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Work process calculation of rotary hybrid energy converting displacement machines

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Abstract

The paper presents the design of rotary hybrid displacement machines with a high indicator isothermal efficiency coefficient, with compressor section improved efficiency and reduced materials consumption. A mathematical model of the hybrid machine pump and compressor section work processes has been developed. The results of the indicator diagrams comparison obtained by computational and experimental methods are presented. In conclusion a schematic diagram of the rotary hybrid displacement machines with hydrodiode is given.

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1. Introduction

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The improvement of displacement compressors and pumps efficiency and effectiveness is the urgent issue for many years. A significant amount of electric energy (up to 10% of the total generated electricity) is spent on the stationary compressors drive. Despite the fact that the displacement pumps and compressors are widely spread in the automotive engineering, the food industry, shipbuilding etc., they are mostly used in the petrochemical industry. In the industry both pressure gas and liquid are required which causes the simultaneous use of compressors and pumps.

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One of the basic methods to upgrade the effectiveness and efficiency of the displacement compressors (piston, screw, rotation, etc.) is to improve the cooling of the compressed gas and to approximate the compression process to the isothermal one as the most thermodynamically efficient. The advantages analysis of the coolant use in displacement compressors has revealed that the film cooling on the operating chamber surface is one of the effective methods of compressors cooling.

This idea can be implemented in the pump and compressor operations in a one power generating unit. In this case, there is a new class of displacement machines - "pump-compressor". The pump and compressor functions combining in one power generating unit allows to organize intensive cooling of the working elements, cylinder-piston group positive seal and to reduce friction work in the cylinder-piston seal due to the reliable fluid friction.

The result is an increase of the indicator isothermal efficiency coefficient (up to 5% due to the intensive cooling), an increase in the compressor capacity (up to 10% due to the cooling and cylinder-piston group seal), 30% material consumption reduction and 5-7 % friction work reduction.

Based on the patent analysis a new structural scheme of rotary hybrid displacement machines (RHDM) is proposed, having two separating plates and two displacement volume: gas and liquid (Fig. 1).

2. Methods

During one rotor revolution the gas is sucked into the compressor chamber 12, the gas is compressed and injected into the compressor chamber 2, the liquid is sucked through the valve 8 and the liquid is injected through the valve 10. Thus, in the structure an intensive cooling of the rotor 15, separation plates 6, 7 and all the working chamber of the compressor section (CS) occurs due to the pump section (PS). In addition, PS provides intensive lubrication of the wearing surface of separation plates 6, 7 and the rotor 15 and carries out the compressed gas positive seal, as there is some hydraulic seal between the discharge line and the suction line of the CS.

![Fig.1. Schematic diagram of RHDM: 1. Body. 2. Compressor injection-compression chamber (CS). 3. Compressor chamber discharge valve. 4, 5. Compression springs. 6, 7. Separation plates. 8. Suction valve of the pumping chamber. 9. Pumping chamber (PC). 10. Discharge valve of the pumping chamber. 11. The gas inlet. 12. The compressor suction chamber. 13. The drive shaft. 14. The crank (eccentric). 15. The shareable rotor.](image-url)
900 angularity of the separation plates 7 is shown schematically, this parameter is determined by the flow requirement of liquid and gas, as well as by a number of other requirements. For each consumer RHDM can be produced with the desired pump and compressor volumetric capacity ratio. This circumstance is an important advantage of the design. Another significant advantage is its high compactness, since the drive shaft is the part of the drive mechanism. Furthermore, in the design the contact surface deterioration in "plate – rotor" does not lead to the interface seal deterioration opposed to o-ring seals wear (in reciprocating layouts of RDM) with the joint disclosing and gradual seal loss.

The RHDM has both functional capabilities and technical features that have not previously been studied, such as the work processes in the gas and liquid chamber of the device. They define the PS and CS productivity and efficiency of the hybrid machines.

Based on the current mathematical modeling methods for work processes of the piston [1, 2, 3] and the rotary compressor [4, 5], as well as mathematical modeling for work processes of the piston pump [6] a mathematical model of CS and PS work processes of the RHDM has been developed.

