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TWO VALVES CONTROL OF HYDRAULIC DRIVE BY WAY OF INVERTED FLOW CHARACTERISTIC

DVOU VENTIILOVÉ ŘÍZENÍ HYDRAULICKÉHO POHONU POMOCÍ INVERZNÍCH PRŮTOKOVÝCH CHARAKTERISTIK

Abstract

Paper deals with control of hydraulic drive by the help of inverted flow characteristics of hydraulic valves. One edge valves or two edges valves, theirs combinations and 3/3-way valves are able to control each chamber of cylinder separately as independent control action members. The case of two three-position three way valves can realize the same functions as a classical three position four-way valve to control hydraulic cylinder, but with an additional control degree of freedom, which could be in several cases profitable. Especially when user can use required criteria in form of required pressures ratio or flows ratio inside of cylinder chambers. In paper the control concept of inverted three-dimensional valves characteristic and inversed dynamic valve behaviour are described.

Intended control concept allows transferring classical hydraulic circuit course and its unsymmetrical static characteristics of circuit realization with single piston rod of cylinder to a course of hydraulic circuit with double piston rod of cylinder.

Abstrakt

Příspěvek se zabývá řízením hydraulického pohonu pomocí inverzních průtokových charakteristik hydraulických ventilů. Ventily s jednou nebo se dvěma škrtícími hranami či jejich vzájemné kombinace a dále pak třícestné, třístavové 3/3 ventily jsou schopny řídit každou z komor hydromotoru odděleně a zvlášť a to pomocí dvou či čtyř nezávislých akčních členů. Vybraný případ dvou 3/3 ventilů muže zajišťovat stejnou funkci jako standardně pro hydraulické pohony užívaný čtyřcestný, třístavový 4/3ventil, ale navíc s novým řídicím stupněm volnosti v podobě další řídicí akční veličiny, což může být v řadě případů výhodné. Obzvláště v případech, kdy může uživatel požadovat jednoduchá řídicí kriteria ve formě tlakových či průtokových poměrů uvnitř komor hydromotoru, což muže pozitivně ovlivňovat jeho chování, které je do značné míry limitováno a ovlivňováno v důsledku užívání jednostranných pístnic. V příspěvku je popsána řídicí koncepce vycházející z invertovaných tří dimenzionálních průtokových charakteristik ventilů a dále z inverzního chování dynamiky ventilů.

Zamýšlená koncepce dovoluje převedení chování klasicky uspořádaného hydraulického obvodu s jednostrannou pístnicí a jeho nesymetrického chování v důsledku nesymetrických statických průtokových charakteristik na chování hydraulického obvodu s pístnicí oboustrannou se statickými průtokovými charakteristikami symetrickými.

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1 INTRODUCTION

Electro-hydraulic systems are usually composed by pressure source, four way flow servovalve and due to space reason usually by hydraulic cylinder with single piston rod. Applications of single piston rod of cylinder lead to unexpected problems in forms of unfulfilled control requirements. One example of that is a task to keep piston position of cylinder in case of vertical arrangement of hydraulic cylinder. In this case and in case of high load on piston a classical four way valve without system adjustment can't be effectively used, which may result in additional cost to system source or to bigger cylinder or to control parts of hydraulic circuit. In addition some undesirable effects as cavitations, low cylinder damping or not accurate achievement of piston position could happen. For example cavitations effects can easily arise in cases, when high speed of lowering of piston is required.

2 CONTROL CONCEPT AND CONTROL ACTION VALUES ASSESMENT

To prevent cavitation effect in case of fast piston ejection or additional installation cost due to maximizing of load force to system source, bigger cylinder or asymmetric control valves one suitable control concept of two independent controlled valves is presented in Figure 1 and more described in [3].

