Justification of parameters of the process of injecting compressed air into the piston air spring in the boom machine working equipment

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Abstract. While lifting a payload, boom machines perform a useless job to overcome gravitational forces of the working equipment gravity force. The piston spring, injected with compressed air to a pressure of $p_w = 10–15$ MPa, ensures balancing gravitational forces of gravity during vertical movements and eliminates the indicated disadvantage. The piston air spring, independently mounted in the boom machine working equipment, represents an energy saving unit. A detailed analysis of physical processes of obtaining high-pressure compressed air for the injection into the air spring has been conducted, new solutions have been obtained, connected with the technology of air compression to high pressure, satisfying definite design requirements.

Key-words: piston air spring, energy saving, balancing, compressed air, compressor, receiver.

1. Introduction

In construction, mining industry, agriculture, a huge number of boom machines are in operation: excavators, loaders, self-propelled boom cranes, which possess a low overall efficiency and consume a huge amount of expensive fuel produced from the extracted oil products. While lifting working equipment, half of the engine’s energy is uselessly wasted for overcoming gravitational forces of gravity of buckets, booms and other elements of the working equipment.

In paper [1] the authors Yogeshkumar R. Chhetra, Rajesh M. Joshi, Krishan Gotewal, M. Manoah Stephen study a negative influence of gravitational forces of gravity of manipulator elements on power characteristics. Passive methods of the gravity force compensation for raising the manipulator efficiency are considered.

In paper [2] the authors Z. Zhihong, W. Yunxin, M. Changxun have performed the dynamic modeling of different positions of a boom. Mechanical parameters have been established allowing to reduce dynamic loads.

In paper [3] the authors Z. Miaofen, S. Shaohui, G. Youping, Z. Dada establish a connection between mechanical and hydraulic control models of the loader operating process. The multiparameter optimization of the engine operation mode by numerical integration methods has been conducted.

In paper [4] the authors H. Xie, G. Zhang have found that in the upper position of the automobile crane boom, the hydraulic system pressure is raised, and vibration is increased at the expense of increasing reduced mass and decreasing a power arm.

In papers [5-8] the authors V.N. Tarasov, I.V. Boyarkina have proposed a new concept in designing energy-efficient working equipment. It is proposed to consider the working equipment energy while lifting the boom, consisting of two components: useful energy of the load lifting and useless lifting energy of gravity forces of working equipment elements. The air spring is mounted between power
hydraulic cylinders for the boom lifting and it works independently, i.e. is not connected with other systems and ensures the gravity force balance of the bucket, boom, arm, and other elements of the working equipment.

The energy saving problem of working processes of this group of machines allows to get a huge economic effect. In a piston air spring compressed air is used as a working medium instead of the hydraulic system working fluid. This circumstance is related to the ecological and economic quality categories of boom machines.

The use of the air spring in the working equipment of boom machines [5-9] requires the substantiation of parameters of physical processes and design solutions connected with air compression to a high pressure.

2. Problem statement

The independent air spring located between the hydraulic power cylinders [5-9] represents the main element of the energy-saving system. The air spring design is similar to the hydraulic power cylinder of the boom of these machines. An advantage of the air spring is represented by the usage of compressed air as a working medium instead of expensive oils or hydraulic fluid.

The problem of filling air springs of boom machines with compressed air at a pressure of $p_w = 10-15$ MPa is a new one, and it requires certain expenses and development of new technologies.

In this paper the problem of creating the technology of compressed air injection into the air spring of boom machines has been solved, and engineering solution of this problem has been provided.

3. Theory

Modern portable and mobile compressor units are produced as multiple piston ones with the high-frequency drive of operating strokes. Such compressors are characterized by high complexity and low efficiency in connection with the deterioration of heat transfer processes during high-frequency processes of air compression [10-11].

For the process of injecting compressed air into the piston air spring of a loader, excavator with a pressure of $p_w = 10-15$ MPa, which volume is about $V_w = 0.06$ m$^3$, we’ll use a process flow diagram with a single-piston compressor.

