EFFECT OF CONNECTED SYSTEM ON GAS PRESSURE CONTROLLERS' STABILITY

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In the paper the influence of dynamic performances of pipe systems connected to the control valve on system stability is studied through mathematical modeling and experimental work. Dynamic interaction between typical pressure control valves and up- and downstream pipelines is investigated. Stability criteria are derived in the form of requirements for input impedances of pipes connected. The best and the worse boundary conditions are deduced for each type of controllers. The best boundary conditions ensure stability even in a case of zero damping of the poppet. The worse boundary conditions determine maximum damping required to keep stability in any other system. To meet the requirements received, effective measures are suggested for each type of the control valve on the basis of special corrective devices (compensators) of the acoustic filter type. A scheme, parameters and location of the devices are determined by the type and parameters of the valve.

Key words: *Control valve, stability, boundary conditions, correcting device, acoustic filter, oscillations damper, pipe system*

1 Introduction

Complexity of modern gas and hydraulic control systems makes it difficult to predict their dynamic properties at the design stage. During experimental finishing of systems, instabilities frequently occur accompanied by pressure disturbances, valves oscillations, noise and vibration being increased without sources of forced oscillations [1, 2, 3 and 4].

There are two principal approaches for correction system dynamics to satisfy stability demand:

- to affect the regulator dynamics;
- to affect the pipework dynamics.

The first way is traditional and mostly developed. However, in a lot of cases the other one is more simple and effective. This approach is based on the essential influence of coupled lines on the fluid regulator operation. The dependence is especially strong in respect to the simplest devices such as fluid control valves [5 and 6]. Interaction between a pressure - reducing valve and the upstream and downstream pipes and its influence on stability are considered in [7 and 8].

The principal possibility to stabilize pressure controllers by the pipe-line response correction is shown in [9, 10 and 11]. As effective means for the system performance correction there are presented special devices of acoustic filter type previously developed for suppression forced oscillations [12, 13 and 14]. Now in-line suppressors are commonly used in industry for broadband pressure ripple reduction and its predictive models are available in the literature [15 and 16]. Having small dimensions and mass these devices are characterized by design simplicity and reliability, their inserting into a system does not lead to great changes in the arrangement. The theory of design and application of such devices being developed for suppressing forced oscillations can't be used for solving stability problems without additional research. An operating principal, an efficiency criteria and design approaches are different depending on the problem. So suppressing forced oscillations is based on the correcting devices' ability to dissipate and to redistribute flow oscillations energy [9, 12, 13 and 17]. The efficiency to eliminate self-excited oscillations is defined by regulators' sensitivity to variations of boundary conditions and by the correcting devices' ability to form desired pipe lines dynamic characteristics [10, 11].

Up today the wide theoretic and experimental researches of correcting devices are carried out (for ex. [9, 10, 11, 12, 14, 17 and 18]). Mathematic models, design algorithms and efficiency criteria are developed for some structures of controllers [19, 20 and 21]. Tens of correcting devices and oscillations dampers are created and successfully operates in gas and hydraulic systems of aircrafts and space vehicles. However, some problems restrain wider application of the correcting devices.

To make correct choice of a scheme, parameters and location of the devices it is necessary to have adequate data about the domain of the acceptable variations of dynamic characteristics of the connected pipes to satisfy stability criterion. In the paper, on a basis of a general approach stability criteria are derived for typical pressure controllers in the form of requirements for boundary conditions. Some simplest compensators satisfying stability criteria are proposed.

2 Stability analysis

The linear model of the system fragment shown in Figure 1 can be converted to the form

$$\begin{bmatrix} \overline{p}_3 \\ \overline{\dot{m}}_3 \end{bmatrix} = \left[B(s)\right] \begin{bmatrix} \overline{p}_1 \\ \overline{\dot{m}}_1 \end{bmatrix} \tag{1}$$

$$\overline{\dot{m}}_1 = -Y_1(s)\overline{p}_1,\;\; \overline{\dot{m}}_3 = Y_3(s)\overline{p}_3, \tag{2}$$

where $B(s)$ - a transfer matrix of the valve, $Y_1(s)$, $Y_3(s)$ - input admittances of upstream and downstream circuits; $\overline{p}, \overline{\dot{m}}$ - small relative perturbations of pressure and mass flow rate; s - Laplace operator.

