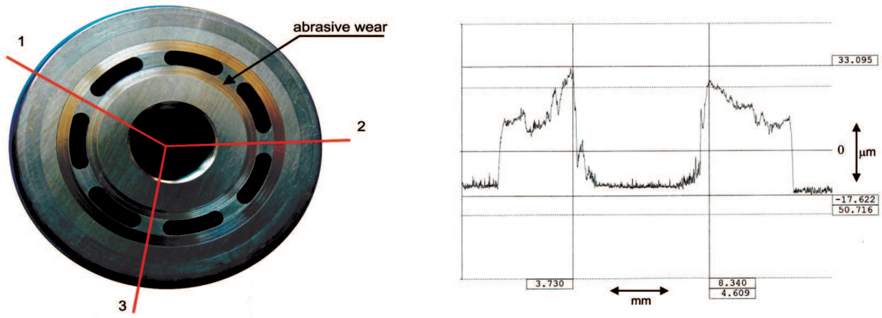


Fig. 3. View of rotor front face and its wear profile

out, (Fig. 4), and that degradation of the surface of transient zone *A* (the pathway from the suction side to the pumping side) was greater than that of zone B.

Fig. 4. View of valve plate and wear profiles of its transient zones

5. Description of phase trajectory method

Classic methods of vibro-acoustic diagnostics of machines, based on time or time-frequency analysis of the measured signals, require some experience from the person interpreting the measurement results [3]. It is especially important in the case of analyzing the effects of vibration and acoustic processes occurring in machines and devices of high complexity, in which the movements of components interact with, and influence one another. Additionally, in many cases proper interpretation of the measured time and time-frequency distributions is hindered by the existence of parasitic components (disturbances), which originate from elements (devices) not associated with the investigated object. In the search for a method allowing for a faster and easier interpretation of the state of the investigated pump, the author proposed applying the method based on phase trajectories .

Detection of an initial phase of damage of individual pump elements (such as valve plate, rotor or swash plate) should not depend on disturbances associated with the work of the remaining components. However, it is impossible to extract information pertaining to a specific element from the available diagnostic signal. Thus, we deal with a machine, a device, or an element working in the environment of other machines, devices and elements, which exert an influence on the examined object. This influence, irrespective of the character of its effect, can be described by functions later called the permanent disturbances. The problem formulated in this way is described in professional literature as stability in the Lyapunov sense, or the technical stability [1].

Analyzing a concrete element of the multi piston pump subjected to forces produced by other elements of the pump or elements of the hydraulic system, we can formulate the equation of the examined motion in the following way:

$$\ddot{x} = f(\dot{x}, x, t) \tag{1}$$

This equation has unique solutions defined by initial conditions. Taking into account the influence of the environment on the examined element represented in the form of a permanently acting disturbance "R", we obtain

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

We can solve the above equation by substitution and by reducing its order, which leads to

$$\dot{x} = f(x, t) + R(x, t) \tag{3}$$

Taking into consideration the fact that the function $R(x, t)$ takes into account acceptable deviations from the steady state and changes in initial conditions, as well as expected internal and external disturbances acting on the considered element (an object or a system) – which might have stochastic or periodic character – it is possible to define dynamic state of an object in terms of technical stability [4].

In the proposed approach we don't need full identification of the system structure, i.e. it is not necessary to precisely define the function$f(x, t)$, we only concentrate on the solutions to equation (3). An effective tool for examining the systems of differential equations is trajectory analysis in the phase space. It follows on the definition of technical stability of a system that, if for initial conditions enclosed in the area ω of the phase space the solutions of the system (3) lie in the area Ω, then the examined element (a configuration, an object) is technically stable (Fig. 5a).

When we examine an actual object, such as a multi-piston pump, we deal with its numerous components that interact with one another. The stability of such a system can be determined by measuring parameters of motion of individual elements. Physically, there are available only the elements connected with the pump's body. Vibration parameters of the pump are associated with the motion of pump components. Therefore, one must choose a proper place on the body, which is geometrically connected with the investigated pump elements. By measuring dislocation and speed of vibration at a chosen point referred to the phase space, we could characterize a class of solutions for partial differential equations pertaining to particular pump elements. The problem of determining the area Ω can be solved in several ways. In the case of multi piston pumps, the preferred approach will be determining the area Ω on the basis of dynamic analysis of a brand-new pump without wear.