Figure 1 shows the process flow diagram of injecting compressed air, which in the first mode ensures a preliminary injection of compressed air from the atmosphere into the receiver under a pressure of $p_1 = 3-6$ MPa, and in the second operation mode it ensures the compressed air injection up to a pressure of $p_2 = 10-15$ MPa.

To ensure the possibility of considering two operation modes of the process flow diagram the valves $B_1, B_2, B_3$ are shown in two conditions: additional (near) – the first mode; basic – the second operation mode.

In Figure 1 the open valve is a light colored without background shading, closed valve is with background shading. The process flow diagram for the first mode ensures the air intake from the atmosphere through the open valve $B_1$, and inlet valve $K_1$ performs the air compression in the compressor’s working chamber and supplies it through the discharge valve $K_2$ and cooler to receiver and air spring, which in this case is a cylinder’s analogue. On three sides the working cavity of compressor is closed by the water cooling jacket, at the same time the compressed air is additionally cooled in water cooler.
Figure 1. Process flow diagram of the compressed air injection by a compressor to an air spring:

$B_1, B_2, B_3, B_4$ – valves; $K_1$ – inlet valve; $K_2$ – discharge valve; $M_1, M_2$ – pressure gauges;

$T_1$ – thermometer.

In order to obtain a high pressure in the process flow diagram in Figure 1 the valves $B_1, B_3$ are closed that allows to connect receiver, having been preliminarily charged with compressed air, with the working chamber of compressor through the open valve $B_2$ and inlet valve $K_1$.

During the piston operating stroke of compressor to the right, there occurs additional compression of the air, which is supplied through the $K_2$ discharge valve to cooler and through the $B_4$ valve to air spring.

The submitted technology peculiarity consists in the fact that in this case the compressor represents a device operating at a low stroke frequency. For charging the air spring as per this diagram a restricted number of the piston movement working cycles are required.

We’ll perform a detailed analysis of the physical processes of obtaining high-pressure compressed air.

The initial condition of the process flow diagram components shown in Figure 1 is as follows: receiver is charged with compressed air up to a pressure of $p_r = 6$ MPa; receiver’s volume $V_r = 0.12$ m$^3$; volume of the compressor’s working cavity $V_k = 0.02$ m$^3$; volume of the air spring $V_n = 0.06$ m$^3$; ambient air temperature $T = 300 \; ^\circ K$, i.e. $27 ^\circ C$.

From receiver the compressed air is supplied through the open valve $B_2$ to the inlet valve $K_1$, which opens and fills the working chamber, in this case the compressor’s piston under pressure takes the left limit position. At the same time, when the compressor is not in operation, air from the working chamber through the $K_2$ valve enters the empty cavity of air spring.

As a result of performed operations, the receiver’s pressure is decreased and becomes similar in the receiver, compressor’s working chamber, and air spring: $p_r = p_k = p_n = 3.6$ MPa. The indicated condition is the initial one for the completion of the first working cycle.

When putting the compressor into operation the piston starts to move to the right, in this case the discharge valve $K_2$ is opened immediately, as the pressure in the cavities prior to the $K_2$ valve and behind the valve is the same $p = 3.6$ MPa.

Thus, while moving the piston to the right as a result of the connection of two cavities $V_k$ and $V_n$, a polytropic process of air compression occurs in them.

The pressure in the air spring is changed according to the equation
\[ p_n = p_{k,o} \left( \frac{V_n + V_k}{V_n} \right)^{n_1} , \]  

where \( p_n \) is the pressure in the compressor and air spring, \( p_{k,o} \) is the initial pressure in the compressor; \( V_k \) is the compressor’s working chamber volume; \( V_n \) is the air spring volume; \( n_1 \) is the polytropic index for air, \( n_1 = 1.25-1.3 \).

As a result of the compressor’s first working cycle the pressure as per formula (1) in the air spring has increased to the value of \( p_n = 5.233 \) MPa.