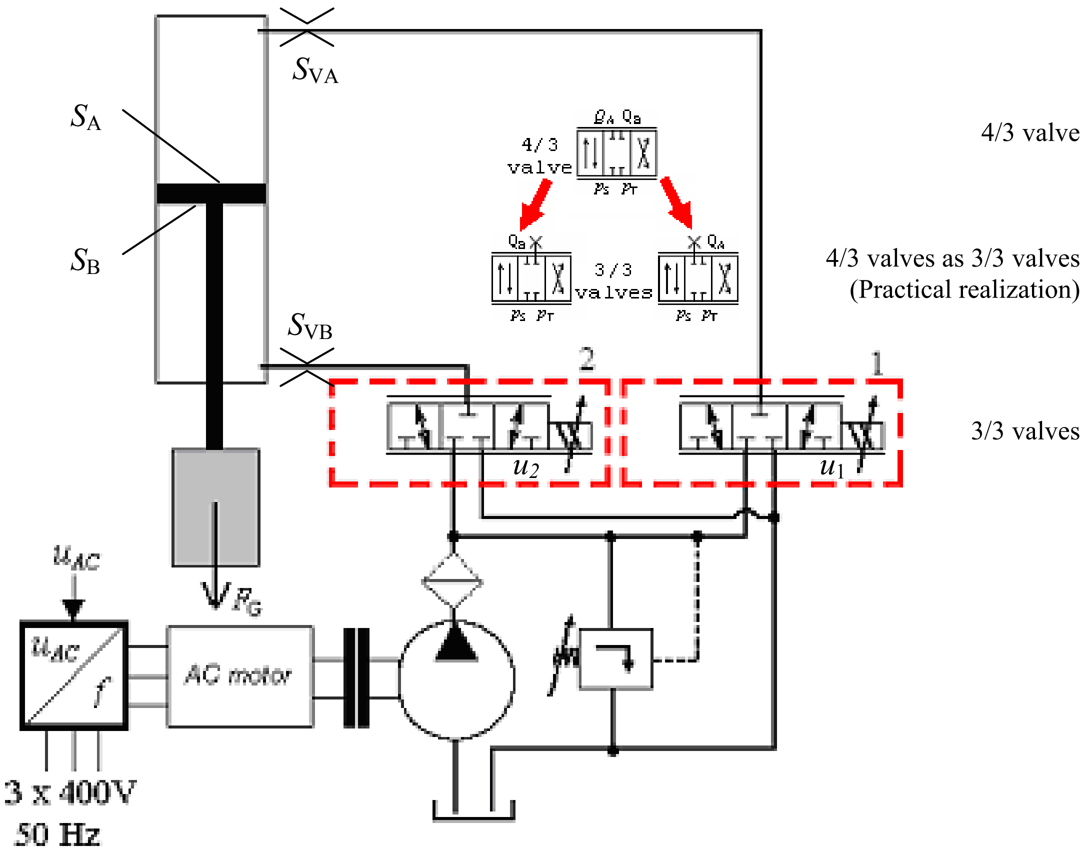

Fig. 1 Control concept for vertical arrangement of cylinder, loaded by high tension force

Advantage of the two valve hydraulic drive arrangement shown in Figure 1 comes from equation (1) for maximum tension load force on piston which is introduced in [4].

$$F_{\max} = \frac{\beta^2}{\alpha^2} \cdot S_B \cdot p_S \text{, where } p_S = \text{system pressure.} \tag{1}$$

$$\alpha = \frac{S_A}{S_B} = \text{Active areas ratio on piston.} \tag{2}$$

$$\beta = \frac{S_{VA}}{S_{VB}} = \text{Flow areas ratio into the chambers of cylinder.} \tag{3}$$

From the equation (1) it is obvious that for the standard arrangements it is possible to reach higher tension force by increasing of system pressure, which will increase energetic demand, or by enlarging of active areas on piston that will lead to bigger cylinder, or by using control valve with asymmetric flow areas, where $\alpha \to \beta$, or in this case by control hydraulic circuit with two separate valves. These can realize full operability of cylinder in all four possible operation quadrants of cylinder, see Figure 1, not only in worst case of vertical arrangement.

