Creation of an suspended compressor for oil production from wells

Sh G Mingulov, I Sh Mingulov

Ufa State Petroleum Technological University, Branch of the University in the City of Oktyabrsky, 54a, Devonskaya St., Oktyabrsky, Republic of Bashkortostan, 452607, Russian Federation

E-mail: vsh@of.ugntu.ru

Abstract. The performed analysis of the design of suspended compressors used in the Republic of Tatarstan and the Orenburg region showed an insufficient degree of compressor pressure increase due to the presence of “dead” space in the compressor cylinder. The study developed a new design of suspended piston compressor for gas extraction from the annular space of the well and its injection into the discharge line, which allows increasing the degree of gas pressure increase by oil filling the gap between the piston and cylinder. The authors elaborated theoretical bases for calculation of hydraulic resistance to piston movement in the oil-filled cylinder of a piston compressor, which showed significant resistance due to small gaps between piston and cylinder.

Keywords: suspended compressor, oil well, annular space, flow rate, leakage

1. Introduction

The efficiency of oil field development depends to a large extent on the efficiency of the technical means and technologies used for oil production [1-8]. Mechanized operation of oil wells involves the partial separation of associated gas from oil at receiving pumps and its accumulation in annular space.

Increase of pressure in annular space results in an increase of back pressure on the reservoir, therefore, in a decrease of fluid inflow to the bottomhole and squeezing of the dynamic fluid level before the pump intake, as well as in gas penetration into the pump, decrease and failure of its supply. This problem is especially acute at high pressures in the discharge lines of wells.

Reducing gas pressure in the annular space is a significant reserve for increasing oil production. Thus, it is very urgent to solve the problem of forced gas pumping from the annular space of production wells. Different geological and field conditions of fields operation require different methods of solving this technical problem to increase well flow rate [9, 10].

2. Materials and methods

Today, a significant share of mechanized wells, 36.6% or about 51 thousand, are operated from SPR [11]. If the reduction of gas pressure in the annular space results in 1 additional t/day of oil production from one well equipped with an SPR, then on a national scale the additional oil production will be equivalent to commissioning of the largest oil field.

One of the effective engineering solutions for increasing productivity of wells equipped with SPR
is the use of suspended compressors for forced gas pumping out of annular space, installed on pumping machines.

Suspended compressors driven by the pumping machine balancer have been used for quite a long time [12]. The principle of their work is to transfer the harmonic movement of the balancer to the compressor piston. In this case, the cylinder or piston swivels to any point of the balancer arm, and accordingly, the piston or cylinder linked to any point of the movable balancer. As a result, the piston makes a reciprocating motion in the compressor cylinder and pumps gas from the annular space to the discharge manifold through suction and injection valves.

Well flow rate increase $\Delta Q$ due to pressure decrease in annular space depends on the degree of bottomhole pressure decrease by the formula:

$$
\Delta Q = (P_w' - P_w'')C,
$$

where $P_w'$ and $P_w''$ are bottomhole pressure before and after compressor start-up respectively; $C$ is the well productivity coefficient.

The gas is pumped with flexible hoses and the flow is controlled with gate valves and check valves. In the case of cluster placement of wells with small distances between the wellheads, it is possible to connect wells operated by ESPs to the compressor.

Theoretically, there are two reasons for the increase in fluid withdrawal when the pressure in the annular space decreases: a decrease in bottomhole pressure increasing the depression on the reservoir, and, the rise of the dynamic level and the improvement of pumping unit operating conditions.

OAO Orenburgneft and PJSC TATNEFT implemented the technology of applying suspended compressors of domestic and imported production (USA). Additional oil production was 0.4-2.7 t/day per well, and it was revealed that the amount of additional oil produced largely depends on the pressure in the discharge manifold. Increasing pressure in the discharge manifold reduces the amount of additional oil produced due to increasing gas leaks through the gap between the piston and cylinder, i.e., the efficiency of the suspended compressor decreases with increasing pressure in the discharge manifold.