It possible to define this area by investigating the phase trajectory for a pump in good working order, taking into account external disturbances. When we analyze vibrations of pump body, we deal with an effect of non-dampened vibrations around the equilibrium position. Then, the image in the phase space representing such an object will be a phase space with the so-called limit cycle. In such a case, the phase trajectory does not approach the point of equilibrium, but changes into a closed curve which goes round that point (Fig. 5b).

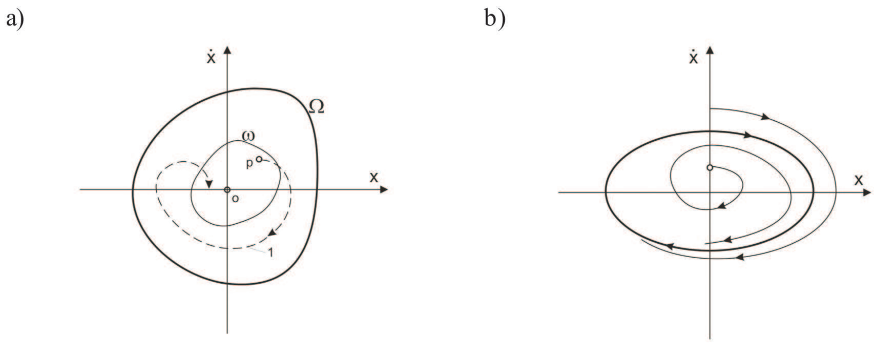


Fig. 5. Illustration of technical stability concept: a) 1 – phase trajectory, Ω – area of permissible deviation from equilibrium state, ω – area of initial conditions, p – initial state of system set out of equilibrium point "o", b) example phase trajectory with limit cycle

The observation of changes of the area containing the trajectories will be used for creating a diagnostic symptom, and then will lead to formulating a diagnostic hypothesis.

It follows from theoretical considerations [1] that it is not known how the area Ω, assumed as technically stable, will change in the result of degradation of an examined element. The formulation of phase space contains information about the energy stored in an examined element (the sum of kinetic and potential energies). It follows from the model of a machine as an energy processor [5] that an increase in the total energy of an examined element leads to an expansion of the area Ω, whose surface grows. This premise will allow identifying the energy structure of the examined object. Observing the elements associated with the working process of the pump (those deciding on the delivery Q_r or the working pressure p_r), in the situation when power efficiency drops, one can expect that the area Ω would shrink. For the elements in which we observe destructive concentration of dissipated energy, this area should expand.

6. Defining phase spaces in the problems of multi piston pump operation

Before we set about using the phase trajectories in the wear analysis of a displacement pump, we should define phase spaces for initial conditions ω (the area of starting point of a phase trajectory) as well as the space of exploitation conditions Ω_r (the area of the pump's work). The definition of these areas is different for each of the assumed measurement points on the pump's body (the rotor, the swash plate and the valve plate) and for each of the assumed directions of measurement (X, Y, Z). Moreover, the area size depends on working conditions of the investigated pump (e.g. operating pressure, rotary speed of the pump shaft, the temperature of the working fluid):

$$(\omega, \Omega_r) = f(mp, os, eksp) \tag{4}$$

where:

mp – assumed measurement points on the pump's body,

os – assumed direction of the measurement,

eksp – working conditions.

Physically, the value of initial conditions ω can be determined based on the courses of vibrations of the pump's body in the conditions when the examined pump does not work, and there are physical phenomena originating from other objects (needed, for example, to ensure proper work of the pump) that exert an influence on the pump and generate vibrations in its elements. In the examined system,, such an object was the booster pump, which was applied in order to ensure comfortable working conditions for the main pump. This pump was the source of vibrations of the body of the examined pump, and these were used to define the area of its initial conditions ω (Fig. 6).