For two valves control concepts and from equation of continuity for each side of chambers of cylinder with single piston rod following equation should be valid.

$$\rho_{oil}.S_A.v_{HM} = \rho_{oil}.S_B.v_{HM} = konst, \tag{4}$$

where $S_A = S_B$ for double piston rod of cylinder.

It is obvious that constant values of oil density ρ_{oil}, the same speed of piston of cylinder v_{HM} and for different action areas of piston S_A and S_B will produce asymmetry, so the equation of continuity will not be valid absolutely. This is due to influence of pressure on behaviour of hydraulic drive, which is together with a flow to each chambers of cylinder and coefficient of ratio of action area on piston α the cause of different speeds of piston ejection and insertion in ratio $\sqrt{\alpha}$.

From the previous assumption the first simple idea of control and calculations of action control values u_1, u_2 on two valves comes, which is trying to respect different speeds by way of implementing speed ratio coefficient of piston $\sqrt{\alpha}$. These action values are represented by equations (5) and (6) and theirs base come from equation (9) and classical symmetric valve usage.

$$u_1 = x_{SV1Com} \tag{5}$$

$$u_2 = -\frac{1}{\sqrt{\alpha}} \cdot x_{SV1Com} \tag{6}$$

In this case we have to assume behavior of valves as sufficiently fast and "proportional" and that to an each action value u have to correspond an adequate spool position x_{SV} proportionally.

The same computation of action control values of two control valves can be obtained based on considerations of flow characteristics that are describing valves course. Figure 2 shows this arrangement, where the upper part and lower part of the figure is realizing so called hydraulic halfbridge. If we know these courses of each valve, than we are able to make inversion of that course and previous direct calculations in equation (5) and (6) replaces calculation in Figure 2.

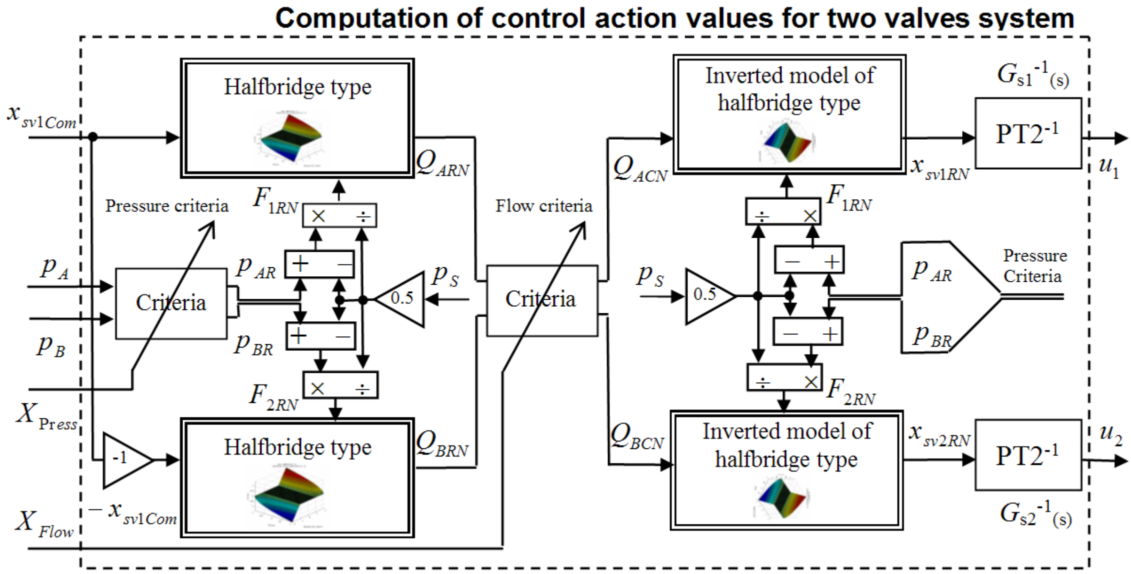

Fig.2 Calculation of action control values for hydraulic drive - control concept with the possibility of criteria control based on changing active areas ratios (flow criteria) and based on changing pressure ratios (pressure criteria)