The mathematical model is based on fundamental laws of energy, mass and motion conservation for gas and liquid, as well as automatic valves gates. Given the small geometric dimensions of the machine working chambers and the significant rate of small vibration transmission in the gas and liquid environment we can assume after [2, 3, 4] with a sufficient degree of accuracy that the investigated processes are reversible and in equilibrium and a model with lumped parameters can be used for the study.

A RHDM has rather complicated dependence of work chamber volume on the rotor shaft rotation angle opposed to reciprocating and conventional rotary compressors. To determine the CS and PS volume change law of the rotor rotation angle \( \varphi \) in the suction and discharge process, the rotor movement is divided into two phases [7]. The first phase, the contact point with the rotor cylinder bore is in the compressor working chamber or \( \alpha < \varphi < 2\pi \) (Fig 2a). The second phase, when the contact point with the rotor cylinder bore is in the pump working chamber, i.e. \( 0 < \varphi < \alpha \) (Fig. 2b).

Fig.2. The calculation scheme for volume change law determining: \( V_{\text{s,c}}^{\text{phase 1}} \) is the suction chamber volume of the compressor section; \( V_{\text{d,c}}^{\text{phase 1}} \) is the compression chamber (discharge) volume of the compressor section; \( V_{\text{p}}^{\text{phase 1}} \) is the pump section volume in suction or discharge process; \( V_{\text{s,p}}^{\text{phase 2}} \) is the suction chamber volume of the pump section; \( V_{\text{d,p}}^{\text{phase 2}} \) is the discharge chamber volume of the pump section; \( V_{\text{s,c}}^{\text{phase 2}} \) is the compressor section volume during suction process.
Suction chamber CS volume changing in phase 1 is defined as the algebraic sum of the following volumes (Fig. 2a):

\[ V_{\text{phase}1}^\alpha = V_{\text{BCDEFGIB}} + V_{\text{JIGJ}} - V_{\text{BCHIB}}, \]  

(1)

Given that the geometrical dimensions of the plates 1 and 2 are the same, VBCDEFGIB volume is defined as:

\[ V_{\text{BCDEFGIB}} = B \cdot \int_{\phi-\alpha}^{\phi} \int_{e \cos \phi}^{R_p} \rho \, d\rho \, d\phi, \]

(2)

where B is the rotor width, \( e \) is the eccentricity, \( R_p \) is the rotor radius, \( R_u \) is the cylinder radius. After reexpressions we obtain:

\[ V_{\text{BCDEFGIB}} = \frac{1}{2} \cdot B \cdot \left\{ R_u \cdot (\phi - \alpha) - \left\{ \frac{1}{2} (\phi - \alpha) + \frac{1}{4} \sin[2(\phi - \alpha)] \right\} e^2 - \right. \\
- e^2 \sin(\phi - \alpha) \left( \frac{R_p^2}{e} \right)^2 - \sin^2(\phi - \alpha) - \\
\left. - R_p^2 \cdot \arcsin \left( \frac{e \cdot \sin(\phi - \alpha)}{R_p} \right) \right\} \\
- R_p^2 \cdot (\phi - \alpha) + \left\{ \frac{1}{2} (\phi - \alpha) - \frac{1}{4} \sin[2(\phi - \alpha)] \right\} e^2 \right\}, \]

(3)

To determine \( V_{\text{BCHIB}} \) volume of the separation plate 2, the Cartesian coordinate system center is associated with the lower plane of the separation plate, and the Y-axis is directed along the symmetry axis of the separation plate. Then:

\[ V_{\text{BCHIB}} = \frac{1}{2} B \cdot \int_{-h/2}^{h/2} \int_{-h/2}^{h/2} dx \int_{y}^{R_u + e[1 - \cos(\phi - \alpha)]} dy \]

(4)

where h is the thickness of the separation plate.