A different approach to defining the area of initial conditions ω consists in determining them on the basis of idle work of the examined pump (without loading its pressure conduit with working pressure). The choice of the area of the pump exploitation conditions Ω_r for an assumed place and direction

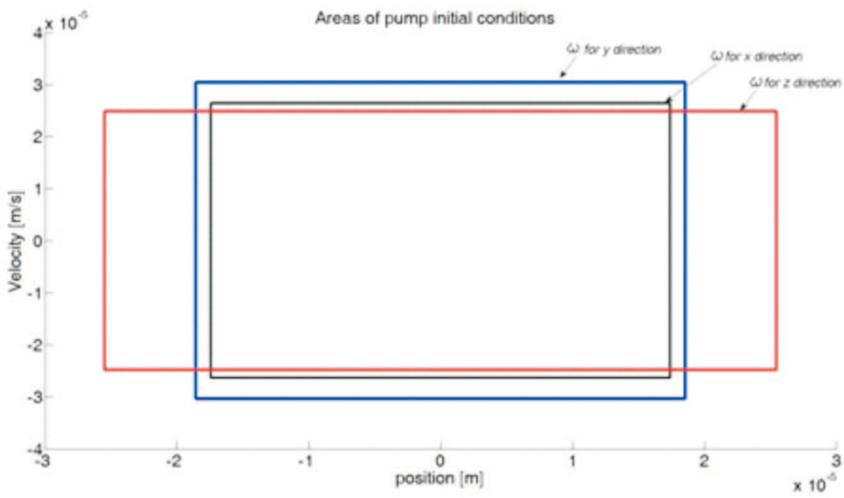


Fig. 6. Area of initial conditions ω determined for examined pump with booster pump set to work

of vibrations of the pump's body, is mainly determined by its operating parameters, i.e.

- working pressure of the pump p_r,
- working speed of the pump's shaft n_r,
- temperature T_r of the hydraulic system in working conditions (or viscosity μ_r of the working fluid).

There are some limitations imposed on these parameters by the pump producer:

- for the working pressure of the pump: $p_r \leqslant p_{nom}$, (where p_{nom}– nominal pressure of pump),
- for the rotational speed of the pump's shaft: $n_{\min} \leqslant n_r \leqslant n_{\max}$, (where $n_{\min}$, $n_{\max}$– minimum and maximum value of rotational speed allowable for examined pump),
- for the exploitation temperature of the pump: $T_{min} \leqslant T_r \leqslant T_{\max}$, (where $T_{\min}$, $T_{\max}$ – minimum and maximum allowable temperature for examined pump),
- for the viscosity of the working fluid: $\mu_{\min} \leqslant \mu_r \leqslant \mu_{\max}$, (where $\mu_{\min}$, $\mu_{\max}$ – minimum and maximum value of working fluid viscosity for examined pump).

In the presented investigations, the area of the pump exploitation values Ω_r was determined for each of the three measurement axis X, Y and Z, and for each of the assumed measurement points on the pump's body (the valve plate, the rotor and the swash plate) separately. During the tests, we maintained a constant value of the pump working pressure (p_r = 70 bar), a constant rotational speed of the pump's shaft, equal to its nominal speed (n_r=1500 rev/min) and a constant temperature of the working fluid (T_r=55°C).

5.1. Results of examinations and final conclusions

The measured accelerations of vibrations of the pump's body were used to determine the trajectories in the phase space for three measurement directions. The method of determining phase trajectories was based on numerical integration of the courses of vibration acceleration of pump's body measured by the transducers mounted in the vicinity of the rotor, the valve plate and the swash plate. The actual shape of the experimentally-determined courses of phase trajectories for a complex mechanical system (such as displacement pumps) contains information on the distribution of the object's energy in a wide frequency range. In many cases, one obtains trajectories of complicated shapes, whose interpretation might be difficult. Taking it into consideration, we applied filtering to smooth the registered trajectory courses, and then the courses of phase trajectories were approximated by substitute curves. In Fig. 7, there is shown an exemplary course of an actual phase trajectory (before applying a filter with moving average) and its approximation (the substitute trajectory), obtained by vibration measurement on the swash plate in the axis Z.