The obtained results allow to determine the air mass supplied to the air spring from the receiver during one cycle of the compressor supply as per the Mendeleev-Clapeyron equation

\[ m_{f,c} = \frac{M p V}{RT} , \]  

where \( m_{f,c} \) is the air mass in the first cycle; \( M \) is the air molar mass, \( M = 0.029 \) kg/mol; \( R \) is the gas constant, \( R = 8.31 \) J/(mol·Kelvin); \( p \) is the receiver pressure in the first cycle, \( p = 3.6 \) MPa; \( V \) is the compressor’s working chamber volume, \( V = V_k \); \( T \) is the receiver temperature, \( T = 300 \) °K, i.e. \( t = 27 \) °C.

As per formula (2) the air mass supplied to the air spring in the first cycle is equal to \( m_{f,c} = 0.838 \) kg. Formula (2) also allows to determine the \( m_n \) air mass in the charged air spring at a pressure of \( p = 10 \) MPa and \( V = V_p = 0.06 \) m³, which is equal to \( m_n = 6.98 \) kg. The \( m_n \) and \( m_{f,c} \) mass ratio allows to judge about the number of cycles \( n_c \) required for injecting compressed air into the air spring.

Consequently, for increasing the gas spring pressure from 3.6 MPa to 10 MPa, a limited number of cycles of the compressed air supply by the compressor are required.

For further consideration of the compressor’s operating process cycles, we use the \( pV \) diagram [10-11].

Figure 2 shows the \( pV \) diagram of the compressor’s operating process for the second cycle. The scales of the variables \( p, V \) and pressure points on the diagram, for specifying the processes, are shown for the second cycle of the compressed air supply.

The compressor’s working chamber volume is determined by the formula

\[ V_k = \frac{\pi D^2}{4} S , \]  

\( D \) is the diameter of the compressor’s working chamber.
where \( D \) is the piston diameter, \( D = 0.2 \text{ m} \); \( S \) is the compressor’s piston operating stroke, \( S = 0.637 \text{ m} \). The compressor’s working chamber volume is \( V_k = 0.02 \text{ m}^3 \). The working chamber volume, as can be seen from formula (3), is proportional to the compressor’s piston movement value \( S \).

At point 4 of the diagram the second cycle is started of the compressor’s piston movement to the left for filling the working chamber with the next air portion. At the same time the \( K_1 \) inlet valve remains closed for some time, as in the previous cycle a small dead volume \( V_m \) remained in the chamber, and compression end pressure is maintained in this volume. That’s why when the piston moves to the left with the \( K_1 \) valve closed, the \( V_m \) residual volume expansion occurs, the working chamber pressure is decreased along line 3-5 of the diagram. At point 5, the inlet valve \( K_1 \) is opened due to balancing the indicated pressures.

The residual volume expansion process is described by the polytropic air expansion equation. The expansion process of the residual dead volume is determined by the formula

\[
V_{m,p} = V_m \left( \frac{p_n}{p_p} \right)^{\frac{1}{n}} ,
\]

where \( p_m, p_n \) are the pressures in the receiver and air spring respectively; \( V_m \) is the dead volume.

For the second supply cycle the expansion volume is equal to \( V_p = 0.001346 \text{ m}^3 \) (see Figure 2).

The obtained result shows that point 4 of the start of piston movement and point 5 of the \( K_1 \) valve opening practically coincide. That’s why in this research the dead volume effect as a result of the adopted design improvements can be disregarded.

For the observability preservation of the ongoing processes’ parameter variation during the analysis we also neglect transient processes of opening the inlet and discharge valves, i.e. the overshooting of the \( K_1 \) and \( K_2 \) valve opening processes can be disregarded, \( \Delta p_{sl} = 0 \) and \( \Delta p_{sg} = 0 \) (see Figure 2).

At point \( I \) of section 4-1 of the diagram the compressor’s working volume was filled with air from the receiver. The piston stops at the left dead center and valve \( K_1 \) closes.

