SCIENTIFIC PAPERS
OF THE UNIVERSITY OF PARDUBICE
Series B
The Jan Perner Transport Faculty
13 (2007)

SERVOMECHANISM FOR CONTROLLING GEOMETRIC VOLUME OF PUMPS

Gabriela KOREISOVÁ, Josef KOREIS

Department of Transport Means and Diagnostics

1. Introduction

For controlling geometric volume of variable pumps, external circuits of automatic regulation are used as ordered by control targets set. External circuits of automatic regulation comprise control variable β_{11} brought to the input of the position servomechanism. The position servomechanism is a part of the controlled pump construction. A dimensionless geometric volume β_1 of the controlled pump is an output variable of the servomechanism. The article describes static and dynamic properties of typical construction arrangement of controlling servomechanisms of variable pumps.

2. Controlling Servomechanism Arrangement

Constructional cut by variable pump is on Fig. 1. Working pistons, (in the upperpart of the cut), bears on a swing board, (left), held in the mean neutral position zero by drawback springs bias. (As seen in the lower part of the cut). P-P perspective as marked on Fig. 1 is depicted on Fig. 2 on the left. A control lever is bolted to the cylindrical segment of the swing board and attached with a stopple at its end with a sliding stone. The sliding stone is inserted into an alveolus in the mean position of the cylinders piston-rod. On the left, drawback springs are seen bearing on the cylindrical segment head of the swing board across the angle plate. There are two drawback springs keeping the swing board in the mean neutral position with a prescribed bias.

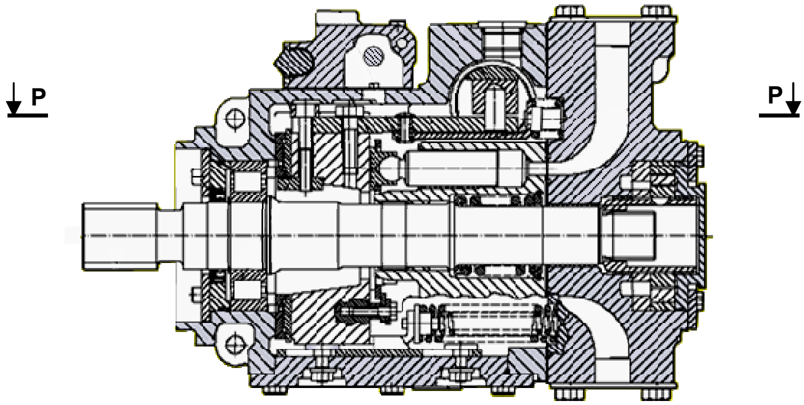


Fig. 1 *Constructional cut by pump*

(Upon a loss of the controlling pressure or with geometric volume control switched off). Drawback springs bias results in insensitivity around zero of the controlling signal important for safety of the operation. All the variable pumps for the closed hydraulic circuit (for construction of HS transfers of mobile machines) are equipped with drawback springs with bias.

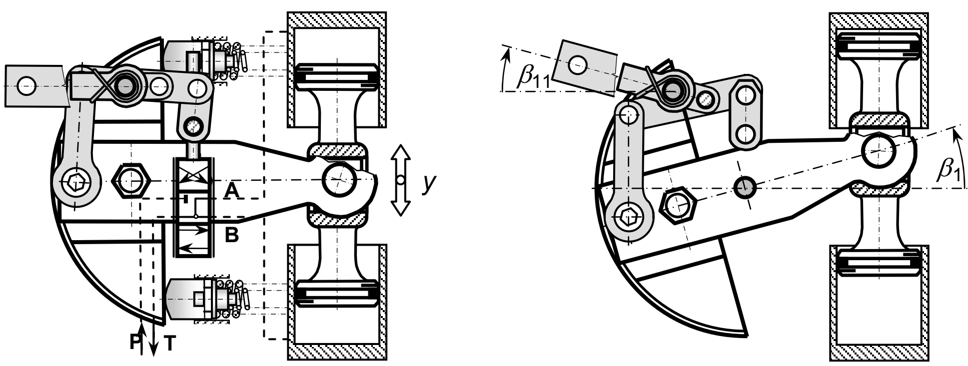


Fig. 2 *Servomechanism in fundamentals and working position*

Lift of the servo-cylinder *y* with the aid of the control lever is transformed into angular deflection of the swing board α_1. Connected to the control lever is a feedback mechanism allowing input deflection of control valve of servo-valve and its backward return towards the neutral position, proportional deflection of the swing board. Fig. 2 presents the feedback mechanism with manual mechanical control. The servomechanism in a tilted position is depicted on Fig. 2 on the right. Let the servomechanism be located in the fundamental neutral position, as indicated on Fig. 2 on the left.