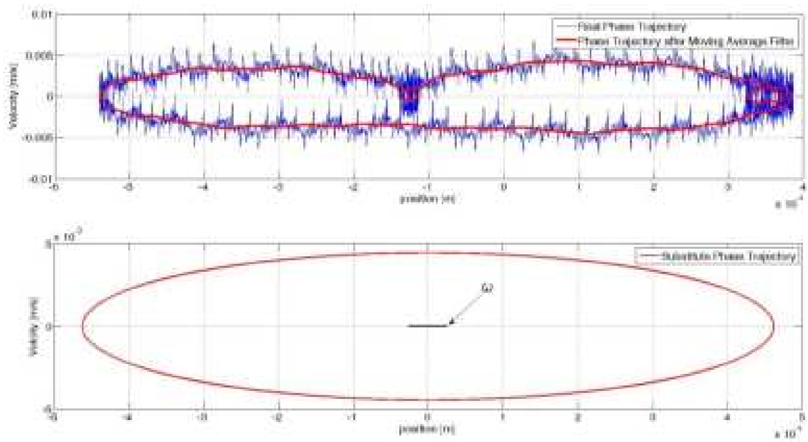


Fig. 7. Comparison of actual and substitute phase trajectories determined from vibration signal measured on swash plate in Z direction

The substitute phase trajectories determined in this way were assigned quantitative measures defined as a dimensionless coefficients of the normalized surface of the area $At_{p,i}$ (5) circumvented by the trajectory. The latter was determined for the measurement point p on the pump's body, and the measurement direction i (of the axis X, Y or Z). At the same time, the unit value $At_{p,i} = 1$ was defined as the normalized area of trajectory of the point p in the direction i, determined at the beginning of exploitation of the pump (the pump in good working order).

$$At_{p,i} = \frac{\dot{k}_{p,i_{,wear}} \cdot k_{p,i_{,wear}}}{\dot{k}_{p,i_{,efficient}} \cdot k_{p,i,efficient}} \tag{5}$$

where:

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

$\dot{k}_{p,i_{,wear}}, k_{p,i_{,wear}}$ – respectively: velocity and displacement for point p on pump body in direction i during exploitation (worn-out pump),

$\dot{k}_{p,i_{,efficient}}, k_{p,i_{,efficient}}$ – respectively: velocity and displacement for point p on pump body in direction i at the beginning of exploitation (pump without wear).

One of the observed symptoms of wear in pump elements was the change in static pressure in the pumping pipe of the pump. It was a clear evidence for the appearance of a leak inside the pump, which could be caused, among other things, by an increase in clearances in the piston-cylinder pairs.

Basing on the measured pressure difference in the pumping pipe of the pump (a difference between the pressure value at the beginning, and during the experiment), we defined two percentage degrees of exploitation wear in the pump: 5% and 30%, which reflected the fall of pressure in the pumping pipe of the pump; respectively by 3.5 bar and 22.5 bar.

The determined substitute phase trajectories for the assumed percentage values of the pump wear, together with the values of their normalized coefficients $At_{p,i}$, are presented in Figs. 8 and 9. Additionally, in Table 1, there are juxtaposed calculated values of the coefficients $At_{p,i}$.

a)

b)

c)

