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exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machinery

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на тему

ПРОБЛЕМЫ ГИДРАВЛИЧЕСКИХ ТУРБОМАШИН

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Computer Simulation of Dynamic Behaviour of a Pump Rotor Assembly with a Balancing Disc

The paper presents preliminary results of computer simulation of dynamic behaviour of a multistage centrifugal pump rotor assembly with a balancing disc in response to a stepwise change of axial thrust. Time characteristics of the disc velocities and displacements as well as the flowing liquid pressures and velocities at the inlet to axial clearance were analyzed as related to the rotating mass and geometric features of the rotor assembly. The computation was carried out using the UNISIM simulation program developed for solution of ordinary differential equations.

Nomenclature

a — diameter ratio, $a = d_{ex}/d_{in}$,
c_1 — mean velocity of liquid in clearance s_1,
c_{in} — mean velocity of liquid at the inlet to clearance s_2,
d_1, d_{in}, d_{ex} — diameters, as shown in Fig. 1,
E_c — modulus of elasticity of the liquid,
F_{dp} — rotor assembly displacement counter-force,
F_w — axial thrust,
ΔF_w — disc vibration driving force,
b — width of chamber I between clearances s_1 and s_2,
l_1, l_3 — length dimensions of clearances as shown in Fig. 1,
m — mass of the rotor assembly of the pump,
p_0, p_1, p_2, p_3 — respective pressures upstream and downstream of the clearances as shown in Fig. 1,
Δp — total drop of pressure in the balancing device, $p = p_0 - p_3$
$\Delta p_1, \Delta p_2, \Delta p_3$ — drops of pressure in specific clearances,
Δp_{in2} — drop of pressure at the inlet to clearance s_2,
$\Delta p_s = \Delta p_2 - \Delta p_{in2}$
s_1, s_2, s_3 — widths of clearances as shown in Fig. 1,
t — time,
V_k — volume of chamber I (Fig. 1),
$x, \dot{x}, \ddot{x}$ — displacement, velocity and acceleration of balancing disc,
ζ_{in1}, ζ_{in2} — coefficients of pressure drops at the inlets to clearances s_1 and s_2,
λ_1 — loss coefficient along clearance s_1,
ψ — stiffness of the balancing system,
ϱ — density of liquid,
ω — angular speed of the rotor assembly.
Index „0" relates to the steady-state condition, before the driving force ΔF_w has been applied.

1. Introduction

In large size rotodynamic multistage pumps the axial thrust is usually taken over by balancing devices which consist of a balancing disc or a balancing disc and a balancing drum (Fig. 1). Computerized analytical methods were developed and implemented in the Institute of Thermal Engineering of Warsaw University of Science and Technology during the last decade to provide means of correct design of such balancing devices. These methods make it possible to design correctly the devices which could operate at the minimum [1] or almost minimum energy losses. In the last case the maximum stiffness $\psi = \partial F_w/\partial s_2$ or the best stability to an increase in the axial thrust F_w or to a wrong estimation of the axial thrust can be guaranteed at the same time [2]. The computations are applicable to the steady-state operating conditions of the pump.

The full evaluation of the design of such a balancing device would still require the assessment of its properties related to the non-steady conditions: start-up, shut-down or sudden changes of the pump operating parameters. This would provide a range of sets of dimensions to select the best configuration of the device to achieve the best compromise between the efficiency and operating conditions.

Dynamic behaviour of balancing devices under transient operating conditions has not been analyzed to a great extent so far. A few papers exist which deal with these problems in relation to behaviour of hydrostatic thrust bearings [3] and to the pumps [4], but they give approximate and qualitative solutions only. They provide means to classify a system as being stable or not, but lack in determination of the degree of stability (high, medium or low damping capacity). The solutions published so far, which have been developed basing on a very roughly simplified pattern of pressure distribution $p(r)$ in the axial clearance, assume linear relationship between the variables and the disc displacement x.

Following the above, an attempt has been made to determine numerically the axial displacement $x = x(t)$ and other dynamic properties of a balancing device which are brought to effect by a stepwise change $\Delta F = F_w - F_{w0}$ of the axial thrust.

