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**Dynamic research of reduction valve**

*The article is purposed to show the principle of work of the reduction valve as an example of linear automatic control system. We are going to consider the constant pressure control device dynamics.*

The reduction valve is represented as a constant pressure control device in the given diagram. In such conditions, a low internal resistance of a pressure source for the whole range of fluid flow variation is assumed. This value of internal resistance is taken independently on the time and the load parameter. The linear throttle of the variable cross-section which depends on the time only is chosen as a load of the valve in Fig.1. and represented as the sum of the constant and variable components $f_{th} = f_0 + f\ (t)$.

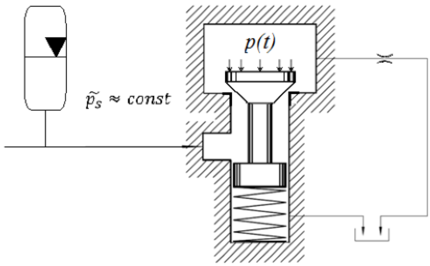


Figure.1 The linear throttle as a load of the valve.

The load linear throttle characteristic inclination $p_H = f(Q)$ must be positive, because at zero and negative characteristic inclination it loses its properties maintain the constant pressure. The variable pressure reducing transition process at the valve outlet after the step reduction in time of the load throttle cross-section is shown in Fig.2. The pressure in the reducing zone rapidly rises because the valve moves downward.

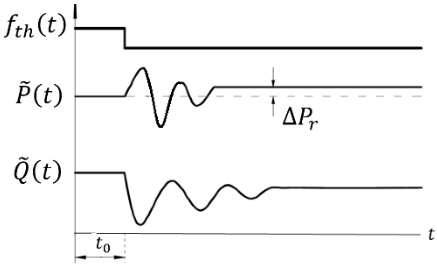


Figure.2 The step reduction in time of the load throttle cross-section.

The fluid flow reduction through the load throttle (third graph in Fig.2) and along with it the reduction of pressure in the cavity of the reducing valve $p_r(t)$ is the second reason of the valve displacement downward.

**The continuity equation of the reduction valve control volume.** Let distinguish the open deformable control volume of the reducing cavity between the cross sections of the throttle control slot of the valve (Fig.1) and before the throttle of the load, have a form:

$$\frac{\mathrm{dM(t)}}{\mathrm{dt}} = \mathrm{M}_{\mathrm{c.\ in}}(\mathrm{t}) - \mathrm{M}_{\mathrm{c.\ out}}(\mathrm{t}) \quad (1)$$

The equation of forces is the second major equality of the dynamics descriptions of the object. The pressure reducing force acts and directed downward on the sliding shutter with mass $m$ in Fig.1. According to the law of momentum change (Newton's second law), the equation of forces projected on the X axis is written as:

$$-\left(\mathrm{p_{r0}} - \mathrm{p(t)}\right)\mathrm{F} + \mathrm{C}\left(\mathrm{h} - \mathrm{x(t)}\right) - \mathrm{K_B}\frac{\mathrm{dx(t)}}{\mathrm{dt}} = \mathrm{m}\frac{\mathrm{d^2x(t)}}{\mathrm{dt^2}} \quad (2)$$

The phenomenon of reducing in the operating point (Fig. 3) depends on the strength of the spring pre-tensioning and the valve saddle area.

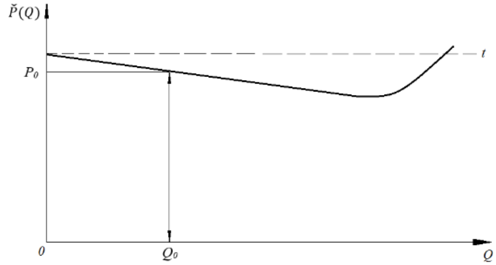


Figure.3

The pressure substitution to the left part of the equation removes the constant components $p_{r0}F = Ch_0$ that do not contain the perturbations *f (t):*

$$m\frac{d^2x(t)}{dt^2} + K_B\frac{dx(t)}{dt} + Cx(t) = -Fp(t) \quad (3)$$

Differential equation of the second order for the variables of zero initial conditions $x_0 = 0$ cannot be integrated, as its right part is an unknown function of the time. Let use Laplace transform to solve this equation. The image of original is denoted by the capital letter X(S) in the operator function $S$ = d/dt the most important property of which is written by three equalities:

$$\int_0^\infty x(t)e^{-St}dt = X(S); \int_0^\infty \frac{dx(t)}{dt}e^{-St}dt = sX(s); \int_0^\infty \frac{d^2x(t)}{dt^2}e^{-St}dt = s^2X(s) \quad (4)$$