The system characteristic equation then can be written as

$$aY_1 + bY_1Y_3 + cY_3 + d = 0, \tag{3}$$

where a, b, c, d=f(s) - real polynomials, components of the matrix [B] defined by structure and parameters of the control valve.

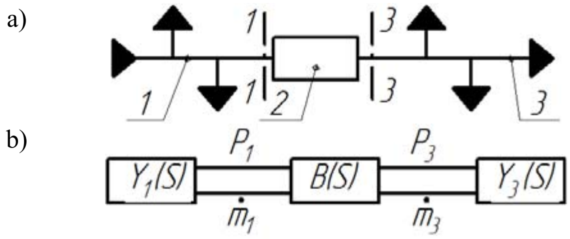

Figure l. Principal (a) and calculation (b) scheme of a closed loop system:
1-upstream pipework, 2- regulator, 3-downstream pipework.

For resistive loads the input impedances Y_1 and Y_3 have real values. Then the Routh criterion applied for equation (3) gives a stability domain in the Y_1OY_3 plane.

For complex input impedances frequency criteria are used. In a general case, when transfer functions of the input impedances are of a type

$$a_{ij}s^i \exp(st_j) \tag{4}$$

The method of D-decomposition is applied to determine stability domains in the complex planes $Y_1(j\omega)$ or $Y_3(j\omega)$. Assuming resistive input impedance of one part of the system, the domain boundary in the complex plane of other part can be described by

$$Y_{1*}(j\omega) = -\frac{c(j\omega)Y_3 + d(j\omega)}{b(j\omega)Y_3 + a(j\omega)}, \tag{5}$$

or

$$Y_{3*}(j\omega) = -\frac{a(j\omega)Y_1 + d(j\omega)}{b(j\omega)Y_1 + c(j\omega)}. \tag{6}$$

In the case when transfer functions may be approximated as

$$Y(s) = m(s)/n(s) \tag{7}$$

where $m(s)$ and $n(s)$ are real polynomials, the Nyquist criterion seems more appropriate.

Then the open loop transfer function with the respect of (5) and (6) can be modified as follows

$$W_1(s) = -Y_1(s) / Y_{1*}(s)$$
$$W_3(s) = -Y_3(s) / Y_{3*}(s).$$

The Nyquist criterion thus can be formularize in the form of requirements for permissible location of pipeline characteristic $Y_i(j\omega)$ referred to the regulator boundary line $Y_{i*}(j\omega)$. Such a form of stability conditions makes it possible to looking for the way of correction separately for each fragment of the system and to use for each one both theoretic and experimental description.

The approach suggested was applied in the research of the systems having the single-stage pressure regulator (Figure 2). The study was limited by the type of passive connected pipeworks having no sources of acoustic energy, which input impedance characteristics are located on the left half plane ($Re\ Y(j\omega)>0$). The experiments were carried out with regulators of aircraft and space systems and air as fluid. The tested regulators were adjusted to decrease friction and to fit pressure and displacement transducers. To obtain various values of input impedances in the experiment there were used pipes of a wide range of lengths and diameters, bottles of variable capacity and throttles of porous material having performances closed to linear ones at a small flow rate.

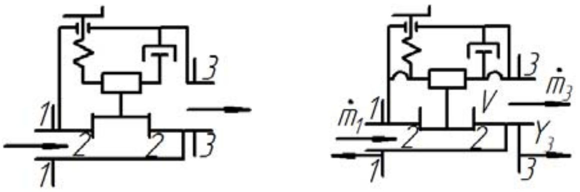

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Figure 2. Diagrams of upstream (a) and downstream (b) pressure control valves

In the course of the investigation the next main results were obtained [10, 11 and 18]:

1. In the plane of the lines' input impedances there is a stability domain, whose sizes tend to be smaller with decreasing the valve's damping factor ζ, but that does exist even with $\zeta=0$ (assuming the natural operator of the spring-mass unit is $T^2s^2+2\varsigma Ts+1$). For the case of a gas pressure reducer the stability domain is situated at so high values of the admittance Y_3 (Figure 3), which is too difficult to obtain. For a relief valve as an upstream pressure regulator the stability domain expands to low values of the admittances Y_3 and Y_1, as a rule not available in practice because of a high pipe capacity in such systems.