After integration (4) we obtain:

\[ V_{\text{BCHIB}} = \frac{1}{2} B \cdot h \cdot \left\{ e[1 - \cos(\phi - \alpha)] - R_u + \right. \\
\left. \left\{ \frac{1}{2} \sqrt{R_u^2 - \frac{h^2}{4}} + R_u \cdot \arcsin \left( \frac{h}{2R_u} \right) \right\} \right\}, \]

(5)

Additional volume \( V_{\text{JIGJ}} \) is defined as:
\[ V_{JIG} = B \int_{x_1}^{x_2} \int_{y_1}^{y_2} dx dy \]  

(6)

Or:

\[ V_{JIG} = B \left\{ R_p \sin(\varphi - \alpha) \cdot \frac{e \sin(\varphi - \alpha)}{2} \sqrt{R_p^2 - e^2 \sin^2(\varphi - \alpha)} - \frac{R_p^2}{2} \arcsin \left[ \frac{e \sin(\varphi - \alpha)}{R_p} \right] \right\} \]

(7)

Substituting the results of the solution (3), (5) and (7) into equation (1) yields an expression for the determination of the volume change law of the suction chamber SC in phase 1, all other volumes in the phase 1 and phase 2 being determined in a similar way.

In constructing the mathematical model of SC work processes the following key assumptions have been made: the gaseous environment is continuous and subject to the ideal gas laws; radial leakage and gas flow in compressor section work chambers are absent; clearance volume is absent; liquid film is uniformly distributed over the elements surface, the break of liquid film in rotating elements is absent, the speed of film movement is equal to zero; the presence of the oil film in the compressor chamber has no practical effect on the compressible gas properties; gas phase velocity is equal to the rotor circumferential velocity; separation plates are constantly pressed to the rotor.

The system of basic MM equations MM of SC work processes can be written as:

\[
\begin{cases}
    dU = dQ_{km} - dI_k + \Sigma_i \cdot dM_{ki} - \Sigma_i \cdot dM_{ko} \\
    dM_k = \sum dM_{kn} - \sum dM_{ko} \\
    V_k = f(\varphi) \\
    m_{knp} \frac{d^2 h_{knl}}{dt^2} = F_{kc} + F_{knp} + F_{kc} + m_{knp} \cdot g \\
    p_k V_k = (k - 1)U \\
    T_{ki} = \frac{p_k V_k}{M_k R}
\end{cases}
\]

(8)

where \( p_k, T_k, V_k, M_k \) are pressure, temperature, volume and mass of the compressed gas; \( i_{k}, i_{ko} \) are specific enthalpy of added and separated gas mass; \( dM_{kn}, dM_{ko} \) are elementary added and separated gas mass due to external mass exchange; \( dU \) is elementary change in the gas internal energy; \( dL_k = p_k (dV_k - dV_k) \) is the elementary contour work, taking into account the geometric change in the work chamber volume of the compressor section by moving the working element and fluid flow from the pump section; \( dV_k \) is the elementary change in the compressor section volume; \( dV_k \) is fluid flow elementary volume; \( dQ_{kc} \) is elementary heat flow between the gas and the work chamber walls; \( m_{knp} \) is the shut-off valve mass of the obturator; \( h_{knp} \) is the current vertical head of the shut-off valve; \( F_{knp} \) is gas power, \( F_{knp} \) is the spring elastic energy, \( F_{kc} \) is shut-off valve resistance force.

In constructing the mathematical model of NC work processes the following key assumptions have been made: the working fluid is a compressible dropping liquid following the friction Newton’s laws and Hooke's law; the gap between the plate side surface and the side cover and the gap between the rotor and the side cover are always filled.
with liquid; fluid velocity in the pump chamber is equal the circumferential speed of the rotor; clearance volume in the pump section is negligible; the kinetic energy of the working fluid in the compression process is neglected.

