LITERATURE CITED

1. S. Aiba et al., Biochemical Engineering, Academic Press (1973).
2. P. V. Danckwerts, Gas–Liquid Reactions, McGraw-Hill (1970).
3. B. Atkinson, Biochemical Reactors, Academic Press (1974).
4. P. I. Nikolaev and D. P. Sokolov, in: Theory and Practice of Continuous Cultivation [in Russian], Nauka, Moscow (1980), pp. 189-215.

THE EXTERNAL CHARACTERISTICS AND PRESSURE FLUCTUATIONS OF A TWO-STAGE PNEUMATIC-CONTROLLED PUMPING SYSTEM

S. V. Lovchev, F. M. Mir-Kasimov, E. P. Moroz, and L. I. Sokov

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Contemporary methods of drilling oil and gas wells present increased demands on pumping equipment, the main ones of which are for increased pressure and power. To achieve higher pressures in existing well pumps would require increasing their weight, overall dimensions, and drive motor power. In addition, increasing the pressure drop on such removable parts and components of pumps as pistons, valves, sleeves, etc., significantly reduces their service life.

One method of increasing the pumping equipment pressure is to use two or more reciprocating pumps connected in series system designed to divide the pressure drop and power between them. However, in practice this principle can only be accomplished using control devices which can equalize ideally the input to the piston pumps. In order to resolve the question of the usefulness of such pumping equipment a special investigation has been carried out in the All-Union Scientific-Research Institute of Petroleum Engineering.

The investigations were conducted on a rig [1] including two type 11Gr piston pumps connected in series and fitted with a fast-acting pneumatic system for controlling the feed over the entire working range. The present work presents certain results from an investigation of the experimental basis for the possible use of a two-stage pneumatically controlled pumping set to distribute uniformly the pressure drop and power between piston pumps, determining the pump and overall unit efficiencies and the pressure fluctuations in various chambers of the equipment in series pump operation.

During testing, the first pump stage of the experimental equipment was fitted with standard 90-mm-diameter cylinder sleeves and pistons, while the second stage was fitted with 80-mm-diameter components. The use in the first stage of larger-diameter sleeves and working pistons was due to the need to provide the best filling of the second-stage pump cylinders under any working condition. The sleeve and piston separators of the pneumatic controllers for the first and second pumps were, respectively, 120 mm and 100 mm in diameter. The double-acting frequency was 100 per min for both pumps. Process water was used as the pumped fluid. A typeVK-5 pneumatic compensator having a useful gas volume of 5 dm^3 was fitted immediately before the inlet collector of the second pump at the end of the intermediate piping. From the conditions of conducting the experiments designed to achieve uniform power and pressure drop between the pumping stages, the gas pressures in the pneumatic systems of the first and second stages were in the ratio 1:2 and were assigned as follows: 2 and 4; 2.5 and 5; 3 and 6 MPa.

The following parameters were determined during the experiments: pump inlet and outlet pressures; pressures in the working chambers of the pumps; gas pressures in the pump pneumatic systems; gas pressure in the pneumatic compensator at the inlet to the second pump; pump feed rate; indications of the dead points of the piston pumps; power and current for the electric motors of both pumps. The mean pressures were measured by a class 0.2 manometer. In addition to this, during the experiments oscillograph recordings were made of the pressures stated as well as the power and current to the pump electric motors.

The pressures were determined by means of type TDD-2-NATI strain gauge transmitters. The equipment used for pressure recording consisted of two type 8ANCh-7M 8-channel amplifiers and two type K-115

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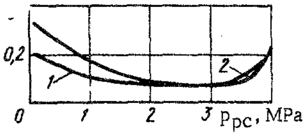

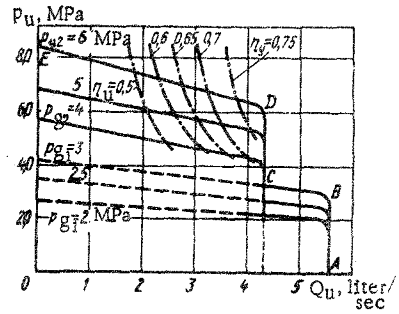

Fig. 1 Fig. 2

Fig. 1. Relation for the relative pressure fluctuation δ and initial gas pressure p_{pc} in the VK-5 pneumatic compensator: 1) at the outlet of the first-stage pump; 2) in the second-stage pump cylinder.

Fig. 2. Relation between the pressure p_{eq} at the pumping equipment outlet and the feed rate Q_{eq} for various values of the gas pressure in the penumatic systems of respectively the first- and second-stage pumps p_{g1} and p_{g2} [$p_{g1} = 0.5p_{g2}$; $p_{pc} = (0.6-0.9)p_{g1}$]: ———) characteristics for pumping set; -----) characteristics for first pump; —·—·) iso-efficiency lines η_{eq} of the pumping set.

oscillographs with type M-1012-1200 loops. Before starting to record the pressure, transmitters were calibrated statically and the results were used to plot calibration curves which were linear over the pressure range measured.

A volumetric method was used to determine the pump feed rate. The times to fill the measuring tanks were measured by stopwatch.