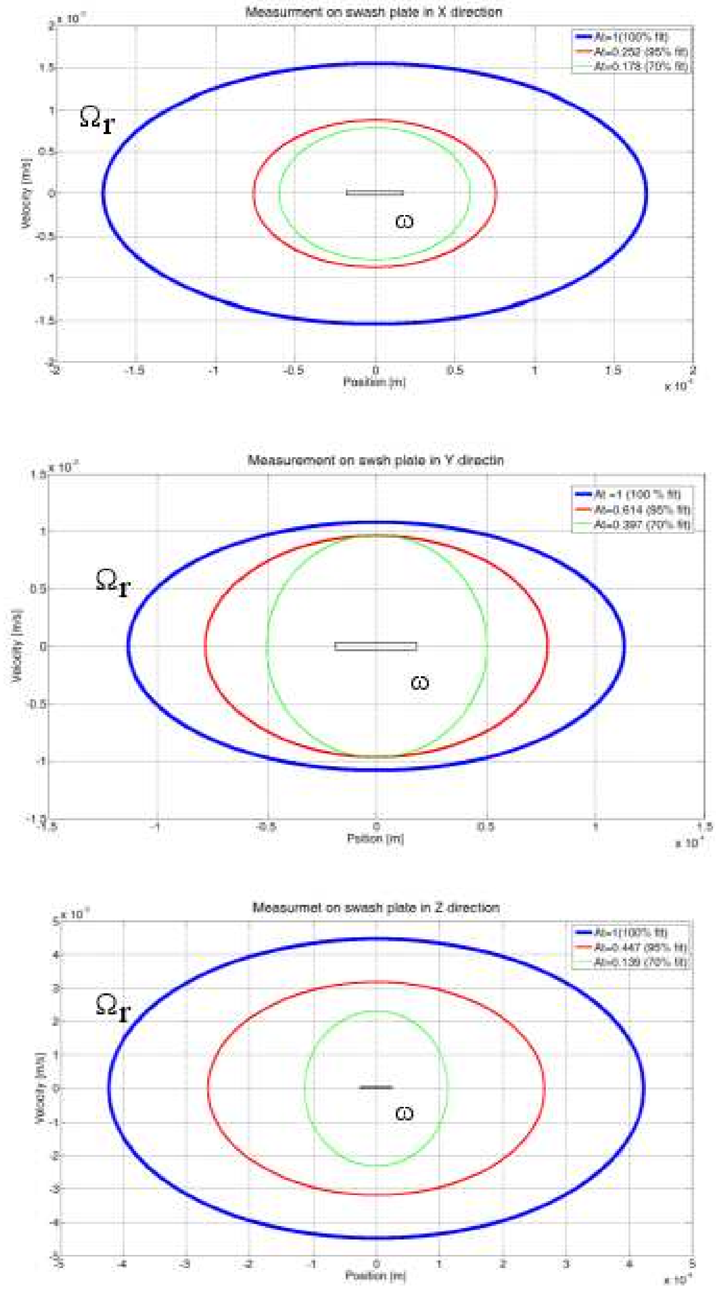
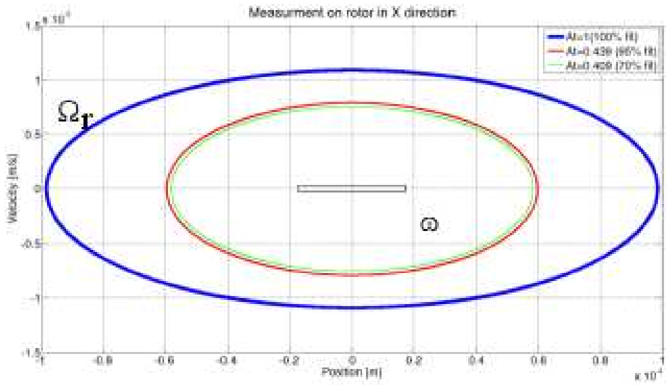
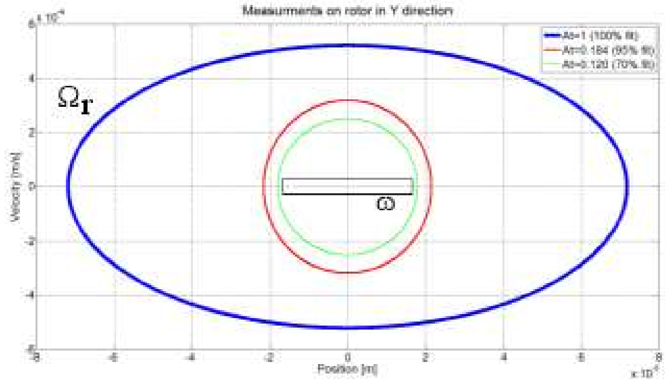


Fig. 8. Courses of substitute phase trajectories determined based on vibration signals measured on swash plate in directions X (a), Y (b) and Z (c)

a)



b)



c)

