TEKA Kom. Mot. i Energ. Roln. – OL PAN, 2011, 11B, 132-142

STATIC CHARACTERISTICS OF VALVE – AMPLIFIERS FOR PNEUMATIC DRIVES OF MECHANICAL SYSTEMS

Valentin Remen, Alexsandr Vylich, Michail Lordkipanidze

Volodymyr Dahl East-Ukrainian National University, Lugansk, Ukraine

Summary. The static characteristics of pneumatic valves - amplifiers of drives of mechanical systems with the improved response curves are reviewed. The relations discharge coefficients μ_1 and μ_2, rigidity of seals and membranes $c_{км}$, $c_{кс}$, $c_к$, and hydrodynamic force **Pг** are by practical consideration are retrieved. The approximating model for determination of discharge coefficients μ is obtained and its adequacy also is determined.

Key words: valve-amplifiers, pneumatic drive, discharge coefficients.

INTRODUCTION

One way of the solution of increase of technical and economic efficiency of mechanical systems is the development and research of new effective devices of monitoring pneumatic drives, that can ensure essential positive effect.

The pneumatic drives with valves - amplifiers with different control and different negative feedbacks find wide use in the mechanical systems as control elements from the small moved masses and practically absence of friction forces, that provides high dynamic properties [1].

Valve-amplifier with pneumatic control and pressure and position negative feedback is the base element [2]. For providing high dynamic qualities it is necessary to choose correctly the geometrical sizes of elements, define influence of energy parameters on its work. From this point of view it would be useful to analyze the valve-amplifier work on the basis of its mathematical model [3].). However, with determination of some values, which are included in the system of equation certain difficulties appeared, related to impossibility of analytical way determination of calculation values. The discharge coefficients μ_1 and μ_2, rigidity of seals and membranes $c_{км}$, $c_{кс}$, $c_к$, and hydrodynamic force $F_{гд}=G_2v$ cover to them. Therefore, in this work the decision about e coefficients determination experimentally is accepted, receipting their approximation dependence on the process parameters and using them in mathematical model.

The valve-amplifier with pressure negative feedback (fig.1) is considered in this paper

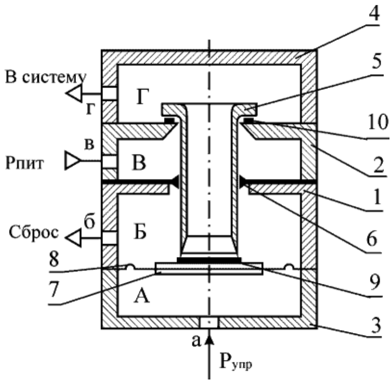


Fig.1 Valve - amplifier

This valve - amplifier is represented by the analogue pneumatic amplifier, in which one

$$p_{вих} = k\, p_{вх},$$

where: $p_{вых}$- upstream pressure (chamber Г),

$p_{вх}$ - pressure of control (chamber A),

к - gain dependent on a ratio of the effective areas of a diaphragm 6, diaphragm 8 with rigid center 7, diameter of the valve.

For maintenance of high dynamic qualities it is necessary correctly to select the geometrical sizes of its elements, to determine influencing on its activity of energy parameters. With this purpose the operational analysis of the valve - amplifier is conducted on the basis of its mathematical model. [3]. However, with determination of some values logging in equations there were definite difficulties, bound with impossibility of determination by an analytical way of their design values. The flow coefficients $\boldsymbol{\mu_1}$ and $\boldsymbol{\mu_2}$, rigidity of seals and diaphragms $\mathbf{c_{км}}$, $\mathbf{c_{кс}}$, $\mathbf{c_{к}}$, $\mathbf{c_{м}}$ and value of a hydrodynamic force **Fгд = G2v** concern to them. The basic purpose of experimental researches of pneumatic valves - amplifiers for mechanical systems of control fnd regulation was:

- Determination of factors of a mathematical model, which one can not be determined analytically or on the basis of a reference data.
- Determination of a degree of adequacy to a mathematical model to experimental data.

RESEARCH OBJECT

For research of static characteristics of the valve – amplifier, determination of flow coefficients $\boldsymbol{\mu}$, viscous friction $\boldsymbol{\alpha}$ and hydrodynamic force $\boldsymbol{\psi}$ the special experimental installation was designed.

The outcomes of static tests are shown on fig. 2 -12.

