Problems equilibration of aggregates on the basis of slow moving stages

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Abstract. The study of new design solutions in the field of reciprocating compressor units led to the creation of units based on low-speed long-stroke compression stages. Regime and design parameters of such units, along with the use of linear drives do not allow to use a flywheel as a means of reducing fluctuations in the load on the drive mechanism. A number of schemes for such units has been proposed, the most successful of which is a design with two pistons moving towards each other. This design allows us to fully balance the unit. In addition, this design has a number of additional advantages, such as increased service life and improved performance of the workflow.

1. Introduction

In the dynamic study and calculation of machines of cyclic action, which also include reciprocating compressor units, the question of the balance of the entire unit is of great importance.

In most machines, the torque on the shaft at varying shaft speeds is variable [1]. In all machines, when the rotational speed changes, the dynamic pressures in kinematic pairs change, and, consequently, the friction forces in them change. In working machines, when the rotational speed of the drive shaft changes, the production resistances, the environment resistance, etc. [2] changes.

The dependence of the moment M applied to the driven shaft of a motor-machine or to the drive shaft of a working machine on the angular velocity of these shafts is called the mechanical characteristic of the machine. Thus, mechanical characteristics are dependencies of the form \( M = M(\omega) \) or \( M = M(n) \), where \( n \) is the rotational speed, measured by the number of revolutions of the machine shaft per minute, equal to \( n = 30 \omega / \pi \) [2].

The forces of inertia of a link making plane-parallel motion and having a plane of symmetry parallel to the plane of motion can be summarized to the force of inertia \( I_s \) applied at the center of mass of the link and to a pair of inertia forces whose moment is \( M_i \).

Force \( F_i \) is the vector of inertia of the link, can be determined by the formula [2]:

\[
F_i = -m \cdot a_s
\]  

where the mass \( m \) is the mass of the link, kg;

\( a_s \) — vector of full acceleration of the center of mass of the link, \( m/s^2 \).

Thus, to determine the inertia force \( F \) and the link of a plane mechanism, one must know its mass \( m \) and the vector of full acceleration as of its center of mass or the projection of this vector on the coordinate axes. Separately, you should pay attention to the average speed of the machine and its coefficient of non-uniformity of motion. The coefficient of non-uniformity of motion characterizes only the difference in the angular velocity of the initial link in the range from \( \omega_{min} \) to \( \omega_{max} \), but does not characterize the dynamics of the motion of this link within one complete cycle of the period of steady motion. The dynamic characteristics of the mechanisms with these values of \( \epsilon \) are different. A comparative assessment of the dynamic properties of a mechanism or machine during a period of
steady motion can be characterized by the dynamic factor $K$, which takes the ratio of the largest value of the angular acceleration next to the square of the average angular velocity $\omega^2$.

$$K = \frac{\varepsilon_{ext}}{\omega^2_{ang}} = \frac{a}{\omega^2_{ang}}$$  \hspace{1cm} (2)

As mentioned above, during steady-state motion, in the general case, the motion of the initial link of a mechanism or machine occurs at a variable speed. These fluctuations in the velocity of the initial link cause fluctuations in the velocities of all the other links of the machine mechanisms, which leads to an increase in the dynamic loads on their links and kinematic pairs. In addition, most of the processes for which the mechanism or machine is used require certain speeds of the working bodies, which is achieved only if the initial link of the mechanism or machine does not have any large deviation of the magnitude of its speed from the target.

The movement of the initial link is the closer to uniform, the larger is the reduced moment of inertia or the reduced mass of machine mechanisms. An increase in the summarized masses or summarized moments of inertia can be done by increasing the masses of the individual links of the mechanisms. In practice, this increase in mass is carried out by landing on one of the shafts of the machine of an additional part having a specified moment of inertia. This part is called the flywheel or flyer. The task of the flywheel is to reduce the amplitude of the periodic oscillations of the velocity of the initial link, due to the properties of the mechanisms of the machine itself, or periodic changes in the ratios between the magnitudes of the driving forces and the resistance forces. By selecting the mass and moment of inertia of the flywheel, you can make the initial link of the mechanism or machine move with a predetermined deviation from some of its average speed.