Thus, the analysis of existing structures of suspended compressors for forced gas pumping from the annular space of wells equipped with SPR shows that the efficiency of suspended compressor application decreases as the pressure in the disposed reservoir increases. The use of suspended compressors becomes inefficient at the pressure of 1.5 MPa or more in discharge manifolds.

3. Results and Discussion

This paper presents an improved piston compressor design with increased pressure rise.

The essence of the new piston compressor development is to create a high degree of pressure increase by filling the gap between the piston and compressor cylinder with fluid (oil), eliminating the appearance of leaks.

Figure 1 shows a schematic diagram of the compressor installation to the pumping machine (see Figure 1, a) and a diagram of the installation of shut-off valves (see Figure 1, b).

For the calculation and design of the compressor, it is necessary to have data on the resistance provided by the compressor piston to the movement of the pumping machine balancer. Reaching significant resistance values will make the work of the pumping machine impossible. In particular, when pumping high-viscosity oil, these resistances can cause the column of rods to hang as the balancer head moves down. In this regard, it is necessary to calculate the friction resistance of the piston in the cylinder. Since there is fluid in the gap between the piston and the cylinder, the friction resistance in the compressor is determined by the hydrodynamic friction of the piston in a viscous medium.
The theoretical basis for calculating hydraulic resistance in an oil-filled compressor. If viscous friction linearly depends on the velocity of movement (layered flow), it is possible to obtain a differential equation that associates the pressure gradient with the acceleration of the mass of the fluid and the friction resulting from the movement. The solution of this equation allows us to obtain mathematical expressions for the profile of fluid flow in the channel and, consequently, for its derivatives: the friction on the piston surface and pressure gradient in the gap.

Later on, when outputting functional dependencies, we took the viscous fluid model and the harmonic laws of fluid flow rate change in the channel and piston movement.

We should note that the maximum friction values take place in the middle of the piston stroke.

When assessing friction forces, we selected the concentric position of the piston in the cylinder. The concentric arrangement of the axes will result in a small increase in resistance compared to the eccentric one (Figure 2).

All these assumptions contribute to a certain convenience in the construction of computational schemes of hydrodynamic resistances.

We should note that the calculated and experimental relations obtained in the future can determine the hydrodynamic friction forces in the compressor only if it is possible to estimate oil viscosity in the cylinder with a certain measure of accuracy.

Figure 1. Schematic diagram of compressor installation: a - to the pumping machine: 1 - balancer; 2 - connection rod; 3 - counterbalance; 4 - balancer head; 5 - suspension; 6 - suspension; 7 - piston rod; 8 - compressor cylinder; 9 - flow line; 10 - compressor suction line; 11 - compressor discharge line; b - to the oil gathering system: 1, 2 - ball valve; 3, 4 - built-in check valve; 5 - check valve; 6 - built-in safety valve.

Figure 2. The scheme of movement of the piston and compressor fluid: 1 - compressor cylinder; 2 - stock; 3, 4 - suction and discharge valves; 5 - seal; 6 - oil.
The initial equation of motion assuming a laminar structure of unsteady fluid flow in the cylinder has the following form:

\[
\frac{du}{dt} = -\frac{1}{\rho} \frac{dP}{dz} + \Omega \left( \frac{d^2u}{dr^2} + \frac{1}{r} \frac{du}{dr} \right),
\]

(2)

where \( du/dt \) is the time derivative of the elementary layer velocity; \( dP/dz \) is pressure gradient along the cylinder axis; \( du/dr \) is the derivative of the velocity along the section radius.

The boundary conditions of integration are determined from the conditions of fluid adhesion to the channel walls. Since the flow in the channel is simultaneous, the speed in each fluid layer will be a function of radius and time, and the pressure gradient will be a function of time only since the flow at each moment is constant along the length of the cylinder. The work [13] obtained the solution of equation (2) for the calculation of hydrodynamic pressure drop in the tubing of rod installations.