The relation of output pressure $\mathbf{p_{вых}}$ and flow $\mathbf{Q_{вых}}$ from pressure of control $\mathbf{p_{упр}}$ at different pressure of a feed $p_{пит}$ and different diameters of conditional passage $\mathbf{d_y}$ is adduced on fig.2 and fig.3.

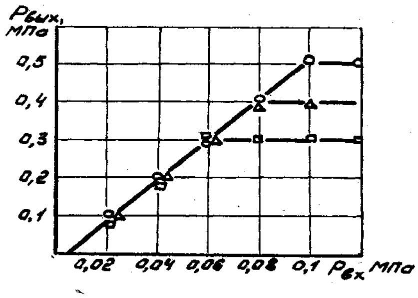


Fig. 2. Relation of delivery pressure from pressure of control at different pressure of a feed

Experiments show (fig.2), the output pressure $\mathbf{p_{вых}}$ linearly depends on pressure of control $\mathbf{p_{упр}}$ and practically does not depend on the geometrical sizes of valves - amplifiers (diameter of conditional passage), i.e:

$$p_{вых} = k_p\, p_{упр},$$

where: k_p- a pressure gain, in this case $k_p = 5$.

Dead zone Δ in all measurement range and for all diameters of conditional passes during tests did not exceed 0,01 МПа, which is quite acceptableas the minimum value of an analog signal of pressure is peer 0,02 МПа.

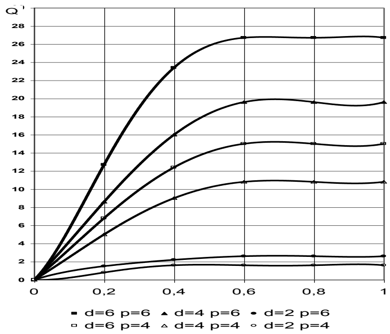


Fig. 3. Relation of the output flow **Q** (л/с) from pressure of control $\mathbf{p_{упр}}$ (ат) at different pressure of a feed $\mathbf{p_{пит}}$ and diameters of conditional passage **dy**

As seen from fig.3, at some pressure of control ($\mathbf{p_{упр}}$ = 0,04 … 0,05 Мпа) the output flow is not augmented, and remains by a constant at a further pressure buildup of control. Up to this value relation of the output flow from pressure of control has linear nature, i.e.

$$\mathbf{Q_{вых} = Kq\, p_{упр}},$$

In a fig. 4 the relations of the output flow $\mathbf{Q_{вых}}$ (l/c) from a height of valve lift $\mathbf{h_к}$ (mm) are shown at different diameters of conditional passage $\mathbf{d_у}$ and parameters of a gas stream $\boldsymbol{\beta}$.

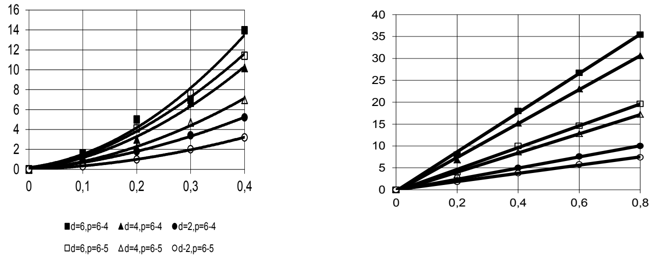


Fig. 4. Relation $\mathbf{Q_{вых}} = f(\mathbf{h_к})$ at $\boldsymbol{\beta} < \boldsymbol{\beta_{kp}}$. and $\boldsymbol{\beta} > \boldsymbol{\beta_{kp}}$

Φs seen from the graphs fig.4 in a case β <βkp the relation of the output flow to a height of valve lift at miscellaneous dy has practically linear nature in an effective range of heights of valve lift, i.e. hк > 0,1мм.

Flow coefficient μ is a composite function dependent as on geometry of the valve - amplifier, and from flow parameters of gas. Generally flow coefficient does not depend from value of a valving slot type gasdynamical of process (supercritical and subcritical flow), difference of pressure, flow regime etc. More often flow coefficient μ is on t published in [4-6],] or on computational relations [7, 8], obtained by results of researches. However, these relations are applicable only to particular pneumatic instrumentation, or for calculation of a flow coefficient μ It is necessary to know other hydraulic parameters (for example factors of local resistances ξ), which one are in turn determined also by experimental way.

