Thrust bearing with fluid pivot

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Abstract. This article describes the conditions arising from the operation of compressor and pumping equipment using traditional thrust bearings with mechanical support and leveling linkage system, which negatively affect the operation of dynamic machines in general and the thrust bearing in particular. The possibility of using the hydrodynamic pressure of an oil wedge for hydrostatic support in a thrust bearing was considered, and what positive effects could arise. The working design of a thrust bearing with fluid pivot and a calculation model are shown, allowing to take into account the hydrostatic oil film when determining the characteristics of the bearing. The results of the first tests of the bearing on the stand allow us to evaluate the performance of the design and determine further improvement and optimization of the design of the bearing with fluid pivot.

1. Introduction
During the operation of compressor and pumping equipment, conditions of rotor axial instability may arise, which may be caused by non-stationary axial load, axial vibration, and operation under off-design conditions. Under axial loads, axial displacement of the bearing occurs due to the flexibility of the alignment systems, subsidence of the pads at their contact points, which leads to unexpected stops to block axial shifts. The axial subsidence of the rotor affects the efficiency and vibrational state of centrifugal compressors. The axial subsidence has a strong effect in compressors having flow parts with narrow rotor and stator channels, and especially in compressors with back-to-back stages. An improper position of the rotor in the stator accelerates the process of fatigue failure of impellers working near the surge mode, leads to an increase in axial vibrations and forces and, as a rule, to early wear and further failure of the thrust bearing.

In connection with the above, the urgent task is to create thrust bearings to exclude:
1. The phenomena of axial power drawdown of mechanical levelling systems (Kingsbury lever-type, spring and other types).
2. Mechanical abrasion of contacting supporting surfaces.
3. Uneven load on the load-bearing elements.
4. Axial swing of the rotor.

2. The fluid pivot and the experience of its application
To create thrust bearings with the elimination of the aforementioned drawbacks, TRIZ, which has experience in improving the reliability and efficiency of dynamic equipment [1], considered the possibility of using the hydrodynamic pressure of an oil wedge to hydrostatically support bearing elements [2] in thrust bearings. Using this design will ensure high bearing and damping ability of the
axial bearing, while reducing the axial dimensions of the bearing and eliminate the axial displacement of the rotor relative to the stator under the influence of axial forces.

Previously, TRIZ developed and successfully put into operation the design of a radial bearing with fluid pivot (see Fig. 1). This technical solution made it possible to avoid mechanical contacts in the bearing and to prevent abrasion of the bearings, which significantly increased the service life of the bearing assembly. The damping properties of the hydrostatic film between the back of the pad and the bearing housing contribute to the attenuation of vibration and increase the radial stability of the rotor [2], [3]. The use of fluid pivot allowed to increase the damping coefficients by 2 times, which led to a decrease in the amplitude peaks at the first critical frequency by 2–3 times when tested on a model rotor [4], [5], [6]. The use of bearings in operating conditions on real units also allows to reduce the vibration level by 2–3 times in comparison with conventional plain bearings with mechanical support. An additional degree of freedom of the pad allows it to be set in the optimal way during operation, which allows the compensation of the angular displacement of the rotor during operation.

![Figure 1. Radial damper bearing TRIZ with fluid pivot.](image)

The above advantages of a journal bearing with fluid pivot are realized in application to thrust bearings. This allowed to provide high bearing and damping ability, reliability and durability of the thrust unit in combination with the simplicity of design.

### 3. Design of a thrust bearing with a fluid pivot

Such thrust bearing with hydrostatic bearing was developed by TRIZ for the PDA 2500-55 pump manufactured by Sumy Nasosenergomash JSC (Fig. 2). The principle of operation of this bearing is that during its operation each thrust pad rests on a self-generated hydrostatic film, which is created as a result of takeoff of a part of the lubricant flow from the hydrodynamic bearing layer on the working surface of the pad. In this case, in a pocket made on the back side of the pad, hydrostatic pressure is created.