The indications of the "dead" piston positions were made by means of a movable contact located on the piston rod of each pump and two fixed contacts attached to the pump body. Signals to the oscillograph loops were supplied by means of type 1.28−NBMT-525 filament batteries. A type N-102 oscillograph was used to record electric-motor powers and currents.

A timing signal was supplied simultaneously to all oscillograph loops to ensure synchronous recording of powers and pressures. The power loops were calibrated before testing. The active loads used were various selection of filament lamps and spiral resistance elements. The calibration curves were used to determine the power of a single electric-motor phase.

The pump power N was calculated from the equation:

$$N=3N_e\eta_e\eta_к,$$

where N_e is the power required by a single phase of the electric motor; η_e is the mechanical efficiency of the electric motor; η_k is the efficiency of the vee-belt drive. The useful power N_u of the pump is determined from the expression

$$Nu=(Qp_0)/10{,}2,$$

where p_o is the pump outlet pressure; Q is the pump feed rate. The overall efficiencies of the pump η_p and of the equipment were determined by means of the following equations:

$$\eta_p=N_u/N \text{ and } \eta_{eq}=N_{eq}/(N_1+N_2),$$

where N_{eq} is the useful power of the pumping equipment; N_1 and N_2 are respectively the powers of the first and second stages. The relative pressure fluctuations δ were determined from the expression:

$$\delta=(p_{max}+p_{min})/p_{av}$$

where p_{max}, p_{min}, and p_{av} are the maximum, minimum, and mean pressures of the cycle, respectively.

The pumping equipment was started in the following sequence. First, the pneumatic systems of the first and second pumps were filled with air in the pressure ratio 1:2. The first and second pumps were then started with fully open outlet valve. The pump feed was controlled by changing the resistance of the outlet piping by means of the valve.

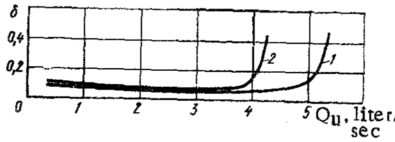

Fig. 3. Relation between the relative pressure fluctuation δ and pumping equipment throughout Q_{eq}: 1, 2) at the outlets of the first and second states, respectively.

Previously obtained universal characteristics for Q−p and η−p served to establish normal functioning and efficiency of each of the pumps in separate working. Additionally, the experiments showed that the relative pressure fluctuation at the outlet of the first and second pumps during their separate operation was quite low (δ = 0.05-0.12) in the feed control range from $Q = 0.85Q_{max}$ to Q = 0.

To check the compensating capacity of the pneumatic controllers, the feed testing with series pump operation was carried out with the pneumatic compensator in the second pump feed piping isolated. Visual observation during pumping operation showed that in spite of the favorable outlet characteristic series operation of the type 11Gr pumps is accompanied by increased vibration of the pipework connecting both stages of the equipment.

From subsequent analysis of the oscillographic recordings of pressures in the different chambers of the pumping equipment it was possible to establish that this effect is due to increased pressure pulsations of the liquid being pumped and is accompanied by nonuniform induction into the second-stage pump cylinders. During the tests this pump worked under comparatively high-pressure conditions at the inlet (up to 4 MPa) and the pressure fluctuation in its cylinders during the suction cycle was considerable (δ = 0.4-0.5).

Therefore, the experiments showed the necessity to have in the inlet collector of the second-stage pump an additional compensating device which should reduce the pressure fluctuations in both the actual pump and also in the piping connecting the first- and second-stage pumps, and should thereby provide for normal operation of pumping equipment where the pumps are connected in series. In order to verify this engineering solution during the course of subsequent experiments, a type VK-5 spherical pneumatic compensator was used, into which gas was pumped at various pressure and oscillographic records were made of pressure at the outlet of the first pump and in the cylinders of the second pump.

Figure 1 shows relations for the relative pressure fluctuations at the outlet of the first pump and in the cylinder of the second pump during the suction cycle as functions of the initial gas pressure in the VK-5 pneumatic compensator. It is evident that the spherical pneumatic compensator fitted at the inlet to the second stage pump can reduce significantly the pressure fluctuations in the fluid being pumped over the whole hydraulic circuit connecting both stages of the experimental pumping equipment. However, the pneumatic compensator has different effects on the flow pulsations in different parts of the circuit.

For instance, small relative pressure fluctuations (δ = 0.08) are maintained in the second-stage pump cylinder even during initial pressures of gas injection in the pneumatic compensator in the range p_{pt} = 2-3.4 MPa which was 54-90% of the pressure at the first-stage pump outlet p_{01} in these experiments. At the same time, as the zone with small relative pressure fluctuations the pressure for the outlet of the first-stage equipment is maintained in a wider range [p_{pc} = 1.2-3.5 MPa (33-95% of p_{01})].

Data relating to pressure fluctuations in the pumping equipment stages during separated and series operation indicate that the pressure fluctuations at the outlet of the first-stage pump are smoothed out during series operation mainly due to the pneumatic controller and to a considerably lesser extent by the VK-5 pneumatic compensator.