A flywheel is like an accumulator of the kinetic energy of the mechanisms of a machine, accumulating it during their accelerated motion and giving it back when it slows down. In some machines, in which the payload periodically varies within considerable limits, the flywheel accumulates very significant reserves of kinetic energy during accelerated motion with decreasing values of payloads. This accumulative role of the flywheel allows you to use the accumulated energy to overcome the increased payloads without increasing engine power. So for example, consider a piston stage with an electric motor and the following parameters: $P_{suc} = 0.1 \text{ MPa}$; $P_{dis} = 10 \text{ MPa}$; piston stroke $0.2 \text{ m}$ and capacity $-3 \text{ m}^3 / \text{ h}$. Then, at an engine rotational speed of $n = 1 \text{ rev / s}$ (low-speed scheme), the flywheel for such a unit will have a diameter of $1 \text{ m}$ and weigh about $200 \text{ kg}$; at an engine speed of $n = 25 \text{ rev / s}$ (high-speed scheme) and with all other conditions being equal, a flywheel with a diameter of $0.5 \text{ m}$ will weigh about $3 \text{ kg}$.

2. Formulation of the problem

The object of the study are low-speed units with the following parameters: geometric - cylinder diameter – 0.05 m; piston stroke - 0.5 m; boundary conditions — the suction gas temperature is 293 K, the suction pressure is 0.1 MPa, and the discharge pressure is 2-10 MPa; the temperature of the cooling environment is 293 K, the heat transfer coefficient on the outer surface of the working chamber is 2000 W / m$^2 \cdot$ K; physical conditions - compressible gas - air; initial conditions — gas state parameters at the lower dead point; cycle time 2 ... 4 s. Thus, as was shown above, the usage of the flywheel in low-speed stage to ensure uniform movement of the mechanism is not applicable. Obviously, the most rational scheme for the best balancing of the entire unit is the opposite scheme with an even number of cylinders. The schemes considered in this paper are presented in figure 1.
3. Theory

The method of calculating the working processes of low-speed long-stroke compressor units is based on a mathematical model with lumped parameters [6, 7]. The design scheme of a piston stage with a linear drive, which is the object of the study, is described in detail in [6].

The input data for the calculation are: suction temperature; discharge pressure and suction pressure; gas constant; gas heat capacity; coefficient of thermal conductivity; cylinder diameter; piston stroke; the value of the dead volume; duty cycle frequency; the characteristics of the construction materials from which the parts that make up the working chamber are made; Wall thickness; valve design parameters; heat transfer coefficients on the outer surfaces of the cylinder stages. The output data of
the calculation results are: the current parameters of the state of the gas; the temperature of the walls of the working chamber; heat and mass flows; integral stage characteristics.

Uniqueness conditions (geometric, physical, initial) and basic simplifying assumptions are presented in detail in [8].

- geometric conditions - cylinder diameter and piston stroke, valve geometry, relative dead volume;
- boundary conditions - suction and discharge pressure, suction temperature; temperature of the cooling medium, heat transfer coefficient on the outer surface of the working chamber. Non-stationary boundary conditions on the inner surfaces of the working chamber walls of the stage were determined by calculating the working cycle of a piston compressor stage using a mathematical model of the working processes of this cycle and calculating the process of heat transfer through the wall to the cooling medium. The boundary conditions on the outer surface of the walls of the working chamber are weakly variable, due to the small amplitude of the temperature change of the walls during the working cycle and almost constant parameters of the cooling medium. From the outer surface of the wall, the heat transfer coefficient \( \alpha_{pr} \) and the temperature of the cooling medium are specified.

- physical conditions - thermodynamic and thermophysical properties of compressible gas;
- initial conditions - gas state parameters in the working chamber in the lower dead point.

