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CONTENTS

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EDITORIAL

DE CE TRAINING PENTRU MENTENANTA IN FLUID POWER?

Ph.D.Eng. Petrin DRUMEA
MANAGER INOE 2000 – IHP

Suntem in plina criza! Realitatea trebuie acum privita si asumata, fara distorsiuni si ascunzisuri: se vor cerne valorile si cei puternici si buni vor rezista. E tot mai limpede ca „secretul reusitei"- al rezistentei si dezvoltarii unei entitati - sunt oamenii bine pregatiti. Valabil si pentru fiecare dintre entitatile care activeaza in domeniul actionarilor hidraulice, ca si pentru domeniu in ansamblul lui. Din pacate, s-a constatat si se constata in fiecare zi ca s-a ajuns in situatia nefavorabila in care lucratorii nostri din domeniu nu mai pot nici macar intretine si repara echipamentele si utilajele importate. Practica a dovedit ca in ultimii ani specialistii din unitatile productive din Romania au pierdut contactul cu noile tehnologii mecano-hidro-pneumatice, avand astfel deseori probleme in a asigura mentenanta utilajelor moderne importate care dispun de astfel de echipamente (utilaje pentru drumuri, utilaje pentru intretinerea curateniei oraselor, utilaje pentru activitatile ecologice si de intretinere a mediului etc.).

A devenit, asadar, evidenta o crestere a deficitului de lucratori specializati, simultan cu cresterea nivelului tehnic al echipamentelor hidropneumatice importate direct sau in componenta unor utilaje complexe. Urmarea este ca de cele mai multe ori, dupa incercari esuate de mentenanta si reparatii, posesorii acestor utilaje si echipamente au fost obligati sa apeleze la specialistii straini. Cheltuielile cu acestia au devenit foarte mari, conducand la o crestere artificiala si nedorita a costului produselor si serviciilor realizate cu aceste utilaje si echipamente.

Domeniul actionarilor hidraulice este marcat de o serie de tendinte ce vor implica modificari structurale in pregatirea lucratorilor care activeaza in cadrul lui.

Printre acestea: electronizarea si informatizarea echipamentelor si subansamblelor; utilizarea materialelor noi cu performante ridicate in fabricatia echipamentelor hidropneumatice; utilizarea unor tehnologii de fabricatie moderne; utilizarea unor fluide de lucru biodegradabile sau nepoluante.

Se admite in general, la nivel european, ca nivelul de pregatire profesionala asigurat de scoala nu spune nimic despre capacitatea unei persoane de a aplica in practica toate sau macar o parte din cunostintele teoretice acumulate. De aici decurge nevoia de pregatire profesionala, de training specializat, care sa contribuie la imbunatatirea capacitatii de adaptare a intreprinderilor si a angajatilor la realitatile tehnologice, ca si la cele economice, spre a-si gasi locul pe o piata a muncii dinamica si competitiva, asa cum este cea actuala.

De asemenea, nici lucratorii si nici managerii acestora inca nu au inteles importanta ecologizarii activitatii si necesitatea aplicarii unor tehnologii de lucru si de mentenanta care sa permita si sa asigure evitarea impactului negativ asupra mediului.

Un deziderat in plus al activitatii de training pentru mentenanta in domeniul actionarilor hidraulice il constituie aceste noi tendinte. Un alt motiv pentru care e necesar training specializat in domeniu este acela ca la nivel national nu a existat si nu exista scoli pentru lucratorii din acest mare camp de activitate si, ca urmare, asa-zisii specialisti sunt proveniti din personal calificat la locul de munca.

Prin toate cele mentionate, trainingul pentru mentenanta in Fluid Power devine tot mai necesar daca vrem ca domeniul si specialistii sai sa se adapteze si sa se dezvolte chiar si in timp de criza.

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EDITORIAL

WHY TRAINING FOR MAINTENANCE IN THE FLUID POWER FIELD?