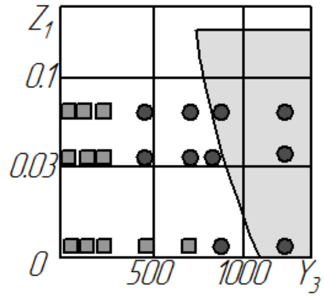

Figure 3. Stability domain of the pressure reducing valve with a resistive load:
- theoretical stability domain, experimental results
-stability, -instability

2. Analysis on the complex planes $Y_3(j\omega)$ and $Y_1(j\omega)$ shows that the response of downstream pipework $Y_3(j\omega)$ has the greatest influence on the pressure reducer stability, and the response of upstream pipework $Y_1(j\omega)$ - on the relief valve stability.

In the general case of passive systems with distributed parameters (transfer function as (4)) the stability domains are located to the right of the curves $Y_3(j\omega)$ and $Y_1(j\omega)$ (Figures 4 and 5) (the curve's branches for $\omega<0$, symmetric relatively real axis, are not shown; $\omega=\omega/\omega_n$, where ω_n – the natural frequency of the spring-mass unit). The ranges of each domain are dependent on other pipe admittance: with decrease of Y_1 (Fig.4) or Y_3 (Figure 5) the boundary lines shift to the left broadening the stability domains. However, a small extension here causes great losses in pipe capacity.

3. Increase in damping ratio leads to essential widening of the stability domain. At the certain value $\zeta=\zeta_0$ the curve $Y_{3*}(j\omega)$ (Figure 4) may be shifted to the left down to the point $Y_3=1$. It is the lowest real Y_3 value, sonic flow conditions being established in the outlet throttle. At such value ζ (in our example $\zeta_0=0.35$) the stability domain spreads nearly all over the right half-plane, and the pressure reducer will be stable practically with any dissipative pipework.

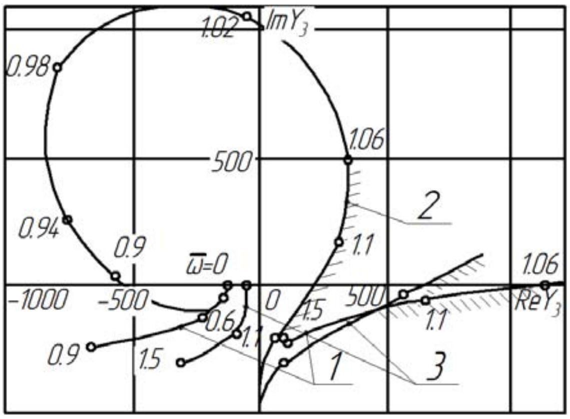

Figure 4. Stability domain of the reducer on the complex plane of input admittance $Y_3(j\omega)$ of a downstream pipeline: 1 - $\zeta = 0$, $Y_1 = 1000$; 2 - $\zeta = 0.05$, $Y_1=1000$; 3- $\zeta = 0$, $Y_1 = 5$

The boundary conditions $Z_1 = 1/Y_1 = 0$ and $Y_3 = 1.0$ should be considered as the worst type of coupled pipelines when the highest value of inherent damping of the pressure reducer is required for the system stability.

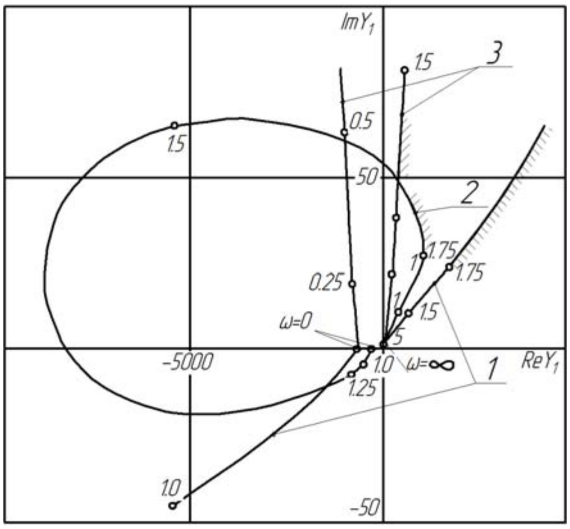

Figure 5. Stability domain of the relief valve on the complex plane of input admittance $Y_1(j\omega)$ of an upstream pipeline: 1- $\zeta = 0$, $Y_3 =1000$; 2- $\zeta = 0$, $Y_3=100$; 3- $\zeta = 0.05$, $Y_3 = 1000$

As for the relief valve it should be accepted zero values of the input impedances $Z_1 = Z_3 = 0$ for the worst boundary conditions.