The main simplifying assumptions are: the gaseous medium is continuous and homogeneous; simulated processes are reversible, balanced, and quasi-static; the parameters of the state of the working gas change simultaneously over the entire volume of the working chamber; the change in the potential and kinetic energy of the gas is negligible; friction heat of piston seals is not supplied to gas; state parameters in the cavities of suction and discharge are constant; flow of the working gas through the valves and gaps - adiabatic and quasistationary; heat exchange between the gas and the walls of the working cavities is convective and can be described by the Newton ─ Richmann formula; heat transfer coefficient at each time point is the same on all internal surfaces of the working chamber; the heat exchange on the external surfaces of the walls of the working chamber is determined with a constant in time heat transfer coefficient at each considered section of the heat exchange surface; there are no internal heat sources in the walls of the working chamber.

The system of basic calculation equations of the refined method of calculation is similar to the known methods of this type [6, 8] and can be written in the following form:

\[
\frac{dU}{d\tau} = \frac{dL}{d\tau} - \frac{dQ}{d\tau} \pm \frac{dm \times i}{d\tau} \pm \frac{dm_{leak} \times i_{leak}}{d\tau},
\]

where \( dU \) is the change in the internal energy of a gas, \( j \) [6], \( dQ \) is the elementary heat flow removed from the gas or supplied to the gas, \( K \), \( dL \) - work done on gas or by gas itself, \( J \), \( dm \) - change in the mass of gas in the working chamber, \( kg \), \( dm_{leak} \) is the change in the mass of gas in the working chamber through chamber leakages, \( kg \), \( i \) and \( i_{leak} \) are, respectively, the enthalpy of the gas passing through the valves and the enthalpy of the gas passing through the chamber leaks (the change of all the above parameters occurs in a short period of time \( d\tau \)).

Equation of state [6]:

\[
P = \frac{\zeta \times m \times R \times T}{V}
\]

where \( P \) is the current gas pressure, \( Pa \), \( R \) is the gas constant, \( J / K \cdot kg \), \( V \) is the current volume of gas, \( m' \), \( m \) is the current mass of gas, \( kg \), \( \zeta \) is the gas compressibility factor.

For the mass of gas passing through the valves in the open state:

\[
dm = \alpha \times \varepsilon \times f \times (2 \times \rho \times \Delta P)^{1/2} \times d\tau
\]
where $\alpha$ is the coefficient of flow for fungal valves. It was adopted according to the data given in [6]; $\varepsilon$ — gas expansion coefficient [6]; $\rho$ is the gas density, kg / m$^3$; $\Delta P$ is the gas pressure difference before and after the valve on the nth number of the next time step, Pa; $f$ — valve flow area, m$^2$ [6].

To determine the mass of gas flowing into the working chamber or leaving it through leaks in closed valves and gaps in the cylinder-piston seal, the following relationship was used:

$$dm_{\text{rea}} = \Phi_{\text{con}} \times \varepsilon \times \left(2 \times \rho \times \Delta P \right)^{1/2} \times d\tau$$

(6)

where $\Phi_{\text{con}} = \delta_{\text{con}} \times P$ is the equivalent area of the gap, m$^2$, $P$ is perimeter (length) of the considered gap, m; $\delta_{\text{con}}$ is the conditional clearance determined by the results of experimental blow downs, m [4].

The process of unsteady heat transfer through the walls of the working chamber is presented in [9].

Details that form the working chamber are divided into small elements. The resulting mesh allows you to simulate the process of heat exchange of compressible gas with a cooling medium, taking into account the process of heat transfer through the walls of the working chamber. The values of elementary heat fluxes through the faces of neighboring elements are determined from the boundary conditions of the 2nd kind (Fourier law). The heat flux from the compressible gas is determined from the boundary conditions of the 3rd (Newton-Richman equation) specified on the inner surface of the working chamber.

The coefficient of heat transfer averaged over the inner surface of the working chamber is determined by the refined formula from the formula proposed by I.K. Prilutsky [8]. As applied to the stage in this research, this formula was clarified - the empirical coefficient $x$ for air is assumed to be equal to 0.27 [8]:

$$\alpha = \lambda \times \left( \rho / \mu \right)^{0.27} \times W^{0.27} \times D^{0.73}$$

(7)

where $\lambda$, $\mu$, $D_{\text{eqv}}$ and $W$ are the current values, respectively, of the thermal conductivity, dynamic viscosity, equivalent cylinder diameter and conventional gas velocity in the working chamber.