By deflecting the manual control lever by angle α the control valve of the servo-valve is shifted by lift x_α into an open position. By opening the control valve of the servo-valve, pressure difference Δp_R is transferred to the heads of servo-cylinder pistons and the servo-cylinder starts moving in the desired direction by lift *y.*

The motion of the servo-cylinder makes the swing board turn in the direction of angle α_1. The turn of the swing board transfers the feedback mechanism to the retrograde motion of the control valve that moves backwards by lift $\Delta x_{\alpha 1}$, towards the neutral position. The entire servomechanism stops in a new balanced position where:

Control valve ajar $\quad x_R = x_\alpha - \Delta x_{\alpha 1}$. (1)

Pressure difference at servo-cylinder $\quad \Delta p_R = K_S \cdot x_R$.

Equilibrium of forces $\quad S.\Delta p_R = k_Y \cdot (y_0 + \Delta y_R)$ (2)

Dimensionless geometric volume $\quad \beta_1 = \frac{V_g(\alpha_1)}{V_{g\max}} = \frac{\alpha_1}{\alpha_{1\max}} = \frac{y}{y_{\max}}$. (3)

Dimensionless input control variable $\quad \beta_{11} = \frac{\alpha}{\alpha_{\max}}$. (4)

In the latter equations K_S is a control valve constant and k_Y is drawback springs constant. In the deflected position, force from pressing just one string counteracts the servo-cylinder hydraulic force. In the new steady state, geometric volume is proportional to the input control variable. ($\beta_1 = \beta_{11}$). Within the course of the transient process, the response $\beta_1(t)$ is delayed in time behind the input change $\beta_{11}(t)$.

3. Types of Steering Used and Static Features of Control

In the technical documentation, types of positional servomechanism steering are derived from the type of energy source of the input control variable. There are three basic types of positional servomechanism steering: „mechanical, hydraulic and electric“.

Mechanical steering has been partially described. The hydraulic scheme and control valve arrangement with mechanical steering is on Fig. 3.

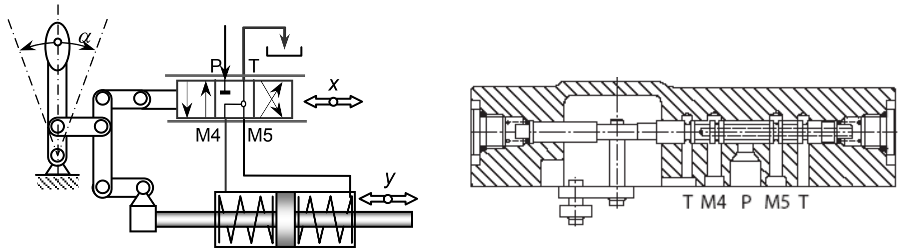


Fig. 3 *Mechanical steering*

For describing static and dynamic properties, dimensionless variables will be used marked with a stripe above the variable symbol. Equation (3) and (4) are transferred into the shape of.

Input control variable $$\beta_{11} = \frac{\alpha}{\alpha_{max}} = \overline{\alpha} = \frac{x}{x_{max}} = \overline{x} \tag{5}$$

Output controlled variable $$\beta_{1} = \frac{V_g(\alpha_1)}{V_{g\,max}} = \frac{y}{y_{max}} = \overline{y} \tag{6}$$

The hydraulic steering arrangement is shown on the Fig. 4.