2. Basic System of Dynamic Equations

Dynamic characteristics of a balancing device at the moment $t = 0$, when the driving force $\Delta F_w(t)$ of any nature (which may be stepwise or sinusoidal) begins to act, can be described by the following system of differential equations [5,6]:

$$
\begin{gathered}
m\ddot{x} + F_{dp} = \Delta F_w(t) \\
\pi d_1 s_1 c_1 + \frac{\pi}{4}(d_{in}^2 - d_1{}^2)\dot{x} - \frac{V_k}{E_c}\dot{p}_1 = \pi d_{in}(s_{20} - x)c_{in} \\
\varrho\dot{c}_1 l_1 + \varrho\frac{c_1{}^2}{2}\left(\zeta_{in\,1} + \lambda_1\frac{l_1}{2s_1}\right) = \Delta p - p_1 + p_3 \\
\varrho\dot{c}_{in}\frac{d_{in}}{2}\ln a + \varrho\frac{c_{in}^2}{2}\zeta_{in\,2} + \Delta p_s = p_1 - p_3 - \Delta p_3
\end{gathered}
\qquad (1)
$$

where:

$$x = s_{20} - s_2 \tag{2}$$

$$F_{dp} = f_1(\dot{x}^2, \dot{x}, \dot{x}c_{in}, x, c_{in}^2) \tag{3}$$

$$\Delta p_s = f_2(\dot{x}^2, \dot{x}, \dot{x}c_{in}, x, c_{in}^2) \tag{4}$$

For explanation of geometric dimensions appearing in the above equations see Fig. 1. In addition to x and $\dot{x}$ there are the following variables: pressure p_1 within chamber I and velocities c_1 and c_{in} within radial and axial clearances s_1 and s_2, respectively.

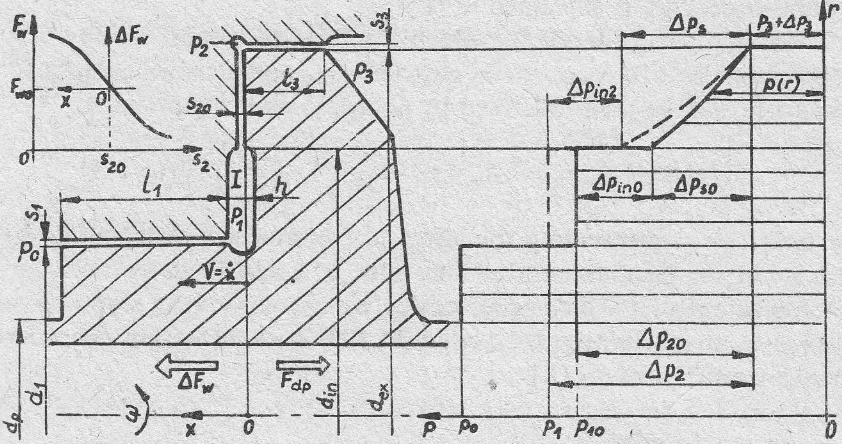

Fig. 1. Geometric features of the balancing device. Pressure distribution and pressure drops are indicated in specific clearances at the moment $t \leqslant 0$ (solid line) and $t > 0$ (dashed line)

The relationships (3) and (4) — for full development see [6] — have been received from the formulae describing distribution of pressure within the axial clearance with one wall displacing in the axial direction [7]. To develop these relationships the flow of liquid within the clearance s_2 with one of its walls moving in the axial direction with constant velocity $\dot{x} = v$ is assumed to be quasi-stationary.

The system of equations (1) with damping force F_{dp} expressed as a complicated non-linear function (3) has been found too complex for the UNISIM program utilized on a rather small PDP computer. Thus, relationship (3) had to be substantially simplified by application of the partial damping coefficient b_1.