For experimental determination of a flow coefficient μ the technique was selected, that the volume flow is determined by experimental way, and the flow coefficient **μ** was calculated by a relation of the experimental flow to idealized, computed on the formula of the Saint-Venant, i.e.:

$$\mu = \frac{Q_{ýêñ}}{Q_{ô}} = \frac{m_{ýêñ}}{m_{ô}} .$$

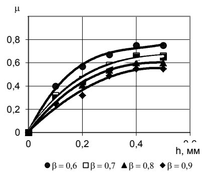
It is possible to approximate and to use experimental values of the flow depending on a difference of pressure, altitude of a slot and diameter of conditional passage then in a mathematical model [3]. Such way of determination of a flow coefficient is known and enough is simple, for its determination the presence of pressure gauges, flowmeters and precise meter of a height of valve lift is necessary only. The solution in the given activity about determination of flow coefficients **μ**, viscous friction α and hydrodynamic force ψ by practical consideration therefore was accepted, and then obtaining of approximating relation.

The method and procedure of the experiments were as follows :

The range of paramttres:

1. Diameters of conditional passages - **dy** = 2, 4, 6 mms.
2. Pressure of a feed (exuberant) - $p_{пит}$ = 0.2, 0.3, 0.4, 0.5 МПа.
3. Back pressure (exuberant) - $\mathbf{p_{пр}}$ = 0.1, 0.2, 0.3, 0.4 МПа.
4. Height of valve lift -**hк** = 0.1 … 1.0 mms.
5. Pressure ratio -**β** = 0.1 … 0.9.

Relations of a discharge coefficient **μ** = f (**h**$_к$) at different parameters of a gas stream (**β**$_{Кр}$) are shown in a fig. 5-6 and fig..7-8.



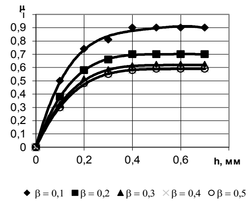
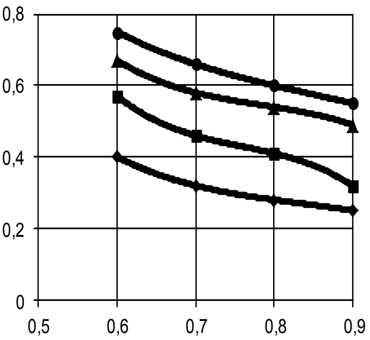


Fig.. 5. **μ=f (h$_к$)** at **β<β$_{кр}$**

Fig. 6. **μ=f (h$_к$)** at **β>βкр**



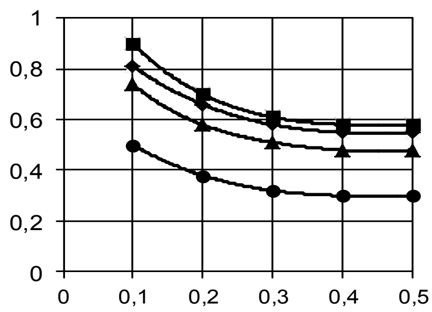


Fig. 7. **μ=f (β)** для **β>β$_{кр}$**

Fig. 8 **μ=f (β)** при **β< β$_{кр}$**

Relations of a discharge coefficient **μ** of flapper – nozzle from parameters of a flow **β**$_{Кр}$ and the altitudes of a slot **hк** are shown on fig. 9,10,11,12.

As seem from the fig. 5 -12 the discharge coefficient **μ** does not depend on diameter of conditional passage of valves - amplifiers, and depends on pressure of a feed, and this relation has linear nature.