To use the concept from Figure 2 the same effect like direct calculations can be theoretically achieved together with using calculation of action control values in equation (5) and (6) – equation (6) is inside of block Flow criteria, see further in text. The word "theoretically" is used due to a reason that the knowledge of flow characteristics and their models and inverse models is not only given by theoretical knowledge, but mainly by the knowledge from real measurements of valves courses and their flow characteristics. Each of flow characteristics is related to a relevant throttling edge and to an amount of nonlinear behaviors and hydrodynamic effects that are arising during valves real operation. Inverted characteristics can be than obtained from flow characteristics, stored for example in symmetric 3D matrixes by easy inversion.

3 MEASURED DATA

To verify the control concept several measurements on two real proportional valves PRL2-06-32-0-24 of ARGO-HYTOS Company were made. From these measurements approximations of flow characteristics for two action throttling edges were performed. Measurement was also done for both directions of opening and closing valves due to requirements to take into account the hysteresis on valves.

During measurement was supply pressure kept on 63Bar and maximal supply flow was about 30 l/min. Mentioned measurement of flow characteristics was made only in area of valve usage, when a piston is loaded only by a pressure load force, but in general the loading can be also tension. Measurement for these areas of loads should have to be constituted from a more difficult arrangement of hydraulic measurement circuit.

For our requirements and full verification of control concept it is necessarily and also possible to obtain areas of flow characteristics for tension loads. This we could manage by the help of interpolation and extrapolation methods together with combination of theoretical knowledge about flow characteristic courses, published, for example, in basic forms in [2] and also [1].

These results combine theoretical and measured data can be seen in Figure 3.and it is showing full flow characteristics on valves and comparison of measured and theoretically calculated flow characteristic courses for areas of loading of cylinder piston by pressure and tension load.

Due to very small hysteresis is possible to consider for a control concept only flow characteristics for edges PA and PB and used them also for control of other edges AT and BT, where A is chamber on side A of cylinder, B on other side of cylinder and T means line to a Tank. From that reason and from reason of only two full flow characteristic for all areas of loads are presented in figure 3. Remark: For system with higher system pressures, bigger flows and loads is necessary to measured and approximate all characteristics for each throttling edge separately.

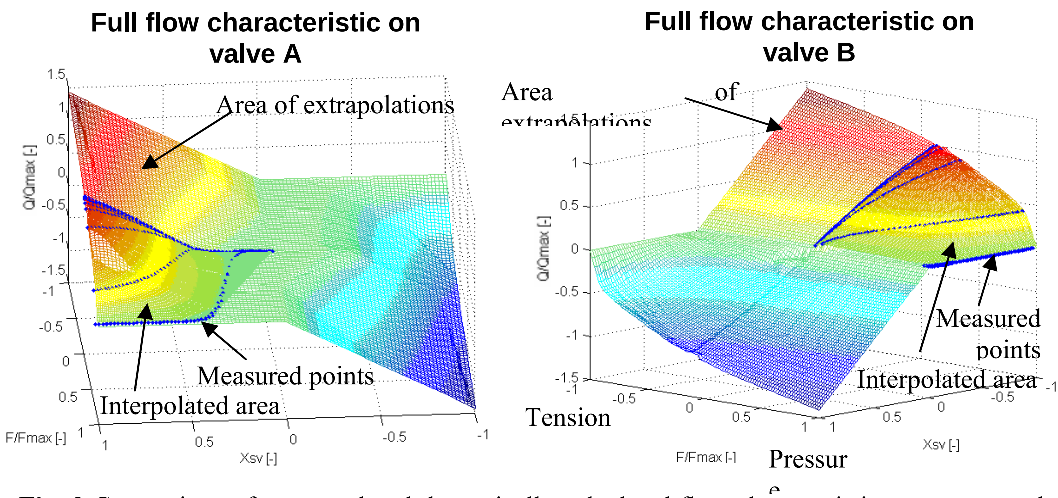

Fig. 3 Comparison of measured and theoretically calculated flow characteristic courses on valves A and B for areas of pressure and tension loads