The solution of this equation requires an additional condition for the compressor conditions characterizing the fluid flow through the annular gap. When the cylinder moves downwards, the volume of the piston displaced from the cylinder determines the liquid consumption per unit of time. Thus, for the cylinder down stroke we have the following:

\[
a^2 v_{pcs}(t) = \int_a^b u(r, t) 2\pi r dr,
\]

(3)

where \( v_{pcs} \) is piston speed; \( a, b \) are piston and cylinder radii, respectively.

The boundary conditions of the integration for the downstream input will the following:

\[
\begin{align*}
u(a, t) &= -v_{pcs}(t); \\
u(b, t) &= 0.
\end{align*}
\]

(4)

We will use an approximate way of integrating the equation. The essence of the method consists in averaging of fluid acceleration (regardless of the sign) along the channel cross-section.

Thus, the left part of equation (2) is a function of time:

\[
\frac{du}{dt} = \frac{\phi(t)}{\int_a^b \frac{du}{dr} dr}.
\]

(5)

We will enter the designation

\[
\phi(t) + \frac{1}{\rho} \frac{dP}{dz} = \psi(t)
\]

(6)

and we show the solution for the cylinder down stroke. The integral of equation (2) considering (6) has the following form:

\[
u(r, t) = \frac{1}{4} (r^2 + c_1 ln r + c_2) \Psi(t) + (c_3 ln r - c_4) v_{pcs}(t),
\]

(8)

where \( c \) is geometrical constant.

The partial derivative (8) in time will be as follows:

\[
du/dt = \frac{1}{4} (r_2 + c_1 ln r + c_2) \Psi(t) + (c_3 ln r - c_4) v_{pcs}(t).
\]

(9)

By substituting (8) in expression (9) and integrating, we get:

\[
\varphi(t) c_5 \Psi(t) + c_6 v_{pcs}(t).
\]

(10)
\[ \Psi (t) = c7\nu_{pcs} (t) \]

and

\[ \Psi (t) = c7\nu'_{pcs} (t), \] (11)

Then

\[ \varphi(t) = c5\nu_{pcs} (t) + c6\nu'_{pcs} (t) = c8\nu'_{pcs} (t). \] (12)

The final dependence for speed is as follows:

\[ u (r, t) = (Ar^2 + B\ln r + C)\nu_{pcs} (t). \] (13)

Given that the functions \( \Psi (t) \) и \( \varphi (t) \) are now known, then, according to (7) we have:

\[ \frac{dP}{dz} = N_1 \nu_{pcs} (t) - N_2 \nu'_{pcs} (t). \] (14)

The multipliers included in equations (13) and (14) have the following form:

\[ A = \frac{1}{E}; \]

\[ E = b^2 - a^2 - (a^2 + b^2)\ln \frac{b}{a}; \]

\[ B = \frac{G}{E} + \frac{1}{\ln \frac{b}{a}}; \]

\[ C = \frac{b}{E} - \frac{\ln b}{\ln \frac{b}{a}}; \]

\[ N_1 = \frac{4\mu}{E}; \]

\[ N_2 = \rho \cdot \left( \frac{a}{b - a} - \frac{1}{\ln \frac{b}{a}} - H \right); \]

\[ H = \frac{G}{E} + \frac{2(a^2 + ab + b^2)}{3E}; \]

\[ G = \frac{a^2 - b^2}{\ln \frac{b}{a}}. \]

Functional dependence (13) is represented graphically as a profile of fluid flow velocity in the gap between the piston and cylinder. The profile shows a high current velocity gradient on the piston surface and therefore the viscous friction of the fluid (Figure 3).
We determine the speed and acceleration of the rod column by the formulas:

\[ v_{pcs.} = \frac{\omega S}{2} \sin \omega t; \]
\[ v'_{pcs.} = \frac{\omega S}{2} \cos \omega t, \]

where \( S \) is the length of the piston stroke; \( \omega = \pi n / 30; n \) is the number of swings.