On the working surface of the pads, where a hydrodynamic bearing layer is formed, the oil is supplied from the inter-pad space through an oil scraper when the rotor rotates. The flow of part of the oil from the hydrodynamic layer into the hydrostatic pocket leads to the fact that a hydrostatic pressure plot is created on the back of the pad, under the influence of which the pad floats and is installed in space in such a way until for a given operating mode (rotation speed, perceived force) equilibrium of forces and moments occurs between the plots of the hydrodynamic pressure Pd on the working surface and the hydrostatic pressure Ps on the back of the pad (see Fig. 3).
In the process of developing and testing the bearing at the TRIZ test stand, the problem of the formation of the initial hydrodynamic wedge at the time of starting the unit in the absence of mechanical support on the back of the pad was solved. Such a problem is absent in the journal bearings, since at the time of start-up, when the shaft lies on the pad, an initial wedge-shaped cylindrical gap already exists between the pad and the shaft neck, in which a hydrodynamic pressure plot is formed during the beginning of rotation. In order to provide an initial hydrodynamic wedge, a section in the form of a wedge with an angle of inclination $\alpha$, which is connected by an opening to a hydrostatic pocket, was made on the working surface of the pad (see Fig. 4).

**Figure 2.** Circular section of a thrust bearing with self-aligning pads with fluid pivot.

**Figure 3.** Hydrodynamic and hydrostatic pressure plots acting on the thrust pad.

**Figure 4.** Thrust pad with the wedge.
The angle of inclination $\alpha$ and the width of the wedge section are selected in such a way that when the thrust disc rotates, hydrodynamic pressure forms on the surface of the wedge section, filling the hydrostatic pocket through the bore. Thus, due to the occurrence of pressure in the hydrostatic pocket, a tilting moment acts on the pad, which sets the pad at an angle to the bearing surface of the thrust disc, and the shape of the gap between the disc and the pad contributes to the formation of an optimal oil wedge on the entire working surface of the pad.

After the formation of the initial wedge, the pressure in the hydrodynamic layer increases. Upon reaching a certain pressure in the pocket, the pad pops up and oil from the hydrostatic pocket is throttled through the end gap between the back of the pad and the bearing housing. In the steady state operation mode, the flow rate of oil throttled from the hydrostatic pocket is equal to the flow rate of the lubricant entering the pocket from the hydrodynamic layer through the bore in the inclined section in the pad.

Previously, TRIZ engineers developed a design of a thrust bearing with a fluid pivot, where for the formation of the initial hydrodynamic wedge during the start-up of the unit, a cylindrical rubber band was installed along the outlet edge at the back of the pad, the mechanical properties and section sizes of which were chosen so that when there is a load on the pad from the side of the thrust ridge, the gum is deformed so that the shape of the gap between the pad and the thrust ridge contributes to the formation of an optimal oil wedge [7]. Currently, to increase the reliability and simplicity of the bearing design, engineers have abandoned this option in favor of a new design with an inclined section on the working surface of the pad.

4. Calculation model of the bearing

In connection with the development of a new design of a thrust bearing with a fluid pivot, it became necessary to create a calculation model that allows us to investigate the ascent of the pad and determine the pressure plots of the hydrodynamic and hydrostatic layer with its various geometric parameters. To do this, it was necessary to make adjustments to the existing calculation methods [8], [9], allowing to evaluate the effect of fluid pivot on the operating parameters of the thrust pad.

When determining the plot of hydrostatic pressure on the back of the pad, an assumption was made about the constancy of pressure in the pocket and the linearity of the law of pressure change in the end gap. The resultant hydrostatic pressure plot is the response of the hydrostatic layer to the load acting on the pad from the hydrodynamic layer side. In traditional thrust bearings, this reaction is transmitted to the pad through the rocking rib, and together with the resultant plot of hydrodynamic pressure creates a moment of force, which turns the pad, forming a bearing oil wedge. As is known, the ratio of the thickness of the oil film at the inlet and outlet of the pad depends on the position of the rocking rib, which has a significant effect on the pressure plot, bearing capacity and other most important characteristics of the hydrodynamic oil layer [10], [11].