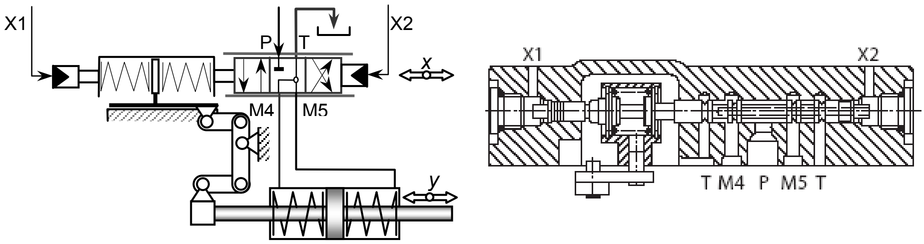


Fig. 4 *Hydraulic steering*

On Fig.4 on the left, there is a control valve cut from catalogue documentation. Symbols X1 and X2 are marked as connecting channels for inducing a controlled pressure difference $\Delta p_{12} = \left| p_{X1} - p_{X2} \right|$. On Fig. 4 on the left, there is a hydraulic wiring diagram and schematic design of the feedback mechanism.

Input control variable $$\beta_{11} = \frac{\Delta p_{12}}{\Delta p_{12\,max}} = \Delta\overline{p}_{12}\,. \tag{7}$$

Output controlled variable $$\beta_{1} = \frac{V_g(\Delta p)}{V_{g\,max}} = \frac{y}{y_{max}} = \overline{y}\,. \tag{8}$$

Possible layout of direct electromagnetic steering is on Fig. 5.

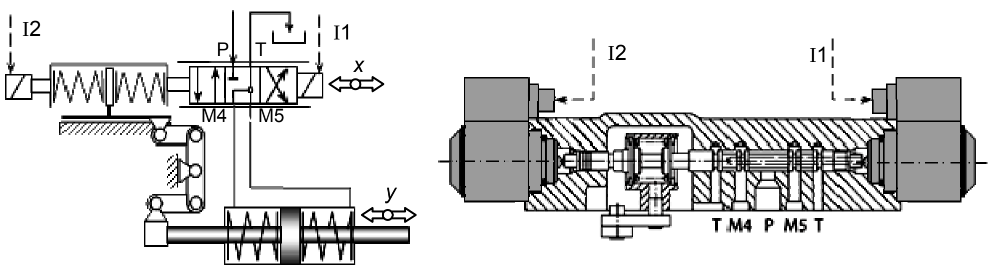


Fig. 5 *The direct electromagnetic steering*

Applied to the control valve head is the force of proportional electromagnets. Unless the spring constant in the feedback casing is changed, each proportional electromagnet must produce a force equal to the one produced by the hydraulic pressure p_X on the front surface of the sliding valve S_X as shown on the preceding figure. Proportional electromagnets are then big and the weight of their core is added to the weight of the sliding valve.

Larger inertial masses cause a minor own frequencies of the servo-valve. The force produced by proportional electromagnet is proportional to the flow I1, or I2, of the input control electric signal. The motion of control valve of the servo-valve is always provided by one electromagnet at a time, counteracting the force of one spring in the feedback casing.

Input control variable $$\beta_{11} = \frac{I}{I_{\max}} = \bar{I} \,. \tag{9}$$

Output controlled variable $$\beta_1 = \frac{V_g(I)}{V_{g\max}} = \frac{y}{y_{\max}} = \bar{y} \tag{10}$$

Often used is two-stage electro-hydraulic steering according to Fig. 6 having more suitable dynamic qualities than the direct control by proportional electromagnets.

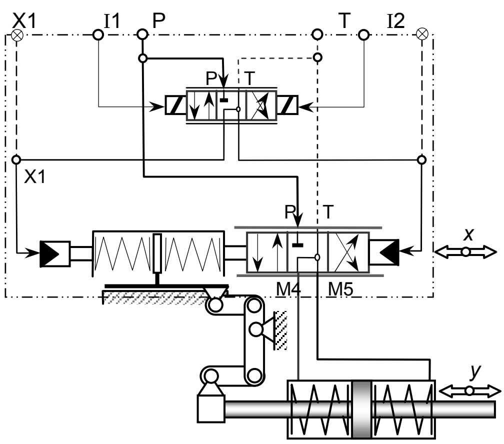


Fig. 6 *The two-stage electro-hydraulic steering*

The first control stage consists of a small electro-magnetically controlled servo-valve, labelled as PCP. (Pressure Control Pilot). The input control variable is flow I1, or I2, brought from a superior electronic control unit to the input of the first stage electromagnets.

