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Control Valve Retaining Laminar Flow

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***Abstract*: The issues of reduction of power consumption are important for any industry, including railway transport. They are solved by numerous approaches where solutions related with decrease in energy loss in transmission channels are the most important. This article describes approach to decrease losses in pipelines upon transmission of compressed air by a new design of control valve characterized by lower losses of compressed air from compressor. This peculiarity makes it possible to maintain lower pressure at compressor outlet, thus decreasing energy consumption for its operation.**

***Index Terms*: control valve, pipeline, gas flow rate, passage, flow coefficient, linear drive.**

I. INTRODUCTION

This article discusses the issue of decrease in pressure generated by compressor stations due to decrease in hydraulic resistance in pipelines [1]. Gas flow rate in pipelines is varied by control valves [2], each of them depending on its design and passage is characterized by its hydraulic resistance ξ. Herewith, the higher is the loss in the valve, the higher is ξ [3,4]. Therefore, using valves with lower coefficients of hydraulic resistances in process cycle, it is possible to decrease cumulative hydraulic resistances of pipelines [5] and, hence, to operate compressor stations with lower generated pressure.

II. METHODS

A. Block Diagram.

A drawback of available control valves is strong deformation of gas flow upon passing via constriction element. Herewith, not only turns of gas flow in valve are considered but also variations of physical structure of the flow comprised of transition from axisymmetric to non-axisymmetric and then reverse to axisymmetric flow structure (gas flow before the valve, in the valve, and after the valve) [6]. It is known [7] that at one and the same effective surface areas of various constriction devices those are preferred in terms of decrease in coefficient of hydraulic resistance which do not turn gas flow and do not vary its structure. Such valve is schematically illustrated in Fig. 1.

The valve is comprised of the plates 1 and 2 suspended on the fixed axle *3*. Each plate has a hole with droplet shape. Herewith, the holes are mirrored with respect to each other. The plate edges are inserted into the guides *4* and 5 of the cam *6* driven by the linear drive 7.

B. Algorithm.

The valve operates as follows. For instance, if it is required to increase gas flow rate in pipeline, the linear drive under the signal p actuates the cam moving it upwards. The cam via the guiding slots *4* and *5* transmits oscillating motion to the plates 1 and *2* to each other in transversal direction to the pipeline axis (passage of inlet and outlet pipeline is dashed in Fig. 1). Herewith, the plate holes are combined forming total passing cross section of the valve. In the case of maximum approach of the plates, the valve is opened completely for gas passage. The holes in the plates form circle of the same surface area as the pipeline passage area.

If, according to the signal p, it is required to decrease gas flow rate in pipeline, the linear drive modes the cam downwards. The cam moves the plates apart via the guiding slots. The combined holes of plates move apart, thus decreasing the total passage area.

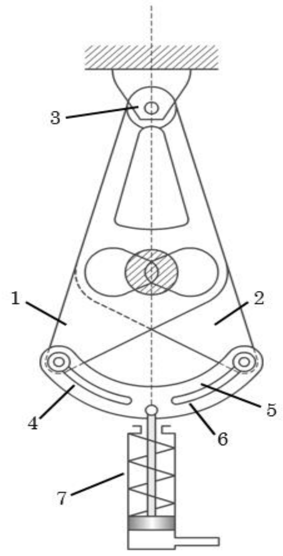

Fig. 1. Schematic view of the control valve: 1, 2 – plates, 3 – axle, 4, 5 – guides, 6 – cam, 7 – linear drive.

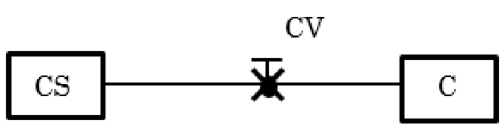

Fig. 2. Schematic view of the communication between compressor station and consumer via control valve.

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The considered principle of valve operation demonstrates possibility to modify the passage shape close to circle; in any case the axis of valve passage coincides with the pipeline axis. The coefficient of hydraulic resistance of the considered valve will be lower than that of any conventional valve with similar passage.

Let us analyze the pressure generated by compressor station as a function of hydraulic resistance of pipeline.