In equation (14), the right parts consist of two summations, one of which includes the viscosity of the fluid and the other includes its density. The first summand, a viscous component, is that part of the gradient spent to overcome hydraulic resistance. The second summand, the inertial component, includes the density of the liquid and thus characterizes the part of the gradient that compensates the inertia of the liquid column.

Figure 4 shows the curves of changes in the dimensionless pressure gradient depending on the cylinder stroke.

\[ \frac{dP}{dz} = \frac{IT\%S}{2} \beta \left[ N_1^2 + (N_2 IT\%)^2 \beta \sin (IT\%t + \Pi \frac{\pi}{2}) \right], \]
where $\chi = \arctg \left( \frac{N_2 \omega}{N_1} \right)$, shows sinusoidal phase shift,

$$\frac{\omega S}{2} \sqrt{N_1^2 + (N_2 \omega)^2}$$

is the amplitude of the harmonic curve.

Knowing the total pressure amplitude, it is easy to calculate the maximum and minimum pressure in any piston section.

To calculate the pressure gradient, the obtained expression (14) requires the interpretation of the physical meaning of the values included in them. For this purpose, we present the viscous component of the pressure gradient in its unfolded form.

$$N_1 \nu_{pcs}(t) = \frac{4\mu \nu_{pcs}(t)}{b^2 - a^2 - (a^2 + b^2) \ln \frac{b}{a}}$$ (16)

If we compare this dependence with the expression for the pressure gradient obtained in the assumption of constancy of piston speed, we can establish their complete analogy. In other words, calculation of viscous component of energy loss in a compressor with variable speed in time is possible according to the laws of steady flow.

To analyze the inertial components of the pressure, we use the following statement.

According to the law on preservation of quantity of movement of a liquid element, total resistance in a cylinder is defined as a superposition of viscous and inertial forces.

The inertial pressure has the following form:

$$\frac{\Delta P}{L} = \rho \frac{dU_{cp}}{dt}$$ (17)

where $U_{av}$ is average flow rate.

Given the basic assumption of averaging the accelerations in solving equation (2), the second term of the right side of equation (14) can be approximated by formula (17). The numerical analysis of these dependencies showed a small discrepancy in the inertial pressure calculations.

Therefore, we can argue that the solution of the equation by the Slezkin-Targ method results in preserving the independence of action of energy loss components in the gap. Thus we can calculate viscous components according to stationary laws of hydraulics, and inertial components - according to formula (17). In both cases it is necessary to have curves of average flow velocity changes over time.

From the reliability aspect of compressor operation, it is possible to have viscous friction forces acting on the balancer when the cylinder moves downwards.

The viscous friction force is determined by the gradient of the flow velocity on the contact surface with the piston body:

$$F_{mp} = 2\pi a l \mu \frac{du}{dr} |r = a.$$ (18)

Derived speed by the radius of the section, according to (18), has the following form:

$$\frac{du}{dr} |r = a = \frac{a^2 - b^2}{a[b^2 - a^2 - (a^2 + b^2) \ln b/a]} OS_{CEC}.$$ (19)

On the other hand, according to (14), we have:

$$\frac{dP}{dz} = \frac{4\mu O_J OS_{CEC}}{[b^2 - a^2 - (a^2 + b^2) \ln b/a]}$$ (20)

We can substitute the speed found in equation (20) $\nu_{pcs}$ to equation (19).
\[ F_{tr} = \frac{\pi(a^2 - b^2)\Delta P_r}{2}. \]  

(21)

Thus, the friction on the piston surface is numerically equal to half the product pressure on the cylinder gap area.

For piston friction resistance and pressure acting on the piston end, we will finally obtain the following formula:

\[ F_{tr} = 0.5\mu nS \cdot \frac{1 + m^2}{\ln m(1 - m)^2}. \]  

(22)

The greatest practical interest is the value of friction arising in the middle part of the piston stroke, i.e. when there is no inertial pressure in the fluid.