The system of calculated equations was solved numerically by the finite difference method. When developing the algorithm, the Euler method of the 2nd order of accuracy was used. The method is described in more detail in [8].

4. Results

The performed calculated analysis showed that for the adopted scheme there is a decrease in performance while ensuring the same cycle time (see figure 2). This is due to increased leakage through the cylinder-piston seal, because with the same gap the gap length in the two-piston circuit will be twice as long.
Figure 2. The dependence of the feed rate on the cycle time:
1,3 is a usual step with pressure of forcing accordingly 2 MPa; 8MPa
2.4 is a stage with two pistons and a discharge pressure, respectively, 2 MPa; 8MPa

Thus, to obtain the performance of a similar performance of a simple stage, it is necessary to increase the volume of the chamber, that is, with the same diameter, increase the stroke of one piston by the value of $\Delta S$ [10,11]:

$$
\Delta S = \frac{V_e}{F_p} \times \frac{\lambda_1 - \lambda_2}{2}, 
$$

where $V_e$ is the capacity, $m^3/s$, $F_p$ is piston area, $m^2$, $\lambda_1$ is the feed rate of the simple stage, $\lambda_2$ is feed rate of the stage with two pistons.

When obtaining identical performance graphs of pressure and temperature for a simple stage and a stage with two pistons coincide.

Another advantage of the adopted scheme is the increased resource life of the lip seal. It is known that the work resource for a friction pair can be determined by the formula [12]:

$$
I_T = I_{k,n} \times \frac{C_{p1}}{C_{p2}} \times \frac{P_{1}}{P_{2}},
$$

where $I_T$ is the theoretical work resource, $h$, $I_{k,n}$ is known work resource, $h$, $\frac{C_{p1}}{C_{p2}}$ is the ratio of piston speeds, $\frac{P_{1}}{P_{2}}$ is contact pressure ratio.

Taking into account the theoretical values of the temperature in the friction zone, with the design conditions of work - 340K [13,14] and the calculated temperature for a known sample [15], determined by the formula [3]:
\[ t_{med} = 0.2 \times t_{int} + 0.7 \times t_{cool} + 1.35 \times n_o + (\varepsilon - 1)(10.5 + \frac{2.8 \times \Delta S}{S_p} - 0.2 \times n_o) \]  

where \( t_{int} \) is the temperature of the intake gas, \( t_{cool} \) is cooling air temperature, \( n_o \) is crankshaft rotation speed, \( \varepsilon \) is the degree of pressure increase, \( \frac{\Delta S}{S_p} \) is relative piston movement from UDP.

Will find the service life of the cylinder-piston seal low-speed stages. In figure 3 shows the dependence of the work resource on the cycle time.

**Figure 3.** Dependence of the theoretical service life on the cycle time for the considered options for the construction of low-speed stages with regard to ensuring equal performance: 1-1.2-1 low-speed stage with two pistons, respectively, for a discharge pressure of 3 and 5 MPa; 1.2 is a low-speed stage, respectively, for discharge pressure of 3 and 5 MPa

5. **Discussion**

If in a two-piston design the piston speed is left unchanged, then compared to the one-piston circuit, the resource remains almost unchanged, and the productivity will increase twice due to the reduction of the cycle time. This will naturally affect the stage workflow. The graph of pressure and temperature changes are presented respectively in figure 4 and 5.
Figure 4. The graph of pressure change in the working chamber during the cycle: 1 for a scheme with two pistons; 2 for a single piston circuit

Figure 5. Graph of temperature change in the working chamber during the cycle: 1 for a scheme with two pistons; 2 for a single piston circuit

6. Conclusion
Conducted in this paper, the analysis allowed to determine the most advantageous scheme in terms of balancing. Such an arrangement with two oppositely moving pistons, as the analysis of the working process has shown, does not impair its efficiency and allows increasing the service life of the piston-piston seal.

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