The output variable of the first control stage is controlled pressure difference

$$\Delta p_{12} = |p_{X1} - p_{X2}|$$

brought to the inputs X1 and X2 of the second control stage.

The second control stage is presented by hydraulic servo-valve with flexible feedback, being of the same version as the hydraulic positional servomechanism on Fig. 4.

Hydraulic servo-valve is a basic control feature mounted to the pump casing. Proportional electromagnets on Fig. 5, or electromagnetic first control stage according to Fig. 6, consist of an attachment mounted on connecting areas of hydraulic servo-valve.

With a two-stage electro-hydraulic steering according to Fig. 6, hydraulic inputs X1 and X2 are plugged, yet they remain available for emergency (proportional), hydraulic control of geometric volume HGR. Similarly proportional electromagnets for direct control are equipped with an emergency mechanic steering button.

Geometric volume HGR is an output control variable of all the mentioned types of positional servomechanism, being represented by dimensionless controlled variable $\beta_1 = y / y_{\max}$.

The input control β_{11} is a dimensionless deviation of control lever of mechanic steering, or a dimensionless pressure difference of hydraulic steering, or dimensionless flow of electric steering.

All the types of positional servomechanism for controlling geometric volume HGR have the same course of dimensionless static characteristics of the control as outlined on Fig. 7.

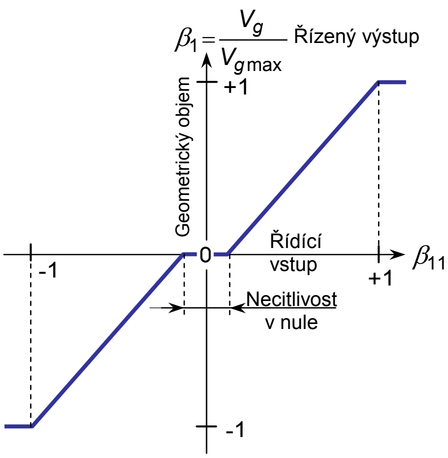


Fig. 7 *The control characteristic.*

Maximal value of the controlled geometric volume is limited by a solid mechanic stop. The course of static characteristics of geometric volume control has a shape of a ramp function with insensitivity around the beginning, intentionally introduced by servo-cylinder springs bias. The regulatory pump has 100% reversal capability.

Geometric volume varies from negative maximal value through neutral to maximal positive value ($\beta_1 \in \langle -1, 0, +1 \rangle$).

A static characteristic of geometric volume control is an odd function being symmetrical by the beginning.

With all the types of positional servomechanisms for controlling geometric volume, there is a dimensionless output variable $\beta_1 = \bar{y}$, where $\bar{y}$ is a dimensionless lift of the servo-cylinder.

4. Dynamic Characteristics of a Controlled Pump Servomechanism

For describing dynamic characteristics of the servomechanism, there is a hydraulic servomechanism shown repeatedly on Fig. 8 being supplemented by labelling of variables and parameters.

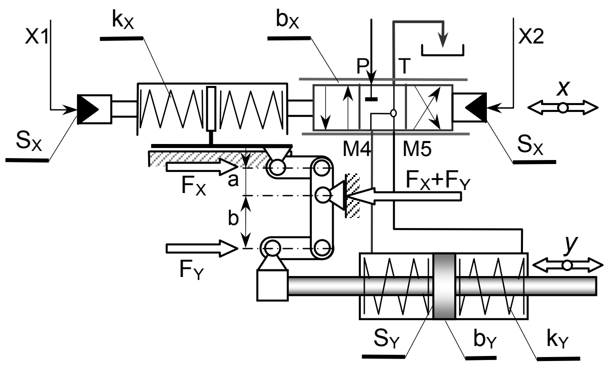


Fig. 8 *Parameters of Servomechanism with Hydraulic Steering*

On inputs X1 and X2 there is an input pressure difference Δp_{12} impacting. Output variable of servo-valve is a controlled pressure difference Δp_{45}, being proportional to the resultant lift of the sliding valve $x - i_Z \cdot y$ The lift of the servo-cylinder y is brought by a compound lever to the spring in the servo-valve casing with feedback transfer ratio $i_Z = a/b$. Variables related to the servo-valve are labelled with X index. Variables related to the servo-cylinder have Y index.