Ph.D.Eng. Petrin DRUMEA
MANAGER INOE 2000 – IHP

We are in full crisis. Reality must be faced and assumed, without any distorsions or hideaways. Values will be sifted and only the competitive and performant will withstand. It becomes clearer and clearer that the secret of success, of withstanding and developing an entity, are the well qualified people. This is valid for each one of the separate entities activating in the fluid power field as well as for the entire field. Unfortunately it has been noticed that we reached that level when we have to face a very unfavourable situation, when our workers cannot perform properly not even the maintenance and repair of the main equipment.

Practice has proven that in the last years our specialists from Romania have lost contact with the new fluid power and mechanic technologies, encountering sometimes serious problems in rendering the maintenance and repair of the important modern equipment (road equipment, cleaning maintenance in cities, equipment for ecological and environmental maintenance activities). It has become obvious a decrease of the specialized workers simultaneously with the increase of the technical level of the hydropneumatic equipment imported directly or which are parts of complex installations.

The consequence is that most of times, after failed attempts to a proper maintenance and repairs, the owners of such installations and equipment had to appeal at foreign specialists. The expenses related to this became higher, leading to an artificial and undesirable increase of the costs of the products and services performed using these equipments and installations. The fluid power field is confronted with a series of transformations and development of new trends which will generate structural changes in training workers from the field.

Among these we mention: the computerization and automation of the equipments and installations, the use of new materials with higher performances, at the manufacture of the hydropneumatic equipment, the use of new manufacturing technologies, the use of biodegradable environmentally friendly hydraulic fluids . It is acknowledged in Europe that the level of professional training ensured in schools doesnt say much about someone s ability of applying into practice all or at least a part of the thoretical knowledge acquired.

From this it derives the need for improving the professional training, which to contribute at the improvement of the capability of adaptation of the enterprises and their employees at the new technological and economic realities, for finding the right place on a dynamic and competitive labor market, such as the actual one. In the same time, neither workers nor their managers have yet understood the importance of ecology in performing their specific activities or of the need for applying new operational and maintenance technologies which to allow and ensure a safe protection of the environment, without any reverse effects. The training activity in the field of fluid power aims to focus on these new trends.

Another reason for ensuring a specialized training in the field is that there are no schools for this broad field at national level, therefore the so called specialized personnel comes from workers qualified at their work place.

The training for maintenance in the fluid power field becomes more and more necessary, imperative if we want to see a d evelopment of the field and its specialists even during the economic crisis.

ISSN 1453 - 7303
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SENSITIVITY OF AXIAL PISTON PUMP ON CONTAMINATION

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ABSTRACT

This paper deals with the determination of hydrostatic pumps contamination sensitivity in accordance with ISO 9632. The test rig, process of measurement and results are described.

Key words: pump, contamination, clean procedure, test procedure

1. Introduction

According to special literature about 75% of the failures in the hydraulic systems are caused by the presence of contamination in the system. It is a reason for the contamination control in hydraulic systems. One of the system's contamination control in the last time is the request on the producers of hydraulic pumps and motors to indicate their contaminant sensitivity. The International Standard ISO 9632 specifies a uniform repeatable test method for determining and reporting the contaminant sensitivity (flow degradation) of a fluid power hydraulic positive displacement pump due to wear caused by silica contaminants using classified AC Fine Test Dust (ACFTD). We measured the flow degradation of two axial piston fixed displacement motors, size 75 cm^3, in accordance with ISO 9632. The motors were measured at pump´s running.

2. Test conditions

The test is carried out under constant conditions of speed, pressure and temperature. The concrete conditions in our case were:

- test pump inlet pressure: 1 bar to 1,5 bar
- fluid temperature: 50 °C ± 2 °C
- test pressure 300 bar ± 2% (6 bar)
- test pump shaft speed: 2000 r.p.m ± 2% (40 r.p.m)
- mineral oil OH HM-32, producer Slovnaft, a.s., γ = 28,8 cSt to 35,2 cSt at 40 °C

3. Structure of test rig

The measuring of the flow degradation of axial piston pumps was realised on the test rig, the structure of which is given in figure 1. The test rig was built in co-operation with company SAUER – DANFOSS, SRN.