Also regarding the pressure-fluctuation reduction in the second-stage pump cylinders during the suction cycle the investigations have shown that in this case the VK-5 pneumatic compensator plays the major role. An initial gas pressure in the VK-5 pneumatic compensator of 60-90% of the first-stage pump outlet pressure is most favorable for decreasing the pressure pulsations of the pumped fluid in the second-stage pump cylinders and also in the pipework connecting both pump stages.

Selection of the appropriate characteristics of the VK-5 pneumatic compensator provided a means of ensuring normal pumping equipment operating conditions. After this the overall characteristics for the set were plotted (Fig. 2) over a wide throughput range for different values of the initial injection pressures for the pneumatic systems of each pump. The relation obtained for the pressure change at the outlet during throughput changes with a pneumatically controlled pump set is determined by the characteristic curves which correspond to a strictly defined value of the initial pressure of the gas in the pneumatic systems of each pump. It has been noted earlier that in these experiments the ratio of the initial gas pressures in the pneumatic systems of the pumps referred to was made equal to 1:2.

The efficiencies of the pumps and equipment were determined under different operating conditions. Analysis of the generalized characteristics showed that the overall efficiency of each pump and the entire equipment is a function of the power and is reduced with both decreasing outlet pressure and with decreasing throughput.

We shall consider the operation of the pumping equipment at, for example, initial gas injection pressure in the pump pneumatic systems $p_{g1} = 3$ MPa (pump No. 1) and $p_{g2} = 6$ MPa (pump No. 2). In accordance with the generalized characteristics the pumping equipment may operate in two different regimes.

The first regime is the startup regime of transient pumping equipment operation. The initial stage of this regime takes place with a fully opened valve on the equipment outlet and is characterized by absence of load on both stages of the pumps (see Fig. 2, point A). This first regime develops further according to the resistance at the outlet from the equipment under conditions of only first pump stage loading. In this period the second pump is unloaded and acts as an additional hydraulic resistance at the outlet of the first pump.

The subsequent two stages of development differ only in the nature of first pump loading, which is loaded at the start in strict accordance with the uncontrolled part of the characteristic (see Fig. 2, region AB) and then it operates in the initial phase of throughput control (see Fig. 2, region BC). In the final stage of the first operating regime the pumping equipment throughput can only be decreased in accordance with the curved characteristic of the first pump.

The second regime is that of established pumping equipment operation and begins to develop from the moment of loading the second pump. This starts in accordance with the uncontrolled region of its characteristic (see Fig. 2, region CD). In the first stage of this regime increasing equipment outlet pressure is accompanied by increasing the load on only the second pump. At the same time, the first-stage pump operates at constant throughput control degree corresponding to maximum capacity of the second-stage pump.

Subsequently, the throughput of the pumping equipment changes in accordance with the DE region of the characteristic (see Fig. 2). Then the ratio of the pressure drops in the pumps over the entire throughput control range is maintained approximately constant and depends on the initial gas pressures in their pneumatic systems. The most rational ratio is when the pressures and powers of the equipment are equal in both pump stages as observed in this actual case.

Consequently, during series operation of pneumatically controlled piston pumps, when increasing pressure at the outlet of the pumping equipment begins to exceed the gas pressure in the pneumatic system of the second pump, the throughput of the equipment is decreased as during operation of an individually operating pneumatically controlled piston pump [2]. It must also be noted that these features of pumping equipment operation in the first and initial stage of the second regime lead to the possibility of using as the high-pressure (last) stage a normal piston pump with a stable characteristic.

However, the expediency of using in the future a similar engineering solution is questionable since the problem of achieving high pressures in borehole pumping equipment is intimately related to improvement of their control characteristics [2]. From this point of view the design of pumping equipment considered here, which includes only pneumatically controlled piston pumps with a curved characteristic is more promising.

The graphical relation between the relative pressure fluctuation and the equipment throughput as shown in Fig. 3 is also evidence of the good possibility of using the pneumatically controlled equipment to obtain high pressures at the outlet with low fluctuations of throughput and pressure. It is evident that small values of the relative pressure fluctuation at the outlet from each equipment stage during series operation occur over a wide control range from $Q = 0.85Q_{max}$ to $Q = 0$. The curve for the relative pressure fluctuation for the first-stage pump is located somewhat below the similar curve for the second-stage pump, and this may be explained by the large volumes of the gas chambers in the pneumatic controller of the first pump.

Consequently, in pneumatically controlled pumping equipment in order to reduce the pressure fluctuations it is necessary to use the largest possible sizes of pneumatic regulator sleeves. For the same reason, during operation of similar equipment with throughputs close to the maximum value it is possible to use a pneumatic compensator at the outlet of the second-stage pump.

LITERATURE CITED

1. S. V. Lovchev, F. M. Mir-Kasimov, E. P. Moroz, et al., "A rig for investigating pneumatically controlled pumping equipment," Khim. Neft. Mashinostr., No. 6, 17-23 (1978).
2. S. V. Lovchev, V. I. Roshupkin, S. L. Zalkin, and V. A. Namaka, Borehole Pumps with Controlled Throughput [in Russian], Nedra, Moscow (1977).