The compressor’s working chamber air filling process is accompanied by a pressure decrease in the receiver according to the equation

\[
p_p = p_o \left( \frac{V_p}{V_p + V_k} \right)^{\frac{1}{n}} ,
\]

where \( p_o \) is the receiver pressure at the beginning of the working chamber volume filling process; \( p_p \) – receiver pressure at the end of the compressor’s working chamber charging process; \( V_p, (V_p+V_k) \) – charged volumes at the beginning and end respectively of the charging process.

At the end of the second supply cycle the receiver pressure decreased to a value of \( p_p = 2.946 \text{ MPa} \) (point 1, Figure 2). Further, according to the \( pV \) diagram the piston starts to move to the right with the \( K_2 \) valve closed, as the compressor’s working chamber pressure \( p_k \) is less than the air spring pressure \( p_n \). There is a polytropic air compression in the compressor’s working chamber. The process occurs along line 1–2 of the diagram. At point 2 the compressor and air spring pressure is balanced, and the \( K_2 \) valve opens.

The \( K_2 \) valve opening torque is determined by the \( V_2 \) volume, which is calculated by the formula

\[
V_2 = V_k \left( \frac{p_n}{p_k} \right)^{\frac{1}{n}} ,
\]

where \( V_k \) is the initial volume of the compressor’s working; \( p_k, p_n \) are the compressors’ working chamber and air spring pressures for the second cycle of the compressed air supply.

As per formula (6), the volume is equal to \( V_2 = 0.0129 \text{ m}^3 \) (see Figure 2). At point 2 of the diagram the compressor and air spring cavities intercommunicate with each other. In connection with the air spring limited volume a further compressed air pressure increase occurs in section 2–3 of the \( pV \) diagram.
while its ejecting into the air spring. Within this period in two cavities $V_k$ and $V_n$, the polytropic air compression process is performed as per the formula

$$p_n = p_o \left( \frac{V_n + V_k}{V_n} \right)^{\frac{n}{n-1}},$$

(7)

where $p_n$ is the air spring and compressor’s working cavity pressure with the $K_2$ valve open; $p_o$ is the initial pressure at the moment of the $K_2$ valve opening; $V_k = V_2$ is the working chamber volume at the moment of the $K_2$ discharge valve opening.

The pressure at point 3 of the diagram in the air spring at the end of the second cycle is $p_n = 6.736$ MPa (see Figure 2).

Thus, all the elements of the second cycle of the compressed air injection from the receiver into the air spring have been considered and analytically presented, regularities of the processes of decreasing the receiver pressure and increasing the air spring pressure with the cyclic compressed air supply from the receiver to the air spring have been established.

Subsequent cycles of injecting compressed air from the receiver into the air spring $n_c = 3, ..., 6$ are performed similarly in accordance with the considered laws and equations.

4. Results discussion

Figure 3 shows the generalized $pV$ diagram of all compressor’s working cycles while injecting compressed air into the air spring.

![Figure 3. Generalized $pV$ diagram of the compressed air injection processes into the air spring: 1-6 – process of filling the compressor’s working chamber with air from the receiver; 7-12 – process of the compressed air supply by the compressor into the air spring.](image)

The generalized $pV$ diagram characterizes the compressor’s technological capabilities. Point 2 characterizes the beginning of the air compression in the compressor’s working chamber.

Digits 1, 2, ..., 6 show the processes of filling the compressor’s working chamber with air from the receiver. Line 1 corresponds to the beginning of the compressed air injection process at a pressure of $p_r = 3.6$ MPa; line 2 characterizes the receiver pressure reduction process in the second injection cycle $p_r = 3.6, ..., 2.946$ MPa. Line 6 corresponds to the receiver pressure reduction process in the last cycle within the injection range $p_r = 1.615, ..., 1.322$ MPa.
Digits 7, 8, ..., 12 indicate the air spring pressure increase processes when injecting compressed air portions into it. It can be seen from the diagram that line 7, corresponding to the first cycle of injecting compressed air into the air spring, possesses a maximum value, i.e. injection occurs during the entire piston movement, and line 12 for the last injection cycle possesses a small value. The length change of lines 7–12 in the cyclic injection process characterizes a decrease in the efficiency of the compressed air injection process from the receiver into the air spring due to the pressure increase in the air spring.