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Meaning of symbols:

M	regulating electric motor KB 102 – 4, P = 147 kW, n = 2450 rpm, producer MEZ Vsetín, Czech Republic
P1	driving rotational axial piston pump, SPV 23, ZTS Dubnica nad Váhom, Slovakia
M1	driving rotational axial piston motor, SMF 22, ZTS Dubnica nad Váhom, Slovakia
TP	pump which is relatively insensitive to contamination, gear pump ZUS 80, Jihostroj Velešín, Czech Republic
R	tank, Department of hydraulic machinery, STU, Bratislava, Slovakia
TV1, TV2	throttle valve, ED 51, HAWE, Germany
V1,V2,V3	ball valves (2 way), HANSAFLEX, Germany
V4,V5,V6	ball valves (3 way)), HANSAFLEX, Germany
CF	coarse filter, type RF BN/HC 950 D O 5 B 1.0, HYDAC, s.r.o, Slovakia
FF	fine filter, type RF W/HC 330 D G 25 B 1.0, HYDAC, s.r.o, Slovakia
ICH	injection chamber, Department of hydraulic machinery STU, Bratislava, Slovakia
PV	relief valve
HE	heat exchanger
RV	pressure reducing valve, Festo, Germany
FA	air filter, ML08, Hoerbiger, Germany
p1,p2,p3,p4	gauge barometers and tensometric pressure transducers , type 11448, 400 bar, ZPA Jinonice, Czech Republic
T1, T2	temperature gauges, PT100, Ahlborn, Germany
FM1	turbinic flow meter, model PDF0N, 60 – 450 l/min, PARKER FluidConnectors
FM2	turbinic flow meter, 60 – 450 l/min, Unimess, USA
LCTS	liquid contamination test system, HYDAC, s.r.o, Slovakia
MC 32	recording device, fy BMC, Germany

4. Process of measurement

Process of measurement comprises the following consecutive parts:

4.1. Qualification of test rig

In the test circuit the gear pump ZU 80 was installed which is relatively insensitive to contamination. The fluid circulated through the filters until the temperature of fluid reached 50° C ± 2° C. Then the fluid circulated through filters until circuit the fluid contamination background code number was 15/10 or better. The injection chamber was drained and filled with the prepared contaminant. The contents of the injection chamber were injected slowly, over 5 minute’s period. The system was operated at the flow rate 105 l/min. The fluid circulated through the control filters until the contamination background code number 15/10 or better.

4.2. Clean-procedure

The test procedure was followed by the clean-procedure. The pump ran at the test speed 2000 rpm, the fluid passed through the coarse filter for 45 min and then through the fine filter for further 45 min.
After contamination level code number was 15/10 (by ISO 4406), or better (see FIG. 2.), clean-procedure was stopped.

4.3.Test procedure

The test pump was installed. The relief valve RV was fully opened. The motive – power circuit was started and the set test pump was adjusted at the shaft speed 500 rpm. The test pump shaft speed was gradually increased over the period 30 minutes to 2000 r.p.m. While