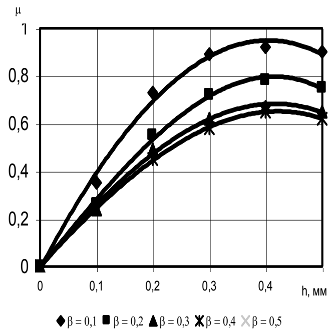


Fig. 9. **μ=f (h$_{ш}$)**

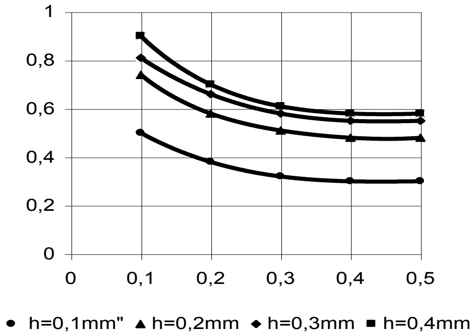


Fig. 10**. μ=f (β**

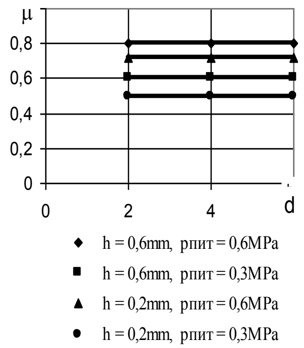


Fig 11. **μ = f (d)**

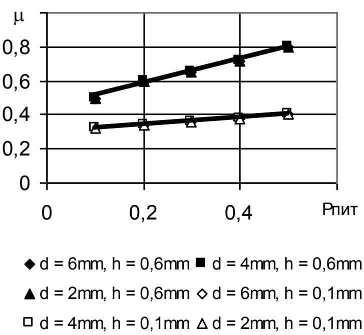


Fig.12. **μ = f (p$_{пит}$)**

In equations of motion includes values rigidity of diaphragms - $\mathbf{c_к}$ and $\mathbf{c_м}$ both seals of the valve and saddle - $\mathbf{c_{кс}}$ both valve and diaphragm - $\mathbf{c_{км}}$. These values in the present activity were determined by an experimental way on known techniques of determination of rigidity of any springs, loading their known efforts and determining in the same time of thickness of seals and course of diaphragms. The experience were conducted some times at load and unloading. The obtained experimental data allow to consider(count) the law of change of rigidity linear (in the given range of variation of parameters), and values **c** - as constants.

Rigidity of seals $\mathbf{c_{кс}}$ и $\mathbf{c_{км}}$ at load and unloading oscillated in limits:

$\mathbf{c_{кс}}$ = 9 … 11 н/mm and $\mathbf{c_{км}}$ = 3 … 5 н/mm.

For calculation were selected the average value of these parameters:

$\mathbf{c_{кс}}$ = 10 н/mm and $\mathbf{c_{км}}$ = 4 н/mm.

To rigidity of equivalent springs $\mathbf{c_к}$ and $\mathbf{c_м}$, , cloning elastic properties of diaphragms made from the same material, were peer each other and their average value

$\mathbf{c_к}$ = $\mathbf{c_м}$ = 4 н/mm.

Equilibrium equation of the valve 5 (fig. 1.) has a kind:

$$p_{\text{ïèò}} \cdot F_{\hat{a}} + M_{kg} + c_k y + M_{mg} + P_{\tilde{a}} + c_m x + p_{\text{àòì}} \left(F_A - F_{\tilde{A}}\right)$$
$$= p_A \cdot F_A + P_{kc} + P_{km},$$

where: $P_{\text{êñ}} = \tilde{n}_{\text{êñ}} \cdot z_{\text{êñ}}$ i $P_{\text{êì}} = \tilde{n}_{\text{êì}} \cdot z_{\text{êì}}$.

From this equation, knowing values of weights of the valve and diaphragm Mк and Mm, effective areas F_A, F_B and $F_Г$, force $P_{кс}$ and $P_{км}$ for definite value of valve lift and diaphragm is possible to determine force Pг, which one is peer:

$$P_{\tilde{a}} = p \cdot F_{\tilde{a}} + G_2 \cdot v .$$

To determine value of hydrodynamic component force Pг it is possible by different ways. For example, knowing value of the flow of drop G_2, temperature T, pressure p and the area $F_г$ is possible to determine flow rate of air in the valve 5 **v** and to count up the hydrodynamic component $P_{гд} = G_2$ v.

However, the same value can be determined on [9] Till these techniques:

$$P_{\tilde{a}\ddot{a}} = \psi \cdot p \cdot F_{\tilde{a}} ,$$

where: **ψ** - coefficient of a hydrodynamic force.

Coefficient **ψ** generally is not a constant and depends on a mode and flow pattern of a liquid, coefficient of local resistances **ξ** geometrical parameters. So for example, in [8] the formula for determination of a coefficient **ψ** is adduced:

$$\psi = 1 + (1 - \xi) \cdot 16 \cdot \mu_{\tilde{n}}^2 \cdot h^2 / d_c^2$$

But under this formula for calculation of a coefficient of a hydrodynamic force it is necessary to determine by an experimental way a coefficient of local resistance **ξ**. Therefore was selected a path of determination of a coefficient of a hydrodynamic force **μ** by a substitution in an equation experimental values of pressure, effective areas and weights at different pressure of a feed.