The calculation model is reduced to determining the magnitude of the resultant plot of hydrostatic pressure on the back of the thrust pad and the coordinates of the point of its application. In the case of fluid pivot, an analogy is made with a support point (in the form of a sphere), when both coordinates of the resultant hydrostatic pressure, circumferential and radial, influence the formation of a bearing oil wedge. If the radial position of the support is improperly selected, the balance between the plots of hydrodynamic pressure in the upper and lower parts of the pad can be disturbed, while some of its working zones can be overloaded, which leads to a decrease in bearing capacity, increased wear and early failure of the pad.

The coordinates of the resultant reaction of the hydrostatic layer are the coordinates of the center of gravity of the pressure plot and are determined by the position of the hydrostatic pocket. Therefore, when designing fluid pivot, the position of the pocket was determined depending on the results of integration of the two-dimensional pressure plot arising at this position of the pocket and the calculation of the coordinates of its center of gravity using the formulas:

$$x_C = \frac{\iint_{S} P_b (x, z) x dS}{\iint_{S} P_b (x, z) dS}, \quad z_C = \frac{\iint_{S} P_b (x, z) z dS}{\iint_{S} P_b (x, z) dS},$$

(1)
where \( p_d(x, z) \) is the plot of hydrostatic pressure, \( S \) is the surface of the back of the thrust pad.

The circumferential coordinate of the resultant hydrostatic pressure \( x_r \), and hence of the hydrostatic pocket, was determined from the condition of the maximum load-bearing capacity of the hydrodynamic layer, since the circumferential position of the pocket, forming the plot of hydrostatic pressure equivalent to the optimal position of the swing rib, requires its displacement towards the outlet edge of the pad.

The radial coordinate of the resultant hydrostatic pressure was determined from the condition of ensuring equal moments relative to its axis of the pressure plots of the hydrodynamic layer, respectively, lower and higher than the radius of the point of application of the resultant.

To determine the pressure plot in the working hydrodynamic layer, analytical expressions were used [9]. The dimensionless pressure function in the layer was specified as

\[
q = q_{pc}(\eta) \varphi_p(\zeta),
\]

where

\[
q = p/p^0; p^0 = 6\mu U L \eta h_2, \quad \mu \text{- lubricant viscosity, } U \text{- peripheral speed at the medium radius, } L \text{- pad length at the medium radius, } h_2 \text{- thickness of the lubricant layer at the outlet of the pad.}
\]

The coordinates given in formula (2) are the relative width of the pad \( \zeta = z/B \), where \( B = r_2 - r_1 \) - pad width, \( r_2 \) and \( r_1 \) are the outer and inner radius of the pad, respectively, lower and higher than the radius of the point of application of the resultant.

The \( q_{pc} \) function is a solution for an infinitely wide pad:

\[
q_{pc} = [\eta^{-1} - k(k+1)^{-1} \eta^{-2} - (k+1)^{-1}] (k-1)^{-1}
\]

The \( \varphi_p \) parameter considers the side leakage of lubricant from the layer and looks as follows

\[
\varphi_p = 1 - \text{ch} \left(2\lambda_p \zeta \right) / \text{ch} \left(\lambda_p \right);
\]

where

\[
\lambda_p = 0.5 \psi_p B/L; \quad \psi_p = \sqrt{2(k^2 - 1) \Phi_{pc}/(1 - k \Phi_{pc})};
\]

\[
\Phi_{pc} = 6[\ln(k) - 2(k-1)/(k+1)]/(k-1)^2;
\]

\[
\Phi_{pc} = 6[k^3 - 1 - 2k \ln(k)]/(k-1)^2(k+1).
\]

This technique completely solves the problem of determining the pressure plot in the hydrodynamic layer, knowing which, the coordinates and the maximum oil pressure in the bearing capacity of the working layer can be found. In order to ensure effective ascent of the pad during the design of fluid pivot, the center of the bore in the pad for lubricant takeoff from the hydrodynamic layer into the hydrostatic pocket was located at the calculated maximum of the hydrodynamic pressure plot and therefore was displaced relative to the center of the hydrostatic pocket, the algorithm for determining which was described above.