5. Consideration of the results
Figure 4 shows the dependence of the compressed air mass $m_n$ entering the air spring on the number of injection cycles $n_c$.

![Figure 4. Dependence of the air mass $m_n$, supplied to the air spring, on the number of cycles $n_c$ of the compressor’s injection.](image)

The considered process flow diagram can be successfully used for filling any cylinders possessing similar parameters with compressed air.

6. Conclusion
For the device of the low-cycle compressed air injection into the air spring the technological parameters of the compressed air injection cycle from the receiver into the air spring have been analytically calculated and substantiated, regularities of the processes of decreasing the receiver pressure and increasing the air spring pressure have been established.
As per the Mendeleev-Clapeyron equation the air mass in each cycle and accordingly the total number of cycles required for injecting compressed air into the air spring have been determined.
The technology ensured the air spring injection just in six cycles has been theoretically substantiated. The proposed technology is a universal one and can be used for the injection of cylinders of specified capacity with compressed air. In this technological process the compressor is used not only as a device for the air compression, but also as a metering device, ensuring the movement of definite volumes and masses of compressed air from the receiver to the air spring.

7. References
[1] Chheta Yogeshkumar R, Joshi Rajesh M, Gotewal Krishan and Manoah Stephen M 2017 A review on passive gravity compensation International conference of Electronics Communication and Aerospace Technology ICECA Volume 1 pp 184 – 189
[2] Zhihong Z, Yunxin W and Changxun M 2011 Simulation and Optimization of the Driving Forces of Hydraulic Cylinders for Boom of Truck Mounted Concrete Pump Intelligent Computation Technology and Automation ICICTA International Conference Vol 1 pp 915 – 919
[3] Miaofen Z, Shaohui S, Youping G and Dada Z 2010 Research on the Multi-domain Modeling and Optimizing Method for Loader Executing System Intelligent System Design and Engineering Application ISDEA International Conference Vol 2 pp 854 – 857
[4] Xie H and Zhang G 2013 Research on Characteristics of the Piloted Follow-up Load Control Valve in Automobile Crane Luffing System Fifth International Conference on Measuring Technology and Mechatronics Automation pp 887 – 892
[5] Tarasov V N, Boyarkina I V and Boyarkin G N 2018 Theory of the piston air spring in the boom machine working equipment II International scientific conference Mechanical Science and Technology Update MSTU 2018 IOP Conf Series Journal of Physics Volume 1050 012085

[6] Tarasov V N, Boyarkina I V and Kovalenko M V 2017 Theory of balancing gravity forces of bodies moved by the boom of pneumatic machines by means of an air spring Omsk Scientific Reporter (Omsk State Technical University) Series instruments machines and technologies No (155) pp 28–32

[7] Boyarkina I V and V N Tarasov 2017 Method of obtaining an additional power in boom working equipment by means of an air spring Omsk Scientific Reporter (Omsk State Technical University) Series instruments machines and technologies No 5 (155) pp 5–9

[8] Tarasov V N, Boyarkina I V, Boyarkin G N and Tarasov S Ye 2015 Energy saving device patent 2558545 of the Russian Federation Published BI No 22 p 9

[9] Tarasov V N, Boyarkina I V, Schlegel I F and Schlegel Ya I 2020 Device for the low-cycle compressed air injection into cylinders patent 2729316 of the Russian Federation Published BI No 22 pp 4

[10] Gabitov F R and Gumerov F M 2016 High pressure Equipment for modern technologies (Moscow KDU Publishing House) (Universal Book) pp 340

[11] Bystritsky G F 2008 Power plant equipment of industrial enterprises (Moscow Academy Publishing Center) pp 304