Operator $S$ = d/dt due to integration with respect to time is considered as constant. After simplification we get the first transfer function of the linear model of the reducing valve in the form of rational fraction:

$$W_x(S) = \frac{X(S)}{\varphi(S)} = \frac{b}{(a_3s^3 + a_2s^2 + a_1s + a_0)} \quad (5)$$

The second transfer function describing transient process changes of pressures reduction in the cavity is obtained by the removing of the image X (d) in favor of the image (Ps) of the system equations. We get image X(s) as:

$$X(s) = -\frac{FP(s)}{ms^2+k_b s+C} \mp -\frac{F}{\theta(s)} P(s)\ . \qquad (6)$$

In this formula the three-term square is designated by symbol $\theta(s) = ms^2 + k_b S + C$. As a result, the equality can be written as the second transfer function of the reducing valve by pressure (TFP)

$$W_p(s) = \frac{P(s)}{Y(s)} = -\frac{b_2 s^2+b_1 s+b_0}{a_3 s^3+a_2 s^2+a_1 S+a_0} \qquad (7)$$

From the theory of linear differential equations stability is provided by fulfill of a system of inequalities for the third order equations:

$$a_0 > 0;\ a_1 > 0;\ a_2 > 0; a_3 > 0;\ a_1 a_3 > \ a_0 a_2; \qquad (8)$$

The fulfill of inequality does not provide the dynamic quality of the object. It is necessary to choose such combinations of coefficients of the denominator of transfer functions at which dynamic quality will be the best. Small overshoots or dips of pressure reduction and minimum time required for a step perturbation.

The proof of the fifth Hurwitz inequality follows from the analysis of denominators (5) and (7) by the frequency method. In the first transfer function valve lift of the transfer function by transition (TFT), there is no differentiation operator in its numerator and therefore is convenient to choose for frequency analysis this function. The replacement of the differentiation operator with an imaginary unit corresponds to the choice of the harmonic perturbation of the throttle slit in the valve load $F(t) = f_0 + f_m \sin(\omega t)$ for some open mid-rise $x_0$ operating point on its head characteristics (Fig. 3) by substituting $S = jw$ in denominator (5) we can select the current $D(w)$ and imaginary parts $iM(w)$. So, we get simple expressions of the active and imaginary parts of TFT depending on the circular frequency $\omega$ :

$$D(\omega) = \frac{a_{(\omega)} b_0}{a^2_{(\omega)} b^2_{(\omega)}}\ ; \quad M(\omega) = \frac{b_{(\omega)} b_0}{a^2_{(\omega)} b^2_{(\omega)}}\ ; \qquad (9)$$

**Nyquist diagram, amplitude and phase frequency characteristics.** It gives us possibility to determine the stability and quality of transient process. Thus, at zero frequency, the partial values of the module consist only of the real part $W_{(j_0)} = \frac{b_0}{a_0}$.

At low frequency $\omega < \omega 90\,, b(\omega) << a(\omega)$ , the real part does not decrease significantly, and the imaginary part grows rapidly with increasing frequency. The phase angle is negative. At medium frequencies in the range, the value of the TFT module can significantly increase due to the imaginary part, with a small value of difference $a_3 \frac{a_0}{a_2} - a_1$. At high frequencies in the range $\omega_{180} - \frac{\pi}{4} < \omega < \omega_{180} + \frac{\pi}{4}$ TFT module at another small value of difference $a_3 \frac{a_0}{a_2} - a_1$ also increases, but due to the increase of the real part of the TFT in the frequency $\omega_{180}$.

## Conclusion

1.) The reduction valve is represented as a constant pressure control device. It keeps a constant fluid flow pressure after the valve and before the load in the case of the control device and its load is connected in series. 2.) In order to describe the dynamics of the considered object it is possible to use two main hydro pneumatic equations: the continuity equation of the system; the equation of forces of the sliding valve; 3.) A significant increase in fluid compressibility coefficient at small values, significantly reduces the stability of the regulator, and in the medium and large values, practically does not change region of stability or even expanding it; 4.) When designing and calculating regulators, it is recommended to select the values of the areas, located on the right branch of the stability curve, since this provides a wide band of operating frequencies.

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