The coordinate of the center of pressure in the hydrodynamic layer is determined by the formula

\[
x_{pc} = \left[k/(k-1) - 0.5 \Phi_3 / \Phi_1 \right] L,
\]

where

\[
\Phi_3 = \Phi_{pc} k_1; \quad k_1 = 1 \cdot \text{th} (\lambda_3)/\lambda_3;
\]

\[
\lambda_3 = 0.5 \psi_3 B/L; \quad \psi_3 = (k - 1) \sqrt{J_1} / \lambda_3;
\]

\[
J_1 = \left\{ \frac{k^4 - 1}{32} + \frac{k^2 \ln k}{(k^2 - 1)^2} - \frac{3}{8} \right\};
\]

\[
\Phi_{pc} = \Phi_{pc} k_0; \quad k_0 = 1 \cdot \text{th} (\lambda_0)/\lambda_0.
\]

Together with the task of determining the location coordinates of the bore and the hydrostatic pocket, the problem of determining their sizes, based on the equilibrium condition, was solved:

\[
\iint_{(S)} p(x, z) dS = \iint_{(S)} p_b(x, z) dS,
\]
where \( p(x,z) \) is the pressure plot in the hydrodynamic lubricating layer from the working side of the pad, \( p_h(x,z) \) is the pressure plot in the hydrostatic layer on the back side; whilst considering the equation of the balance of flow rates:

\[
Q_o = Q_s, \tag{8}
\]

where \( Q_o \) is the lubricant consumption through a bore in the pad from a hydrodynamic layer in a hydrostatic pocket, \( Q_s \) is the side leakage of lubricant from a hydrostatic pocket.

To determine the flow rate \( Q_o \), the hydraulic loss formula was used when throttling oil through the bore:

\[
Q_o = \frac{\pi d_o^2}{4} \sqrt{\frac{p_{\text{max}} - p_k}{\zeta_c \rho}}, \tag{9}
\]

where \( d_o \) is the diameter of the bore in the pad, \( p_{\text{max}} \) is the maximum oil pressure in the hydrodynamic working layer, \( p_k \) is the oil pressure in the hydrostatic pocket, \( \zeta_c \) is the hydraulic resistance coefficient of the bore, \( \rho \) – lubricant density.

\( Q_s \) flow rate is defined by the formula

\[
Q_s = \int_{-\frac{h_T}{2}}^{\frac{h_T}{2}} h_T d\theta \frac{\mu_l n_{\psi_p}^2}{r_k^2}, \tag{10}
\]

where \( r_k \) is the radius of the hydrostatic pocket, \( r_e \) is the distance from the center of the hydrostatic pocket to the outer contour of the pad from the back, forming an end gap; \( h_T \) is the end clearance between the back of the pad and the bearing housing. The last two values depend on the current value of the angle \( \theta \).

The dimensions of the bore and pocket were determined in several iterations. Initially, for a given pocket diameter, the pressure in the pocket \( p_k \) was determined from the equilibrium condition (7). Then, the pressure plot in the hydrodynamic layer was calculated and the maximum pressure \( p_{\text{max}} \) was found, after which the oil flow rate through the hydrostatic pocket and the value of the pad "ascent" were determined from the flow balance equation (8) as the end gap between the pad and the bearing housing, at which the flow balance is achieved. The influence of additional leaks from the hydrodynamic layer on its bearing capacity was estimated as follows. First, the value of lateral leaks from the wedge was determined in the absence of flow towards the hydrostatic pocket according to the formula [3]:

\[
Q_{\text{lateral}} = 0.5(k-1)(1-k_p)BU h_2 \psi_p^2/12. \tag{11}
\]

Then, flow through a hydrostatic pocket calculated by formula (9) was added to this value. After that, from equation (11), a new value of \( \psi_p \), was determined, which was then used in formula (5) when calculating a new pressure plot in the hydrodynamic layer.

Using this algorithm, after a series of iterations, the optimal dimensions of the bore in the pad and the hydrostatic pocket were obtained, which ensure the ascent of the pad and the operation of fluid pivot with minimal leakage from the hydrodynamic lubricating layer. These leaks also contribute to the cooling of the pads, as they pass through the bore, which is located near the thermally loaded area of the working surface of the pads.

5. Experimental studies of the bearing

At the moment, the design of the thrust bearing with fluid pivot has passed the first tests on a test stand (Fig. 5).