4 CONTROL BY THE HELP OF PRESSURE AND FLOW CRITERIA

Control concepts presented in Chapter 2 provide, in ideal case, the possibility of control of hydraulic drive to a quality control, which is defined by a level of knowledge of action members, consequently by flow characteristics. Generally, we could say, if we have 100 percent knowledge about behavior of control valves and their correct inverse models it is possible to achieve good control quality of hydraulic drive not only in a close loop, but also in the open loop circuit. Figure 2 shows control concept with the possibility of criteria control with extension about changing active areas ratios (flow criteria) and with extension to changing pressure ratios (pressure criteria) in cylinder chambers.

If we consider influence of coefficient ß in hydraulic circuit we can find equations of steady state variables for piston ejection and insertion as are steady state speed of ejection v_{vys}, steady state speed of insertion v_{zas} and than equation of speed ratio v_{RATIO} between ejection speed and insertion speed, equation (7). Equation (7) can be than easy used as a base of flow control criteria of hydraulic circuit with two valve concept. Same possibilities we could obtain from steady equations of p_A pressure in chamber A and from steady state equation of p_B pressure in chamber B, which could create a base for pressure criteria control than.

$$v_{RATIO} = \frac{v_{vys}}{v_{zas}} = \sqrt{\frac{\alpha - \frac{F}{F_0}}{\beta^4 \cdot \left(1 + \frac{F}{F_0}\right)}}\text{, or} \qquad v_{RATIO} = \frac{v_{vys}}{v_{zas}} = \sqrt{\frac{\alpha}{\beta^4}} \text{ for } F = 0\text{,} \tag{7}$$

and where $F_0 = p_S \cdot S_B$ and F/F_0 can be obtained as $F/F_0 = (\alpha \cdot p_A - p_B)/p_S$. (8)

5 SUMMARY

In this paper a new control concept of a hydraulic drive with two separate control valves and their action values is introduced. Next, it is briefly focusing and describing possibilities of computation of action control values for two valves control together with final introduction of criteria control concept, where flow criteria and pressure criteria can be easily applied and that can act on single piston rod of cylinder behavior. In paper is introduced control concept with two 3/3 valves which can realize same functionality as classical 4/3 valve concept, but additionally it can allows to use single piston rod of cylinder in general arrangement without any restriction by help of suitable separate control of each chamber of cylinder under the flow and pressure criteria. Also an approximation of measured flow characteristics together with comparison with theoretical flow characteristics in pressure and tension areas of load of proportional valves PRL2-06-32-0-24 were presented.

Nomenclature

Variables:

F, F_{x*}	Load on piston, load force of halfbridge	S_{VA}, S_{VB}	Flow area, side A and side B of cylinder
F_G	Load in gravity direction	u_x	Action control value
$G_{Sx(s)}$	Transfer function of valve	$\dot{x}_{hm}$, v	Piston velocity of cylinder
p_A, p_{A*}	Pressure, adjusted pressure - chamber A	x_{sv1Com}	Common required value on valve
p_B, p_{B*}	Pressure, adjusted pressure - chamber B	x_{sv}, x_{svx*}	Spool position of valve
p_s	System pressure	α	Active area ratio
Q_A, Q_{A*}	Flow, adjusted flow - chamber A	β	Flow area ratio
Q_B, Q_{B*}	Flow, adjusted flow - chamber B	***Subscripts***	
Q_s, Q_{max}	System flow, maximal flow	N	Normal value
S_A, S_B	Piston area, chamber A and B	R	Required value
Indexes		RN	Required normal value
x	1 = valve 1, 2 = valve 2	CN	Corrected normal value

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