The initial steady state is disturbed by step change in an input control pressure difference $\Delta p_{12}(t) = H(t).\Delta p_{12K}$, which is expected to keep its new constant value $\Delta p_{12}(0) + \Delta p_{12K}$ for the whole time of the monitored transient process. On the basis of equilibrium of forces and lifts on Fig. 8 we may derive image transfers for dimensionless deviations of variables:

For servo-valve: $$A_X(s) = \frac{\overline{x}(s)}{\Delta \overline{p}_{12}(s)} = \frac{1}{T_{2X}^2 \cdot s^2 + T_{1X} \cdot s + 1} \qquad (11)$$

For servo-cylinder: $$A_Y(s) = \frac{\overline{y}(s)}{\Delta \overline{p}_{45}(s)} = \frac{1}{T_{2Y}^2 \cdot s^2 + T_{1Y} \cdot s + 1} \qquad (12)$$

The dependence of the output pressure difference of the servo-valve $\Delta \overline{p}_{45}$ on lift $\overline{x}$ is a non-linear pressure characteristic of the servo-valve, being linearized by a straight line crossing the centre of the symmetry according to Fig. 9. (On Fig. 9, a half of the characteristic is for positive values of the lift $\overline{x}$ only).

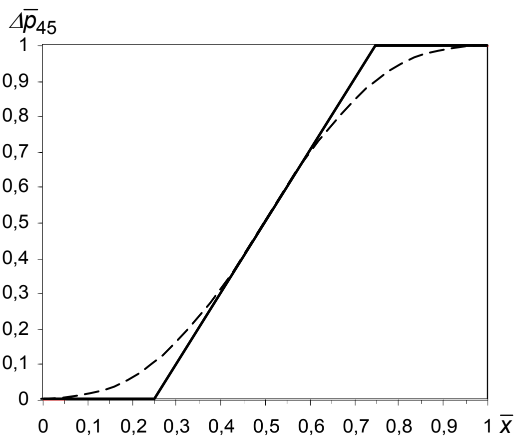


Fig. 9 *The pressure characteristic of servo-valve*

The dashed course measured $\Delta \overline{p}_{45} = f(\overline{x})$ is replaced by linear course depicted by a firm line. Linear replacement is valid for interval $\overline{x} \in \langle 0{,}25; 0{,}75 \rangle$. The linear replacement directive is constant K_S, being of value

$$\overline{K}_S = \frac{\Delta \overline{p}_{45}}{\Delta \overline{x}} = \frac{1}{0{,}75 - 0{,}25} = \frac{1}{0{,}5} = 2$$

The characteristic on Fig. 9 is dimensionless. Constant $\overline{K}_S = 2$ is also dimensionless being valid only for dimensionless variables.

Pressure difference on the output of the servo-valve without feedback is

$$\Delta \overline{p}_{45} = \overline{K}_S \cdot \overline{x} = 2 \cdot \overline{x} \qquad (13)$$

Pressure difference on the output of the servo-valve with feedback and with transfer ratio of the feedback mechanism $i_Z = 0{,}5$ is:

$$\Delta\bar{p}_{45} = \bar{K}_S \cdot (\bar{x} - i_Z \cdot \bar{y}) = 2 \cdot \bar{x} - \bar{y} \tag{14}$$

The block diagram of a servomechanism with hydraulic servo-valve is on Fig. 10.