4. For the special case of systems, the transfer function of which can be polynomial approximated as (7), the next stability condition is derived. In accordance with the Nyquist criterion to stabilize pressure reducer it is quite enough to place not all the curve $Y_3(j\omega)$ to the right of the locus $Y_{3*}(j\omega)$ (Figure 4) but its definite part in a range between frequencies at the crossing-points of the curve $Y_{3*}(j\omega)$. For stabilizing system having the upstream pressure regulator only that part of the curve $Y_1(j\omega)$ must be situated to the right of $Y_{1*}(j\omega)$ (Figure 5), which begins from a frequency at the point of intersection the locus $Y_{1*}(j\omega)$.

5. Different results are obtained for the check valve [11]. Its diagram is similar to the relief valve's (Figure 1 a) and a distinctive feature is a low pressure drop at a high inlet pressure. For this reason the upstream and downstream pipework has almost similar impact on system stability. The worst boundary conditions for the check valve are zero values of the input impedances $Z_1 = Z_3 = 0$. In this case the valve stability is determined only by damping in the spring-mass unit. For the general case of transfer function (4) an increase of resistive component and decrease of reactive one lead to widening stability margin.

These results may be used as initial data for system correcting in various cases of the regulator arrangements.

3 Measures for dynamic correction

The requirement features of pipework characteristics can be obtained so by choosing pipeline parameters, as by inserting into system special corrective devices. The first approach uses as a rule, on the design stage, when creating former dynamic properties permits to avoid many difficulties on the testing stage. Such an approach being employed on the operational development stage is not always effective and needs many system reconstructions. On this stage it is more beneficial to use corrective devices.

In some cases the task can be easy solved by the simplest means as throttles, bottles etc., using features of a concrete pipework coupled. So, when a regulator is loaded with a pipe ended by a bottle, a throttle with the resistance equal to wave pipe impedance being placed at the open end makes the input impedance real. As it follows from stability domain configurations such a load matching can be sufficient for stabilizing a relief valve. To stabilize a pressure reducer it is necessary besides that the load input admittances should be of its stability domain (Figure 3).

These requirements in respect to a downstream line are easily carried out when a pressure reducer operates with a bottle as a load. Then the proper value of admittance can be obtained by putting a throttle with definite resistance at the reducer outlet.

It should be noticed that an additional steady state error of control arising due to a cascade setting of throttles up to corrector structures is usually not a large in view of a low input impedance required for a controlled pressure line (at the experiments the pressure drop on the throttle did not exceed 0.3% of the controlled pressure).

In many systems the pressure reducer load has a low value of input admittance lying far out of the stability domain. It may be essentially increased only by inserting into the pipework the devices with capacitive elements. The simplest ones among that are devices of *R-C* (resistance - capacitance) and *R-L-C* (resistance - inductance - capacitance) types that give an opportunity to increase the real part of admittance in a definite frequency range.

Figure 6 shows the gain-phase characteristic $Y_H(j\omega)$ of the input admittance of the *R-L-C* type device (the Helmholtz resonator). The curve has a circle form with a diameter in inverse ratio to the resistance *R* of the resonator throat. The *R - C* type device characteristic looks like upper part of the circumference in Fig.6.

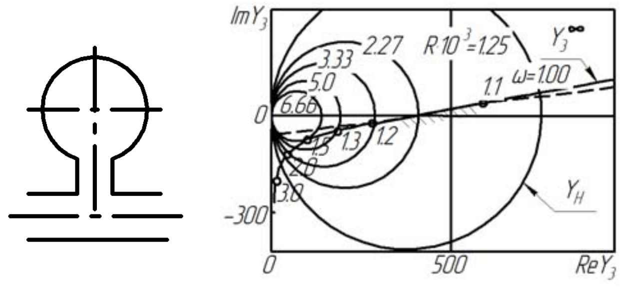

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Figure 6. Diagram of resonator (a) and gain-phase characteristics (b) of the reducer and the resonator at various values of a throat resistance ratio $\overline{R} = R \cdot \dot{m}_{max} / p_{30}$