3. Partial Damping Coefficient

As the disc moves in the axial direction, pressure p_1 will change with resulting change in pressure drop Δp_2 and hence a change in velocity c_{in} and a change in pressure distribution within the clearance s_2. All these add up to the axial vibration damping force:

$$F_{dp} = F(\dot{x}) + \Delta F_{bal}(x) = F_1(\dot{x}) + F_2(\dot{x}) + \Delta F_{bal}(x)$$

Due to the very close relationship of the variables as expressed by equations (1) this force cannot be practically resolved into component $F(\dot{x})$ related exclusively to the disc displacement velocity $v = \dot{x}$ (proper damping force) and component $F_{bal}(x)$ related exclusively to the displacement. Instead, there can be quite easily estimated two approximate values: the value of component $F_1(\dot{x})$ of the damping force which is connected with the damping effect in the clearance s_2 and the value of the partial damping coefficient $b_1 = F_1(\dot{x})/\dot{x}$. This should be done at several selected values of v when $x = 0$ and $\Delta p_2 = \Delta p_{20} =$ idem (which is the case of a balancing device without damping clearance s_1 and with $p_0 = p_1 =$ idem and $p_2 =$ idem). This rather simple computation procedure is explained in [6].

The remaining part of force F_{dp}, which is related to compressibility of the liquid within chamber I and to reactions of clearances s_1 and s_3, and includes the proper balancing force, can be then described by an approximate expression:

$$F_2(\dot{x}) + \Delta F_{bal}(x) = \frac{\pi}{4} d_{in}^2 (p_1 - p_{10}) \left[1 - \left(\frac{d_1}{d_{in}}\right)^2\right] \chi (a^2 - 1) \tag{6}$$

where χ is a factor representing the effect of the pressure distribution $p(r)$ within clearance s_2 on the balancing force. The value of χ can be determined for a particular configuration of the balancing device by calculating its static characteristic curve $s_2(F_w)$ by means of existing programs [2]. A computation procedure for an exemplary layout is given in [6].

4. Results of Computation

The correct performance of the balancing device in response to a stepwise driving force ΔF_w can be considered as a proof of its proper operation at any smaller load variations practically met under operating conditions. Numerical simulation was carried out [6] for the said kind of driving force with the values of the ratio $\Delta F_w/F_{wo}$ equal to 0.05 and 0.10, since more severe changes are very rarely expected in practice.

Figures 2 and 3 show computation results as obtained for a particular configuration of balancing device used in the 15Z28 feed pump ($Q = 275$ m^3/h, $H = 1820$ m, $\Delta p = 15.5$ MPa, $n = 4660$ rpm) of the dimensions (Fig. 1): $d_{ex} = 320$ mm, $d_{in} = 200$ mm. $d_1 = 120$ mm, $l_1 = 250$ mm, $s_1 = 0.24$ mm, $l_3 = 30$ mm, $s_3 = 0.47$ mm. For $t < 0$: $F_{wo} = 393.2$ kN, $s_{20} = 81.2$ µm, $\psi_0 = 5.6$ kN/µm, $p_{10} = 10.2$ MPa, $c_{10} = 30.6$ m/s, $c_{in0} = 54.7$ m/s. Intitial values ($t = 0$) were as follows: $\dot{x} = 0$, $x = 0$, $\Delta F_{wo} = 0.05\ F_{wo}$.

The computation results can be concluded as follows:

a) Axial vibrations of the rotor assembly deviate severely from a linear function.

b) A new balance conditions stabilize within $t = (0.5, ..., 1.2)\ 10^{-2}$ s which indicates very fast damping action. However, the damping is milder than that shown in case of idealized linear systems [5,6] in spite of rather high values of coefficient b_1 *(four to six times than that found according to [4]).*