The experiments have shown, what a coefficient of a hydrodynamic force **ψ** depends on pressure of a feed and its mean values are peer:

For pressure of a feed $\mathbf{p_{пит}}$ = 0,3 Мпа **- ψ** = 1,4,

$p_{пит}$ = 0,6 Мпа **- ψ=** 1,56.

In calculations the values for the valve in a closed position were adopted **ψ** = 1 (valve in a closed position) and **ψ** = 1,48 (the valve in a open position).

Coefficients of viscous friction **α** to were determined experimentally on [10]

Equation of motion of the valve and diaphragm [3] is possible to present as:

$$\frac{dv_{\hat{e}}}{dt} = \frac{1}{M_k}\left[P_{kc} + P_{km} - M_k g - c_k y + P_m F_{\hat{A}} - pF_k\right] - \frac{\alpha_k v_k}{M_k} .$$

$$\frac{dv_m}{dt} = \frac{1}{M_m}\left[p_a F_{\grave{A}} - p_{\text{àòì}} \left(F_{\grave{A}} - F_{\tilde{A}}\right) - P_{\tilde{A}} - P_{\text{êñ}} - P_{\text{êì}} - \tilde{n}_{\grave{i}}\, x - M_{\grave{i}}\, g\right] - \frac{\alpha_{\grave{i}}\, v_{\grave{i}}}{\grave{I}_{\grave{i}}} ,$$

(for constant speed $\mathbf{v_к}$ and $\mathbf{v_м}$ and coefficient of a hydrodynamic force **ψ** = 1 (the valve is gone together with a diaphragm), knowing earlier definite values of rigidity of diaphragms and seals, and also it's effective areas, is possible to determine coefficient of viscous friction $\alpha_г$ and $\boldsymbol{\alpha_м}$.

The experiments have shown, that in the given pressure range at constant rigidites of seals and diaphragms, for the data of geometrical parameters it is possible to consider coefficients of viscous friction as constants and:

$\alpha_г$ = 20Нс/м and $\boldsymbol{\alpha_м}$ = 10Нс/м.

All experimental values will be used in a non-linear mathematical model as design values.

RESEARCHE RESULTS

With the purpose of reduction of number of experiments at preservation of veracity of the obtained data the method of orthogonal planning [11] was applied. Generally it is possible to dedicate following representative problems of planning of experiment:

Determination of an extremum of a function in the field of its research.

Selection of eligible model for the description of object or determination of parameters of a known functional connection.

Determination of adequacy to a mathematical model of composite process, at which one realization of physical experiments is connected to definite handicappings.

At usage of a method of Box - Wilson [12,13] approximating functions are represented as a power series and more often will use a polynomial of the second order:

$$F = C(1)\cdot x^2 + C(2)\cdot x + C(3)\cdot x\cdot y + C(4)\cdot y + C(5)\cdot y^2 + C(6),$$

where: W (1), W (2), W (3), W (4), W (5), W (6) - coefficients of an interpolation polynomial;

x, y - varied factors.

The verifying calculations have shown necessity of usage of a matrix of planning of the second order. The factors are a height of valve lift both nozzles $\mathbf{h}_k$ and pressure ratio on input and output $\boldsymbol{\beta}$.

The designed valves - amplifiers have following ranges of change of parameters subcritical - $\boldsymbol{\beta}$ = 0.1-0.5, $\mathbf{h}$ = 0.1-0.7мм, supercritical mode $\boldsymbol{\beta}$ = 0.6-0.9, $\mathbf{h}$ = 0.1-0.5мм. Therefore for $\boldsymbol{\beta} \le \boldsymbol{\beta}_{кр}$ the index plane was selected -$\mathbf{h}$ = 0.4мм, $\boldsymbol{\beta}$ = 0.3, step of variation on each parameter $\boldsymbol{\Delta\beta}$ = 0.2 and $\boldsymbol{\Delta h}$ = 0.3мм. For $\boldsymbol{\beta} \ge \boldsymbol{\beta}_{кр}$ an index plane h = 0.3мм and $\boldsymbol{\beta}$ = 0.75 with a step $\boldsymbol{\Delta\beta}$ = 0.15 and $\boldsymbol{\Delta h}$ = 0.2мм and for a flapper – nozzle $\mathbf{h}$ = 0.3, $\boldsymbol{\beta}$ = 0.3 and $\boldsymbol{\Delta\beta}$ = 0.2, $\boldsymbol{\Delta h}$ = 0.2мм.