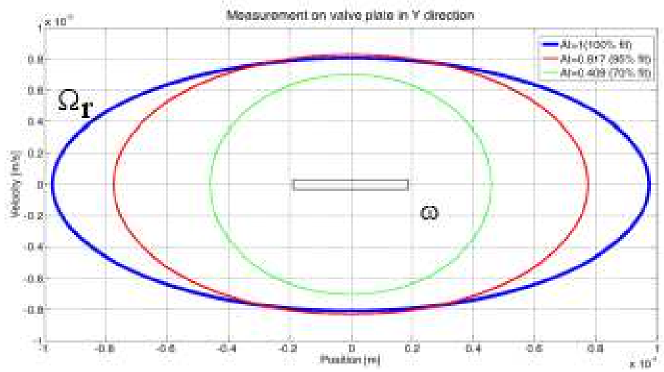


Fig. 9. Courses of substitute phase trajectories determined based on vibration signals measured on rotor in directions X (a), Y (b), and on valve plate in direction Y (c)

Table 1.

Comparison of $At_{,p,i}$ coefficients

Measurement place on pump's body: swash plate			
i measurement direction:	*Pump with no wear*	*Pump with 5% wear*	*Pump with 30% wear*
X	1	0.925	0.588
Y	1	0.764	0.253
Z	1	0.516	0.262
Measurement place on pump's body: rotor			
i measurement direction:	*Pump with no wear*	*Pump with 5% wear*	*Pump with 30% wear*
X	1	0.526	0.194
Y	1	0.249	0.398
Z	1	0.516	0.262
Measurement place on pump's body: valve plate			
i measurement direction:	*Pump with no wear*	*Pump with 5% wear*	*Pump with 30% wear*
X	not measured	not measured	not measured
Y	1	1.078	0.646
Z	1	0.516	0.262

The courses of phase trajectories determined in this experiment indicated substantial sensitivity of their area surface (in all three measurement axes) to the increase of wear in the pump elements (in the system of rotor and the control plate). Referring to the energy model of the machine, one may conclude that in the assumed area of vibrations measurement (on the swash plate, on the valve plate and in the rotor system of the pump) with transducers placed in all three measurement axis, one observes a decrease in the total energy in the investigated pump, accompanied by a progressive degradation of its component elements. The results of measurements, presented in Figs. 8 and 9 show that the use of phase trajectories may be a good indicator of wear symptoms, useful for diagnostics and monitoring of the state of wear in a displacement pump. Parameterization of the obtained phase trajectories by assigning a quantitative measure (the coefficient $At_{p,i}$) to the observed trajectory change makes it possible to relate them to the degree of degradation in the analyzed component of the pump.

The research was supported by the Grant No. N501325135 from the Committee for Scientific Research (KBN)

Manuscript received by Editorial Board, August 18, 2010

REFERENCES

[1] Batko W.: Nowe idee w budowie systemów monitorujących. Przegląd Mechaniczny, nr 11 z 2007, str. 43-47.
[2] Murrenhoff H., Scharft S.: Wear and Friction of ZRCG-Coated Pistons of Axial Piston Pumps. International Journal of Fluid Power. Vol. 7, no. 3, November 2006.
[3] Stojek J. Zastosowanie nieparametrycznych metod analizy sygnału w ocenie stanu zużycia zespołu wirnika pompy wielotłoczkowej. Hydraulika i Pneumatyka, nr 3, czerwiec 2010.
[4] Bogusz W.: Stateczność techniczna. Państwowe Wydawnictwo Naukowe, Warszawa 1972.
[5] Cempel C.: Diagnostyka maszyn. Wyd. Międzyresortowe Centrum Naukowe, Radom 1992.
[6] Stryczek S.: Napęd hydrostatyczny. Wydawnictwo Naukowo Techniczne, Warszawa 1995.
[7] Żółtowski B., Cempel C.: Inżynieria diagnostyki maszyn. Praca zbiorowa. Wydawnictwo PTDT, Radom 2004.
[8] www.hommelwerke.de
[9] www.labview.pl

Wykorzystanie trajektorii fazowych w eksploatacji pompy tłokowej osiowej

S t r e s z c z e n i e

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 zużyciem zespołu pompy.