c) Transient state is very strongly influenced by dimension h, i.e. the width of

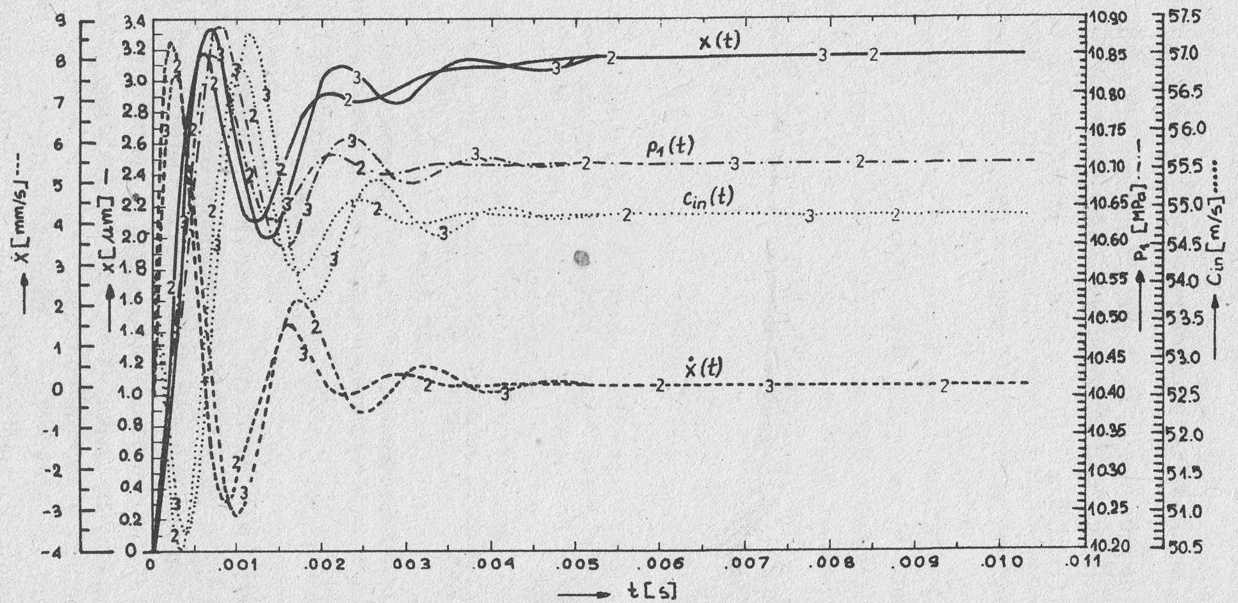

Fig. 2. Time characteristics: $x(t)$, $p_1(t)$, $c_{in}(t)$, $\dot{x}(t)$ as plotted for one of the configurations of balancing devices in the 15Z28 feed pump. These are solutions to the system of equations (1) in response to a stepwise driving force $\Delta F_w/F_{w0} = 0.05$. Curves *2* and *3* correspond to the rotor assemblies of 200 kg and 300 kg mass, respectively

chamber I between the disc and the counterdisc (Fig. 1) considered as a measure of volume of this chamber. The volume $V_k = \frac{\pi}{4}(d_{in}^2 - d_1^2)$ has been varied within $\left(\frac{1}{3}, ..., 2\right) V_{kp}$, where V_{kp} is volume of the chamber of basic width $h = 30$ mm. V_k is the value through which the influence of the liquid compressibility on dynamic properties of the balancing device will manifest as results from the second equation in the equation system (1). In Fig. 3, there are shown time characteristics $x(t)$ referred

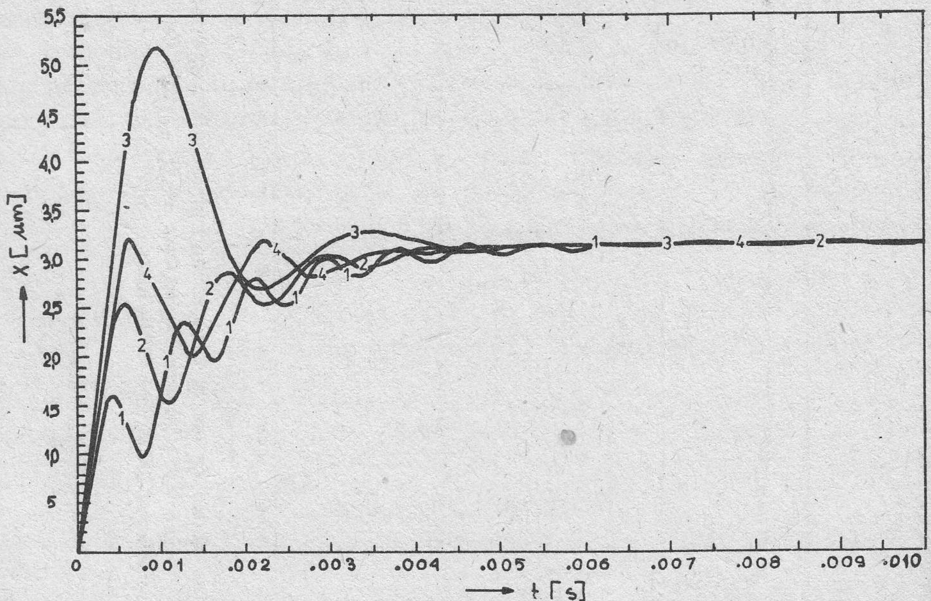

Fig. 3. Effect of the width h of chamber I between the disc and the counterdisc on time characteristics $x(t)$ with configuration of balancing device as considered in Fig. 2 and the mass of rotor assembly equal 300 kg