After processing the experimental data obtained by the response function equation in the form of second-degree polynomial has the form:

$$\boldsymbol{\mu} = A + B\mathbf{h} + 3\,\boldsymbol{\beta} + D\mathbf{h}^2 + E\,\boldsymbol{\beta}^2 + F\,\boldsymbol{\beta}\,\mathbf{h}.$$

1. Subcritical mode:

A = 0,56256, B = 3,5915, C = -1,1919, D = -3,1679, F = -1,0257

2. Supercritical mode:

A = 0,33117, B = 2,897, C = -1,8009, D = -2,6393, F = -1,0257

3. Flapper – nozzle:

A = 0,17648, B = 3,9293, C = -1,3826, D = -4,1215, F = -1,3826.

To test the adequacy of the approximation obtained dependences of the coefficients are assessing the significance of the regression equations. Let's consider, that the coefficient is significant, if its absolute value is more than admissible error. Such check of a significance of coefficients of an equation of a response function has not allowed it to simplify. For check of adequacy of the obtained approximating relations the criterion of the Fisher is used. The dissipation of experimental points concerning an equation of a response function is characterized by a residual dispersion or dispersion of adequacy:

$$S^2{}_{\grave{a}\ddot{a}} = \frac{\sum_{1}^{n} \delta_{\ddot{e}}}{f},$$

where: f - a difference between number of experience and number of factors of model, which one are computed by results of these experience:

f = K- (n + 1),

where: (n + 1) - number of factors of model.

At determination of adequacy to approximating model f = 5.

The dispersion of reproducibility $S^2_{\div}$ is determined by a measuring error (systematic and random errors). Estimated value of Fisher's exact test:

$$F_{\eth} = \frac{S^2_{\grave{a}\ddot{a}}}{S^2_{\div}},$$

compared with the table (at a given confidence level and the relevant degrees of freedom) is exceeded the table value of the adequacy of the hypothesis is rejected. In our case, the approximation model Fp = 1.8, and valued F_T = 2.1, which indicates the adequacy of the equation obtained the response function.

On the fig. 13-15 the comparative relations of a discharge coefficient **μ** are shown from value of valve lift and backlash a flapper - nozzle for different parameters of gasdynamic process **β**.

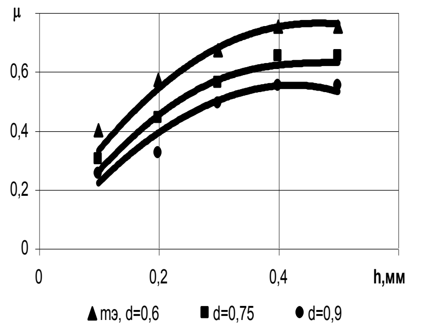


Fig. 13. μ= f(h) при β>$β_{кр.}$

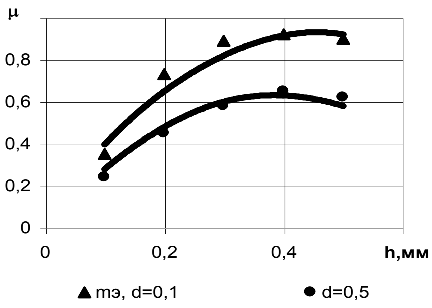


Fig. 14. μ=f (h) при β ≤ $β_{кр.}$

As can be seen from the graphs, the approximation depending accurately describe the processes taking place and these equations can be used in a mathematical model of the pneumatic positioner mechanical systems for the calculation of static and dynamic characteristics and of the numerical experiment.

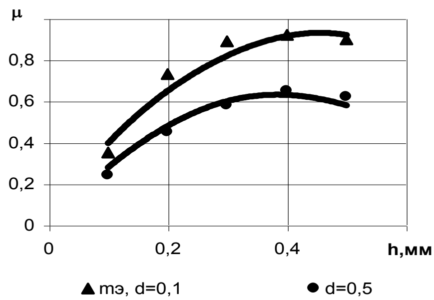


Fig. 15. **µ= f (h)** for flopper – nozzle at $\beta \le \beta_{кр}.d_у$=2мм

CONCLUSIONS

To determine the coefficients in the system of equations of the experimentally determined static characteristics of the positioner. Empirically found dependence of discharge coefficients µ of construction-setting parameters positioners and gasdynamic regimes. Are retrieved of rigidity of diaphragms - $\mathbf{c_к}$ i $\mathbf{c_м}$, seals of the valve and saddle $\mathbf{c_{кс}}$, valve and diaphragms - $\mathbf{c_{км}}$., value hydrodynamic component of forces **Pг** and coefficients of viscous friction $\alpha_г$ and $\boldsymbol{\alpha_м}$.