For calculating of such devices installed at the beginning of the downstream pipeline it can be assumed $Y_3(j\omega) = Y_H(j\omega)$. So putting the $Y_H(j\omega)$ response on the locus $Y_{3*}(j\omega)$ one can see that beginning from the definite value of *R* the parts of the $Y_H(j\omega)$ curves find itself to the right of $Y_H(j\omega)$ even in the case of the worst upstream pipeline at $Z_1 = 0$ $(Y_1 \rightarrow \infty)$. Estimating the coordinates of the crossing points and using stability conditions above one can determine the values of *C* and *L* for the each value of *R*, and then the design parameters of the devices can be established with following calculations of the volume (*V*) of the chamber and the length to diameter ratio of the throat (*l/d*).

The investigations show that using the devices of a resonance type (*R-L-C*) gives an opportunity to make design of smaller overall dimensions than in the case of *R-C* type devices.

For further decreasing of the dimensions or increasing the stability margin the resistance may be installed parallel to the resonator throat.

Figure 7 presents the theoretical and experimental results for the pressure reducer studied. At the chamber volume of the reducer's sensing element $50 \cdot 10^3$ mm^3 the stable operation was reached at the same volume of the resonator chamber: $50...80 \cdot 10^3$ mm^3.

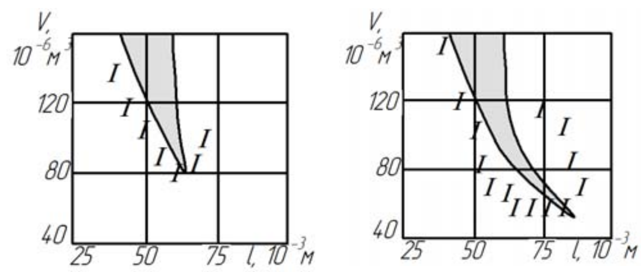

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Figure 7. Stability domain of the pressure reducer in a plane of resonator parameters:
a - $R=4 \cdot 10^6$ $1/m/s$; b - $R=7.5 \cdot 10^6$ 1/m/s; (grey) - theoretical boundary of stability domain; ***I*** -transition zone between stable and unstable behavior (experiment)

The great advantage of such devices besides design simplicity and small dimensions is the absence of additional control error.

To obtain stability of the check valve having small inherent damping by action on the boundary conditions is the challenging task. Mostly it can be solved by the proper valve design. If such approach is not sufficient then more complicated devices of an acoustic filter type should be used. Some of them are considered in [11, 12 and 18]. The great advantage of some circuits is that their designing are independent of pipework dynamics but definite efficiency can be guaranteed even at the worst load.

In our case the devices with resistive wave impedance are of the most interest (Figure 6).

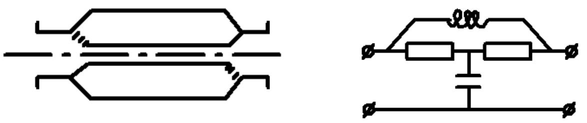

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Figure 8. The diagram (a) and equivalent circuit (b) of the correcting device with resistive wave input impedance

As it was shown in [12] its characteristic impedance can be matched over a wide frequency range to a resistive type being constant and independent of coupled pipework. Application of this type correcting device for solving stability problem was considered in [22].

4 Conclusion

1. ***Stability criteria*** for gas pressure controllers are derived in terms of demands for ***input impedances*** of pipes connected.
2. ***The stability conditions*** derived give an opportunity to determine parameters of the controller being stable with arbitrary system so as to determine system frequency response suitable for the given controller.
3. Stability of gas pressure controllers can be obtained by means of ***external devices*** even in the case of ***zero damping*** of mechanical parts.
4. It is derived ***the best*** and ***the worth*** boundary conditions for each controller’s type:
   - once ***the best*** boundary condition is realized then ***zero damped*** controller will be stable;
   - once the controller has ***enough inherent damping*** to be stable in ***the worth*** boundary condition then it will be stable in any other system.
5. Gas pressure reducer can be stabilized by ***Helmholtz resonator*** with by-pass restrictor.
6. In the simplest cases of the systems, stability of gas pressure reducer /relief valve can be obtained by ***the restrictor*** placed at the outlet /inlet.

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