Curve *1*: $h = 10$ mm, curve *2*: $h = 20$ mm, curve *3*: $h = 60$ mm, curve *4*: $h = 30$ mm

to different values of width h. It can be clearly seen that the increase in h is followed by an increase in the highest value x_{max} of instantaneous displacement at the beginning of the response movement. At $h = 60$ mm it is twice as large as that at $x_0 = \Delta F/\psi_0$ which would stabilize at the new steady-state operating conditions. This indicates that displacements x_{max} may in certain cases exceed permissible values. Designers should then avoid chambers of too big volumes.

d) The mass of rotor assembly m has no significant influence on the vibrations (Fig. 2). With the rotor mass increased by 50 per cent the amplitudes rise by 5 to 10 per cent and complete damping occurs within a little longer period. It will then be sufficient to estimate rotor assembly mass at the design stage within 20 to 30 per cent accuracy.

e) Other results indicate [6] that length l_1 of the axial annulus has much less effect on the dynamics of the balancing device than it could be expected.

5. Remarks and Final Conclusions

1. These computations are the first published that provide quantitative description of the time characteristics in relation to all variables of balancing system, i.e. displacement, velocity and pressure.

2. The results obtained enable the estimation of such properties of balancing devices as stability, degree of vibration damping, maximum vibration amplitude and duration of transient operating condition.

3. Severe non-linearity of transient phenomena indicates that the linearization of the motion equations, generally used in such cases, may yield solutions of insufficient accuracy.

4. A test stand has been constructed in the Institute of Thermal Engineering and research work commenced to verify the values of coefficient b_1 responsible for damping in the axial clearance. Further computer simulations are being planned. Experimental tests on dynamic behaviour of rotor assembly of feed pumps used in 200 MW or 360 MW power units are intended as well.

References

[1] K. Jackowski, W. Jędral, *Wpływ wybranych czynników na wynik optymalizacji energetycznej zespołu bębna i tarczy odciążającej* (On the influence of selected factors on the energy optimization of a balancing drum and disc system). Prace IMP, no. 83—84, 1983, pp. 73—81.

[2] K. Jackowski, W. Jędral, M. Rostkowska, *Badania sztywności oraz opracowanie kompletnego algorytmu projektowania układów z tarczami odciążającymi.* Sprawozdanie z pracy wykonanej przez ITC P.W. dla OBR Pomp Przemysłowych (Research on stiffness and a complete design algorithm of systems with balancing discs. Report on the work carried out by the ITC P.W. for the Industrial Pumps Research and Development Centre). Warsaw 1981.

[3] M. Heil, M. Savci, *Dynamische Eigenschaften hydrostatischer Axiallager bei kleinstmöglichen Gesamtleistungsaufwand.* Konstruktion, no. 4, 1975.

[4] V. A. Martsinkovski, *Analiz osevoi ustoichivosti rotorov pitatel'nykh nasosov.* Energomashinostroenie, no. 4, 1973; also: Lopastnye nasosy, Leningrad, Mashinostroenie 1975.

[5] W. Jedral, M. Rostkowska, *Wprowadzenie do numerycznej symulacji układów z tarczami odciążającymi.* Sprawozdanie z pracy wykonanej przez ITC P.W. w ramacn PR-8.6.4.08.04 na zlecenie IMP PAN (Introduction to numerical simulation of systems with balancing discs. Report on the work carried out by the ITC P.W. under the PR-8.6.4.08.04 project at the IMP PAN order). Warsaw 1982.

[6] W. Jędral, M. Rostkowska, *Numeryczna symulacja własności dynamicznych układów z tarczami odciążającymi.* Sprawozdanie z pracy wykonanej przez ITC P.W. w ramach PR-8.6.4.08.04 (Numerical simulation of dynamic properties of systems with balancing discs. Report on the work carried out by the ITC P.W. under the PR-8.6.4.08.04 project). Warsaw 1983.