Obtained an approximation model to determine the discharge coefficient and µ is defined by its adequacy.

REFERENCES

1. Morgan Jones,2004.: Duilding Valve Amplifiers//Ntwnes.-363 p.
2. Osenin Y.. Remen V., 2010: Accuracy increase of positioning of pneumatic drives for mechanical systems. TEKA Commission of Motorization and Power Industry in Agriculture, V XB, p. 95-99.
3. Osenin Y.. Remen V., Epifanova O., 2010.: Mathematical model of valve - amplifiers for pneumatic drives of mechanical systems. TEKA Commission of Motorization and Power Industry in Agriculture, V XC, p. 255-260.
4. Andruxin N.N., Polin L.P., 1971.: To a problem of experimental determination of a discharge coefficient in the distribution device pneumatic drive.- Pneumatic drives and control systems. M.: p.236-239.
5. Govorkov V.C., Chalatov E.M., 1971.: A way of determination of a discharge coefficient.- Pneumatic drives and control systems. M.: p.233-236.
6. Esin V.I., Nugnov V.I., 1971.: Determination of a drag and discharge coefficient in composite pneumatic systems.- Pneumatic drives and control systems. M,: p.230-232.
7. Dmitriev V.N., Gradecky V.G., 1973.: A fundamentals of a pneumoautomatics. M.: Mashinostroenie, - p.360.
8. Bogacheva A.V., 1966.:. Pneumatic devices of automatic control systems. M.: Mashinostroenie.- P.240.

9. Adler Y.P., Markova E.V., Granovsky Y.V., 1971.: Planningof experiment by looking up of optimal conditions. - M.: Nauka. - p. 286.
10. New ideas in planningof experiment 1969.: / Under red. V.V. Nalimov. - M.: Nauka. - p.334.
11. B Bundy. Methods of optimization. 1988.:- M.: Radio i sviaz. - p.128..
12. Zalmanzon. L.A. Running elements of pneumatic devices of control and management. It is M.:?? SSSR.1961.-p.247.
13. Zaporozhets V.P. 1976.:Research of expense descriptions of membrane strengtheners for stream sequencing control system.- Stream technique. Theses of lectures VI Mezhdunarod. Conf. Yblonna. M.. P.91-95.
14. Popov D.N. 1977.: Dynamics and adjusting of hydraulic and pneumatic systems: Uchebnik dlya mashinostr. VUZoz.-M.: Mashinostroenie, 1977. -p.424.
15. Suliga S.V. 1982.:Research of transients in membrane pneumatic amplifier by computer.- Iz.vuzov. Mashinostroenie,, № 6, p.53-58.
16. Popov D.N., Suliga S.V. 1982.: Bases of calculation and choice of optimum diameters of twocascade membrane pneumatic amplifiers.- Vestnik mashinostr., № 4, p.7-15.

СТАТИЧЕСКИЕ ХАРАКТЕРИСТИКИ КЛАПАНОВ - УСИЛИТЕЛЕЙ ПНЕВМАТИЧЕСКИХ ПРИВОДОВ МЕХАНИЧЕСКИХ СИСТЕМ

Ванетин Ремень, Александр Вялых, Михаил Лордкипанидзе

Аннотация. Рассмотрены статические характеристики пневматических клапанов-усилителей приводов механических систем с улучшенными динамическими характеристиками. Опытным путем найдены зависимости коэффициентов расхода $\mathbf{\mu}$ от конструктивных параметров позиционеров и газодинамических режимов, твердости мембран - $\mathbf{c_к}$ і $\mathbf{c_м}$, уплотнений клапана и седла- $\mathbf{c_{кс}}$, клапана и мембрані - $\mathbf{c_{км}}$., величину гидродинамической составляющей силі $\mathbf{P_г}$ и коєффициенты вязкого трения $\mathbf{\alpha_к}$ і $\mathbf{\alpha_м}$. Получена аппроксимационная модель для определения коэффициента расхода μ и определена ее адекватность.

Ключевые слова: клапан-усилитель, пневматический привод, коэффициенты расхода.