The tested thrust bearing consists of 8 self-aligning pads and oil scraper mounted in the inter-pad space. In parallel with the hydrostatic bearing, a thrust bearing with a mechanical spherical bearing and lever leveling system in the same overall dimensions is tested (Fig. 6). The bearing pads of the two types of tested bearing are made of steel with a layer of babbit applied to the working surface. TRIZ thrust bearings with self-aligning pads with mechanical support along the sphere have an increased bearing capacity due to the use of rollers in the lever levelling system, where the rolling friction coefficient is several times less, the sliding friction coefficient between the levers of the traditional system, as well as installation in the inter-pad space of the oil scrapers, which provides an individual drainage of hot oil coming from the previous pad, an individual supply of lubricant to the
working surface of the pad. The bearing capacity of such bearing is 1.5-2 times higher than the traditional one [12]. Traditional thrust bearings mean thrust bearing design with a Kingsbury lever levelling system, the organization of the lubrication supply to the bearing by means of an oil bath without the removal of hot oil.

![Figure 5. TRIZ test stand: a) side view; b) thrust disc before the installation of the bearing.](image)

The results are processed and compared for two bearings.

![Figure 6. Thrust bearing: a) bearing with the fluid pivot; b) bearing with mechanical support; c) self-aligning pads of two bearings.](image)
The operating parameters of the stand during the experiment:
Rotation speed - 5000 rpm;
Oil supply pressure - 1.2 bar;
Oil consumption through the bearing - 78 l/min;
Oil supply temperature: 40-43 °C;
Axial force on the bearing: 0 - 7000 kgf;
Place of installation of the temperature sensor in the pad - thermally loaded zone near the outlet from the pad, in the direction of rotation;
The place of installation of the pressure sensor is the cavity of the hydrostatic pocket on the back side of the thrust pad;
Location of the shaft end displacement sensor - thrust bearing end cover.

**Figure 7.** Test bench readings during testing of the thrust bearing with fluid pivot.

As can be seen from the test results, the bearing temperatures are comparable (see Fig. 8). The axial displacement (subsidence) of the bearing with fluid pivot is less than that of a mechanically supported bearing with a lever levelling system, which is due to the absence of this system in the fluid pivot bearing (see Fig. 9).

The repeatability of the experimental results was ensured through multiple tests. Differences between the results of individual experiments amounted to 1-2 percent.

The damping ability was not evaluated in this experiment. Registration of the pressure in the hydrostatic pocket, the movement of the shaft end, confirm that a hydrostatic pressure plot is formed on the back of the thrust pad (see Fig. 10), under which the pad ascents and is installed in space, like a pad with an offset mechanical support in the circumferential direction by rotation of the thrust disc. An oil film has formed on the back side of the pad, by analogy with the radial support with hydrostatic support, and has additional damping.

The repeatability of the experimental results was ensured through multiple tests. Differences between the results of individual experiments amounted to 1-2 percent.
The temperature mode of the pad showed that the bearing with thrust pads with a fluid pivot has a high bearing capacity comparable to the high bearing capacity of a thrust bearing with self-aligning pads with a mechanical bearing.

In order to achieve all the tasks set, thrust bearings reversible and non-reversible, pad and ring thrust bearings with a flat and spherical bearing were also developed, manufactured and successfully preliminarily tested, which exclude:

1) The phenomenon of axial force drawdown;

**Figure 8.** Comparison of bearing temperature conditions.

**Figure 9.** Comparison of axial movement of bearings.
2) Mechanical abrasion of contacting supporting surfaces;
3) Uneven load on the load-bearing elements;
4) Swing of the rotor by effectively damping axial vibrations by hydrostatic damping.

![Figure 10. Change in oil pressure in the hydrostatic pocket during changes in axial force.](image)

Further tests will provide the necessary information to create a unified series of thrust bearings, optimization of designs and design models of bearings.

6. Conclusion

Thus, the developed design of the thrust bearing with a fluid pivot allows to eliminate the disadvantages of traditional thrust bearings with mechanical support and provide the bearing with high load-capacity and additional damping capacity, while reducing the axial dimensions of the bearing.

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