The system of mathematical model basic equations includes the mass conservation equation, the energy conservation equation in the form of the Bernoulli equation to the suction and discharge processes, the dynamic equation of pump section gate automatic valves. The complete operating cycle of the pump section is divided into the following processes: compression, discharging, suction:

**Compressing process**

\[
\begin{align*}
V_{ni} &= f(\varphi) \\
\sum_{i=1}^{n1} dV_{ni} &= \sum_{i=1}^{n2} dV_{ni} + \sum_{i=1}^{n2} dV_{ni} \\
\frac{dp_{ni}}{V_{ni}} &= \frac{E_{\text{ac}} \cdot dV_{\text{ac}}}{V_{ni}} \\
\end{align*}
\]

\(9\)

**Discharging process**

\[
\begin{align*}
V_{ni} &= f(\varphi) \\
\nu_{n2} &= \frac{\nu_{ni} f_{1-\tau} \cdot d\tau - dV_{\text{sym}} - dV_{\text{ac}} - dV_{\tau1} - dV_{\tau2}}{f_{1-\tau} \cdot d\tau} \\
p_{ni} &= p_{nu} + \rho_{\infty} \left( \alpha_2 \frac{\nu_{n2}^2}{2} - \alpha_1 \frac{\nu_{n1}^2}{2} \right) + \left( \Delta h_{c_{n2}} + \Delta h_{l_{n2}} + \Delta h_{u_{n2}} \right) \rho_{\infty} g \\
m_{\text{up}} \frac{d^2 h_{\text{ac2}}}{d\tau^2} &= F_{\text{up}} + F_{\text{up}} + F_{w} + F_{\text{wg}} \\
\end{align*}
\]

\(10\)

**Suction process**

\[
\begin{align*}
V_{ni} &= f(\varphi) \\
\nu_{n4} &= \frac{\nu_{ni} f_{1-\tau} \cdot d\tau + dV_{\text{ac}} + dV_{\tau1} + dV_{\tau2} + dV_{\tau\text{cb}}}{f_{1-\tau} \cdot d\tau} \\
p_{ni} &= p_{nu} + \rho_{\infty} \left( \alpha_1 \frac{\nu_{n1}^2}{2} - \alpha_1 \frac{\nu_{n1}^2}{2} \right) + \left( \Delta h_{c_{n1}} + \Delta h_{l_{n1}} + \Delta h_{u_{n1}} \right) \rho_{\infty} g \\
m_{\text{ac}} \frac{d^2 h_{\text{ac1}}}{d\tau^2} &= F_{\text{ac}} + F_{\text{ac}} + F_{w_{ac}} + F_{\text{wg}} \\
\end{align*}
\]

\(11\)

where \(dV_{\text{ac}}\) is the elementary change in the pumping section volume by the kinematics of motion mechanism; \(dV_{\text{ac}}\) is the elementary change in the working chamber volume of the pump section due to leaks and flow of working fluid, respectively; \(n1, n2\) are the number of discharges and sources of the working fluid respectively; \(m_{\text{up}}\) is the gate reduced mass; \(h_{\text{sym}}\) is the current vertical head of the gate valve; \(F_{\text{up}}\) is the flow resistance power; \(F_{\text{up}}\) is the spring elastic force; \(F_{w}\) is the gate weight force; \(F_{w_{ac}}\) is the gate resistance force; \(F_{w_{ac}}, F_{w_{ac}}, F_{w_{ac}}, F_{w_{ac}}, m_{\text{ac}}\) are the suction valve forces; \(p_{ni}\) is the current pressure in the pump section working chamber; \(E_{\infty}\) is the fluid pressure bulk modulus; \(p_{\text{ac}}\) and \(p_{\text{ac}}\) are the pressure in the suction and discharge pipes, respectively; \(\Delta h_{c_{n1}}\) is the pressure loss at sudden contraction; \(\Delta h_{c_{n2}}\) is the pressure loss at sudden expansion; \(\nu_{n1}\) is the fluid velocity in the II-II section (after discharge and suction valve, respectively); \(\nu_{n2}\) is the fluid velocity in
the section I-I (in the pump working chamber); $\rho_{sc}$ is the fluid density; $\Delta h_{l_{w}}, \Delta h_{\text{sc}_{w}}, \Delta h_{l_{n}}, \Delta h_{\text{sc}_{n}}$ are the friction loss and inertial loss of discharge suction processes, respectively; $dV_{T_1}^{''}, dV_{T_2}^{''}$ is the elementary volume of the working fluid entering the pump section working chamber through the gaps formed by the plate and side recess, respectively, for plate 1 and plate 2 (in case the discharged fluid is not supplied under the plate); $dV_{T_1}^{''} + dV_{T_2}^{''} = 0$; $dV_{T}^{AB}$ - are the end leakage through the gaps formed by the rotor and end covers in the area between the rotor and plates contact points; $dV_{\text{sc}_{w}}, dV_{\text{sc}_{n}}$ is the elementary volume of fluid passing through the leakiness of the discharge and suction valves, respectively; $dV_{\text{sym}}$ is the elementary volume of fluid passing through the end and radial clearances; $f_{I}, f_{II}$ is the square of the I-I and II-II cross-sections, respectively (cross-section I-I is carried out through the minimum distance between the rotor and the cylinder, and the cross section II-II is carried out through the discharge (discharging process) or suction (suction process) pipe).