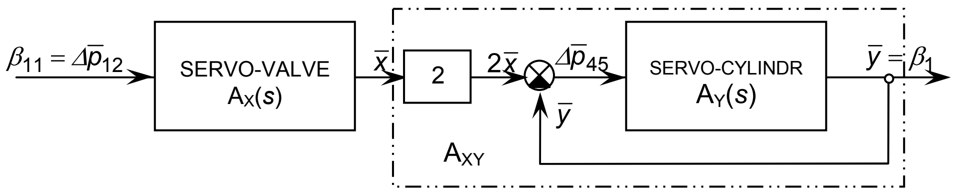


Fig. 10 *The configuration of servomechanism with hydraulic servo-valve*

The image transfer $A_Y(s)$ is valid for the servo-cylinder itself without the impact of feedback. According to Fig.10 is $\bar{y} = A_Y(s) \cdot \Delta\bar{p}_{45} = A_Y(s) \cdot (2\bar{x} - \bar{y})$, where $\bar{y} \cdot (1 + A_Y(s)) = 2 \cdot A_Y(s) \cdot \bar{x}$.

The resultant transfer of servo-cylinder with feedback is:

$$A_{XY}(s) = \frac{\bar{y}(s)}{\bar{x}(s)} = \frac{2 \cdot A_Y(s)}{1 + A_Y(s)} \tag{15}$$

By substituting AY(s) into (15) according to equation (12) we obtain the resultant transfer of servo-cylinder with feedback as:

$$A_{XY}(s) = \frac{\bar{y}(s)}{\bar{x}(s)} = \frac{2 \cdot A_Y(s)}{1 + A_Y(s)} = \frac{2}{T_{2Y}^2 \cdot s^1 + T_{1Y} \cdot s + 2} = \frac{1}{T_{2XY}^2 \cdot s^2 + T_{1XY} \cdot s + 1} \tag{16}$$

The resultant transfer of servo-cylinder with feedback (16) is at the same time an image transfer of the control servomechanism with direct mechanic steering of the lift of slide valve $\bar{x}$ (without the servo-valve). The feedback halves the numerical values of time constants. The following is valid for time constants in the equation (16):

$$T_{2XY}^2 = 0{,}5 \cdot T_{2Y}^2, \quad T_{1XY} = 0{,}5 \cdot T_{1Y}$$

The resultant image transfer of the entire servomechanism is in the form of:

$$A_\beta(s) = \frac{\beta_1(s)}{\beta_{11}(s)} = A_X(s) \cdot A_{XY}(s) = A_X(s) \cdot \frac{2 \cdot A_Y(s)}{1 + A_Y(s)} \tag{17}$$

where $A_X(s)$ is an image transfer of the entire servo-valve according to (11)

5. Choosing Numerical Values of Dynamic Parameters

Proportional image transfer of the 2nd order has three equivalent shapes:

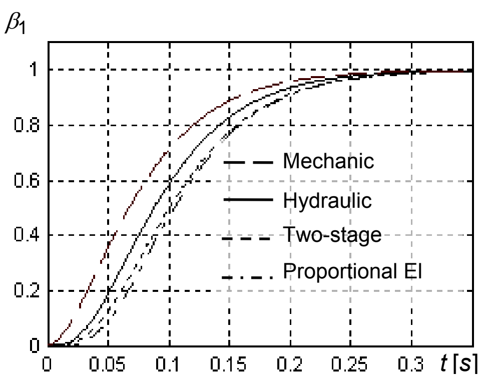
$$A_2(s) = \frac{\bar{r}_0}{T_2^2 s^2 + T_1 s + 1} = \frac{\bar{r}_0}{T_2^2 s^2 + 2\delta T_2 s + 1} = \frac{\bar{r}_0 . \omega_0^2}{s^2 + 2\delta\omega_0 s + \omega_0^2} \tag{18}$$

To determine time constants, numerical values of natural frequencies were chosen.

For the servo-cylinder $\omega_{0Y} = 25\ s^{-1}$, for hydraulic servo-valve $\omega_{0H} = 100\ s^{-1}$, for electromagnetic servo-valve $\omega_{0E1} = 62{,}5\ s^{-1}$.

For a small electromagnetic servo-valve PCP of electro-hydraulic two-stage steering $\omega_{0E2} = 125\ s^{-1}$. For all the servo-valves, the same numerical value of the dumping factor on the limit of aperiodicity $\delta = 1$. Subsequently, servo-cylinder's time constants values were calculated as well as were those of individual servo-valves by relations: $T_2 = 1/\omega_0$, $T_1 = 2\delta/\omega_0$.