[7] W. Jędral, *Przepływ cieczy w szczelinie poprzecznej ze ścianką wirującą i przemieszczającą się w kierunku osiowym.* VI Krajowa Konferencja Mechaniki Cieczy i Gazów, Porąbka-Kozubnik 1984 (Liquid flow through an axial clearance with rotating and axially moving wall. VI-th National Conference on Liquid and Gas Mechanics, Porąbka-Kozubnik 1984). Zeszyty Naukowe Politechniki Śląskiej, Energetyka, no. 87, 1984.

Numeryczna symulacja dynamiki zespołu wirującego pompy z tarczą odciążającą

Streszczenie

W pracy przedstawiono wstępne wyniki numerycznej symulacji dynamiki zespołu wirującego wielostopniowej pompy odśrodkowej z tarczą odciążającą. Własności dynamiczne urządzenia odciążającego (rys. 1) opisuje nieliniowy układ równań różniczkowych (1). Celem uproszczenia rozwiązania równań (1) przedstawiono siłę tłumiącą F_{dp} (3) w postaci (5), gdzie $F_1(\dot{x})$ jest siłą tłumiącą drgania osiowe tylko w szczelinie osiowej s_2. Współczynnik tłumienia $b_1 = F_1(\dot{x})/\dot{x}$ można oszacować w przybliżeniu zgodnie z [6,7]. Pozostałą część F_{dp} można ocenić na podstawie (6) stosując charakterystykę statyczną $s_2(F_w)$ rozważanego urządzenia.

Obliczenia numeryczne przeprowadzono dla urządzenia odciążającego pompy zasilającej 15Z28 pod wpływem skokowej zmiany $F_w/F_{wo} = 0{,}05$ naporu osiowego F_w. Na rysunku 2 przedstawiono krzywą $x(t)$ przemieszczenia w funkcji czasu oraz inne charakterystyki czasowe urządzenia odciążającego. Maksymalna amplituda wibracji rośnie z h (rys. 3) w wyniku ściśliwości przepływającej cieczy (patrz: drugie równanie układu (1), gdzie V_k jest objętością komory I a E_c jest modułem sprężystości cieczy).

Wpływ masy zespołu wirującego m na charakterystykę czasową nie ma wielkiego znaczenia (rys. 2).

Duża nieliniowość charakterystyk czasowych wskazuje, że linearyzacja równań (1) może w takich przypadkach prowadzić do rozwiązań o niewystarczającej dokładności.

Численная симуляция динамики вращаюегося агрегата насоса с разгрузочным диском

Резюме

В работе представлены предварительные результаты численной симуляции динамики вращаюегося агрегата многоступленчатого насоса с разгрузочным диском. Динамические свойства разгрузочного устройства (рис. 1) описывает нелинейная система дифференциальных уравнений (1). С целью упрощения решения уравнений (1) демпфирующая сила F_{dp} (3) представлена в форме (5), где $F_1(\dot{x})$ является силой демпфирующей осевые колебания только в осевом зазоре s_2. Коэффициент демпфирования $b_1 = F_1(\dot{x})/\dot{x}$ можно оценить в приближении согласно с [6, 7]. Остальную часть F_{dp} можно оценить на основе (6), пользуясь статической характеристикой $s_2(F_w)$ рассматриваемого устройства.

Численные расчёты проводились для разгрузочного устройства питательного насоса 15Z28 под влиянием скачкообразного изменения $F_w/F_{wo} = 0{,}05$ осевого напора F_w. На рис. 2 представлена кривая $x(t)$ перемещения в функции времени, а также другие временные характеристики разгрузочного устройства. Максимальная амплитуда вибрации растёт вместе с h (рис. 3) ввиду сжимаемости протекающей жидкости (см.: второе уравнение системы (1), где V_k обозначает объём камеры I, а E_c — это модуль упругости жидкости).

Влияние массы вращающегося ротора m на временную характеристику не имеет большего значения (рис. 2).

Большая нелинейность временных характеристик показывает, что линеаризация уравнений (1) может в таких случаях вести к решениям недостаточной точности.