3. Results and discussion

The system of differential equations given above has no analytical solution. As a result, we use numerical methods for the ordinary differential equations solution. The Cauchy problem is solved with the following initial conditions: for the compressor section: $\varphi=0$; $p_{ki}=p_{kv}$; $p_{ni}=p_{nv}$; $T_{ki}=T_{kv}$; for the pump section: $\varphi=0$; $p_{ki}=p_{kv}$; $p_{ni}=p_{nv}$; $T_{ki}=T_{kv}$.

Fig. 3 shows the obtained indicator diagrams of one alternative of the RHDM work processes numerical simulation at the following values of machine design and operational parameters $\alpha=900$, $p_{kv}=0.1$ MPa, $p_{nv}=0.3$ MPa, $p_{kv}=0.1$ MPa, $p_{nv}=0.6$ MPa, $\varphi_{p}=0.043$, $K_p=0.217$, $n_{ob}=600$ r/min, $T_{st}=300K$. The indicator diagrams obtained experimentally are presented for comparison.

Thus, these data allow us to draw the following conclusion. The RHDM mathematical model reasonably describes the actual physical processes occurring in the RHDM chambers, the difference between the calculated and experimental data does not exceed 9.8%. This makes it possible to use the mathematical model to analyze the impact of design and operational parameters on RHDM operation.

![Fig.3. Comparison of the indicator diagrams of compressor and pump chambers obtained by: 1. computational method, 2. experimentally](image)
RHDM parametric analysis of the proposed constructive solutions and experimental studies [8, 9] have revealed contradiction due to the fact that in order to ensure a high efficiency of rotary machines operating with rolling rotor operate with a high rotor speed (2000-3000 r/min), while the fluid pumps operate with a frequency less than 400-750 r/min due to the high density of the working fluid the small switchgear flow area. To improve the pump section efficiency it seems appropriate to use switchgear having no moving parts and allowing to provide different resistance to the fluid flow as it moves forward and backward [10]. Schematic diagram RHDM with hydrodiodes as PS switchgear is presented in Fig. 4. It should be noted that the installation of separation plates with the flare angle of 180º is preferable (than 900), which is primarily caused by balancing of the inertia forces acting on the separation plates on the rotor side.

Fig.4. Schematic diagram of RHDM with hydrodiodes: 1. compensating tank. 2. Liquid. 3. inlet strainer. 4. Cooling system. 5. Body. 6. Suction gas filter. 7. Oil separator. 8. Pressure gas consumer. 9. The pressure regulator. 10. Compressor chamber of discharge-compression (CS). 11. The compressor chamber pressure valve. 12, 13. Compression springs. 14, 15. Separation plates. 16. The suction box of the pumping chamber. 17. The pumping chamber (PS). 18. The discharhe pump chamber box. 19. The gas inle. 20. The compressor suction chamber. 21. The drive shaft. 22. The crank (eccentric). 23. Shareable rotor. HD - hydrodiodes.

Hydrodiode installation (one for pump chamber inlet and the other one at the outlet) will allow to form a unidirectional fluid flow in the compressor cooling system, sufficient to remove the compression heat or fluid circulation in any process loop.

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