The course of transient responses of geometric volume control by servomechanism with various types of servo-valve is on Fig. 11.



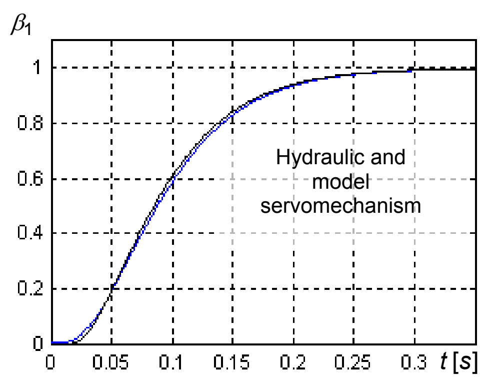


Fig. 11 *The characteristic of the time response of servomechanisms*

6. Conclusion

When analysing dynamic properties of HGR, it is beneficial to apply one model servomechanism. For a model servomechanism, 2nd order proportional image transfer with transportation lag was determined by parametric analysis in the shape of:

$$A_\beta(s) = A_{52}(s) = A_{54}(s) = \frac{\beta_1(s)}{\beta_{11}(s)} = Delay(0{,}018).\frac{1}{0{,}0016.s^2 + 0{,}08.s + 1} \qquad (19)$$

The course of the model characteristic is on Figure 11 on the left. By the use of a model servomechanism, the composition of simulation model gets markedly reduced for modelling complex systems incorporating a controlled pump. In the simulation model there will be, instead of two transfer elements and feedback according to Fig. 10, only one block „Transfer Fcn“ with a model image transfer (19). Using a single model servomechanism, it is possible to compare influence of different types of external control circuits.

Lectured by: prof.Ing. Jozef Turza, CSc.

Submited: 23.2.2008

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Resumé

SERVOMOCHANISMY PRO ŘÍZENÍ GEOMETRICKÉHO OBJEMU HYDROGENERÁTORŮ

Gabriela KOREISOVÁ, Josef KOREIS

V článku jsou popsány statické a dynamické vlastnosti typických konstrukčních uspořádání ovládacích servomechanismů regulačních hydrogenerátorů. Pro řízení geometrického objemu regulačních hydrogenerátorů se používají vnější obvody automatické regulace, uspořádané podle stanovených cílů řízení. Vnější obvody automatické regulace tvoří řídící veličinu β_{11}, která je přivedena na vstup polohového servomechanismu. Výstupní veličinou servomechanismu je bezrozměrný geometrický objem β_1 řízeného hydrogenerátoru.

Summary

SERVOMECHANISMS FOR CONTROLLING GEOMETRIC VOLUME OF VARIABLE PUMPS

Gabriela KOREISOVÁ, Josef KOREIS

In the article, static and dynamic properties are described of typical constructional ordering of steering servomechanisms for regulative pumps. To control geometric volume of pumps, outer circuits for automatic recall regulation are used, sequenced according to the control objectives given. Outer circuits of automatic regulation form control variable $\beta_{11,}$ brought into the input of positional servomechanism. Positional servomechanism is a part of the construction of controlled pump. Output variable of servomechanism *is* a dimensionless geometric volume β_1 of the controlled pump.

Zusammenfassung

SERVOMECHANISMISMUS ZU DEN KONTROLLEN DER HUBVOLUMEN DER VARIABLEN PUMPEN

Gabrila KOREISOVÁ, Josef KOREIS

Im Artikel sind beschriebene statische und dynamische Eigenschaften für die typische bauliche Einrichtung der Servomechanismus für Variable hydrogenerators. Zum Steuerung von Hubvolumen der Variable hydrogenerators, um äußere Kreislauf für automatische Rückrufregelung, aufbauest entsprechend der gegebenen Kontrollenzielsetzungen zu benutzen. Der Äußere Regelungkreislauf bildet Führungsgröße β_{11}, das wird in Eingang der Positionsservomechanismus geholt. Ausgang zur Steuergröße der servomechnismus β_1 ist dimensionlose Hubvolumen des Variables hydrogenerators.