Therefore, in the future the \( F_{tr} \) value will mean the maximum hydrodynamic load, and we will not take liquid inertia into account.

4. Conclusion

1. The performed analysis of suspended compressor design used in the Republic of Tatarstan and Orenburg region showed an insufficient degree of compressor pressure increase due to the presence of "dead" space in the compressor cylinder.

2. The authors developed a new design of suspended piston compressor for gas extraction from the annular space of the well and its injection into the discharge line, which allows increasing the degree of gas pressure increase by oil filling the gap between the piston and cylinder.

3. They elaborated theoretical bases for calculation of hydraulic resistance to piston movement in the oil-filled cylinder of a piston compressor, which showed significant resistance due to small gaps between piston and cylinder.

References

[1] Solar Thermal Electricity Global Outlook 2016 URL: http://www.solarpaces.org/new-web-nasertic/images/pdfs/GP-ESTELA-SolarPACES_Solar-Thermal-Electricity-Global-Outlook-2016_Executive-Summary.pdf

[2] Yakupov R F, Mukhametshin V Sh, and Tyncherov K T 2018 Filtration model of oil coning in a bottom water-drive reservoir Periodico Tche Quimica 15 (30) 725-733

[3] Mukhametshin V V, and Kuleshova L S 2020 On uncertainty level reduction in managing waterflooding of the deposits with hard to extract reserves Bulletin of the Tomsk Polytechnic University. Geo Assets Engineering 331(5) 140–146 DOI 10.18799/24131830/2020/5/2644

[4] Levitt D, Jackson A, Heinson C, Britton L N, Malik T, Dwarkanath V, Pope G A 2009 Identification and Evaluation of High-Performance EOR Surfactants SPE Reservoir Evaluation & Engineering 12(2) 243-253 DOI: 10.2118/100089-PA

[5] Tyncherov K T, Mukhametshin V Sh, Paderin M G, Selivanova M V, Shokurov I V, Almukhametova E M 2018 Thermoacoustic inductor for heavy oil extraction IOP Conference Series: Materials Science and Engineering (MEACS 2017 – International Conference on Mechanical Engineering, Automation and Control Systems) 327 (4) (042111) 1–8 DOI:10.1088/1757-899X/327/4/042111

[6] Mukhametshin V V, and Andreev V E 2018 Increasing the efficiency of assessing the performance of techniques aimed at expanding the use of resource potential of oilfields with hard-to-recover reserves Bulletin of the Tomsk Polytechnic University. Geo Assets Engineering 329 (8) 30–36

[7] Zeigman Yu V, Mukhametshin V Sh, Khafizov A R, and Kharina S B 2016 Prospects of Application of Multi-Functional Well Killing Fluids in Carbonate Reservoirs SOCAR Proceedings 3 33–39 DOI: 10.5510/OGP20160300286

[8] Mukhametshin V V 2018 Rationale for trends in increasing oil reserves depletion in Western Siberia cretaceous deposits based on targets identification Bulletin of the Tomsk Polytechnic
University. Geo Assets Engineering 329(5) 117–124
[9] Valeev M D, Sevastyanov A V, Nigay Yu V, Tret'yakov R S 2014 Technology for increasing the productivity of oil wells Exposition Oil & Gas 6(38) 53–56
[10] Sevastyanov A V, Ivanov A A, and Fatkullin A S 2014 Technology of gas removal from oil wells annular area Oilfield Engineering 9 54–55
[11] Kamaletdinov R S 2014 Service of the mechanized well stock Oil and Gas Vertical 7 58-59
[12] Mak-Koi Ch 2004 The gas compressor working from the balance of the pumping unit is useful in various field operations Oil and gas technologies 3 44-46
[13] Valeev M D, and Isanchurin B A 1973 On a non-stationary problem of hydrodynamics of a rod deep-pumping installation Proceedings of BashNIPIneft Ufa 37 25-33
[14] Slezkin N A 1955 Dynamics of a viscous incompressible liquid (Moscow: State publishing house of technical and theoretical literature)