Let us assume that from a compressor station *CS* compressed gas is supplied to consumer *C* (Fig. 2) via pipeline equipped with the control valve *CV*. The consumer uses constant and predefined amount of gas *G*, which is formed by preset valve passage. The equation of gas flow rate in adiabatic process, provided that a short pipeline is used, will be as follows:

$$G = \mu f \sqrt{\frac{2gk}{k-1} p_{CS} \rho_{CS} [(p_C/p_{CS})^{2/k} - (p_C/p_{CS})^{\frac{k+1}{k}}]} \quad (1)$$

where μ is the coefficient of gas flow rate; f is the pipeline passage area; g is the acceleration of gravity; k is the adiabatic exponent (k = 1.4); p_{CS}, ρ_{CS} are the pressure and density of gas, respectively, at outlet of *CS*; p_C is the gas pressure at inlet to *C* (p_C is preset value), ρ is the gas density. The coefficients ξ and μ are interrelated by nonlinear function illustrated in Fig. 3, from where it follows that the higher coefficient ξ corresponds to the lower coefficient μ . Thus, the considered valve will be characterized by lower ξ than any other serial valve with similar passage and higher μ [8]. It should be mentioned that the gas flow rate G in Eq. (1) is preset for generation of pressure p_C at consumer site. In order to retain this value upon increase in the coefficient μ (that is, when the considered control valve is used in the pipeline), it is required to decrease other variables in Eq. (1) [9]. The most suitable variable for solution of this problem is the pressure p_{CS} at the outlet from compressor station [10]. Therefore, while using the control valve with axisymmetric passage in the system aimed at generation of preset gas flow rate at consumer site, it is required to decrease the generated pressure at system inlet (pressure p_{CS}**)**, which leads to decrease in the pressure generated by compressor station and, hence, decrease in power consumptions.

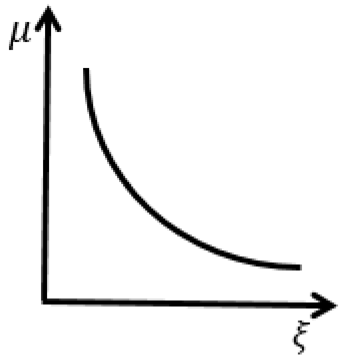

Fig. 3. μ as a function of ξ.

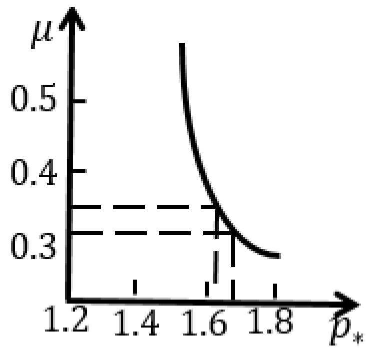

Fig. 4. Graphical solution of flow rate equation.

III. RESULTS AND DISCUSSION

Let us consider the following actual example. Let us assume that the control valve is used in the system. The following variables are preset: $G = 10.6$ m³/s, $p_{CS} = 5.0$ MPa, $\rho_{CS} = 6.0$ MPa, total coefficient of the system flow rate is $\mu =$ 0.3.

Let us write Eq. (1) as follows:

$$\frac{G^2}{\mu^2 f^2} = \frac{2gk}{k-1} \frac{\rho_{kc}^2}{RT_{kc}} [(p_C/p_{CS})^{2/1.4} - (p_C/p_{CS})^{2.4/1.4}],$$

where $\rho_{CS} = p_{CS}/RT_{CS}$.

Denoting the known variables as follows :

$$0.2\,\frac{0{,}2G^2 RT_{kc}}{f^2 gk} = B, p_C^{1.43} = a, p_C^{1.7} = b,$$

it is possible to write:

$$\mu = \pm\left[B/\left(ap_{CS}^{0.57} - bp_{CS}^{0.3}\right)\right]^{0.5} \quad (2)$$

Aiming at simplification of Eq. (2) let us denote:

$p_{CS}^{0.3} = p_*$, since $p_{CS}^{0.57} \approx p_{CS}^{0.6}$

In this case the following is valid for Eq. (2):

$$\mu = \pm[B/(ap_*^2 - bp_*)]^{0,5} \quad (3)$$

The equations can be readily solved graphically. Figure 4 illustrates such solution for positive μ. Let us use common value μ for a system with the valve illustrated in Fig. 1: $\mu = 0.34$ (for a system with conventional valve $\mu = 0.3$).

Then, the respective $p_* = 1.66$ (see Fig. 4). On the basis of the aforementioned:

$p_{cs} = \sqrt[0,3]{p_*} = 5.4$ MPa.

Therefore, while using the considered control valve in the compressed gas line, it is possible to decrease the gas pressure generated by compressor station in the considered example from 6.0 to 5.4 MPa.

IV. CONCLUSION

This article considered air as the working medium, however, general conclusions concerning reasonability of commercial application of control valve of the considered type can be applied also to hydraulic systems.

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