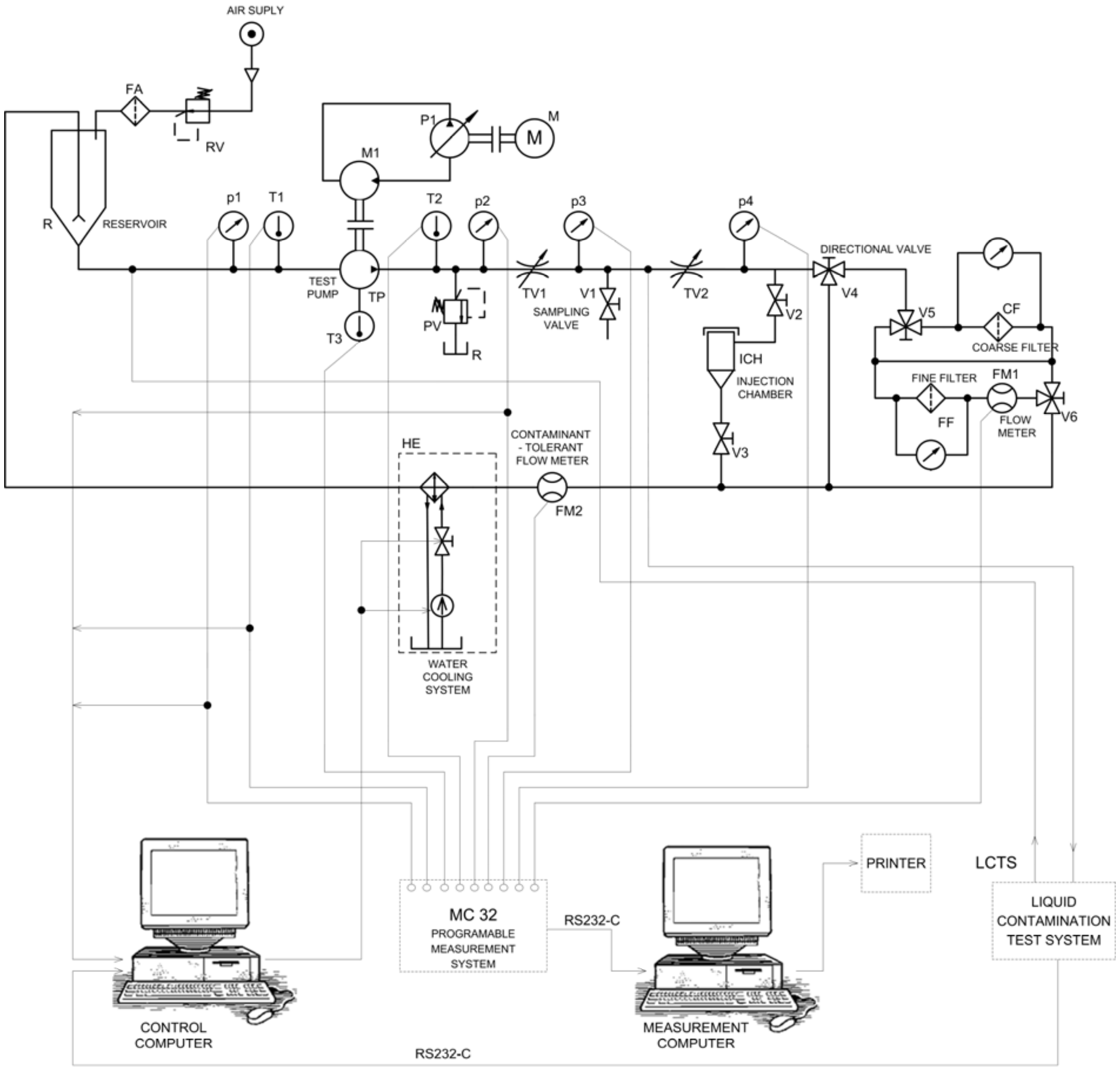

FIG. 1 Test rig

the pump inlet temperature of hydraulic fluid was maintained at 50° C ± 2° C and the test pump inlet pressure at 1 bar to 1,5 bar, the test pump ran at speed 2000 r.p.m:

a) for 5 min at minimum load
b) for 15 min at 75 bar
c) for 15 min at 150 bar
d) for 15 min at 225 bar
e) for 60 min at 300 bar

The system pressure was reduced to minimum and then the pump was stopped. 12 sample slurries were prepared in accordance with ISO 9632. The test pump was started at the speed 500 rpm against the minimum system pressure (relief valve RV1 was opened) with the coarse and fine filters in the circuit. The test pump shaft speed was gradually increased over the period 15 minutes to 2000 r.p.m.

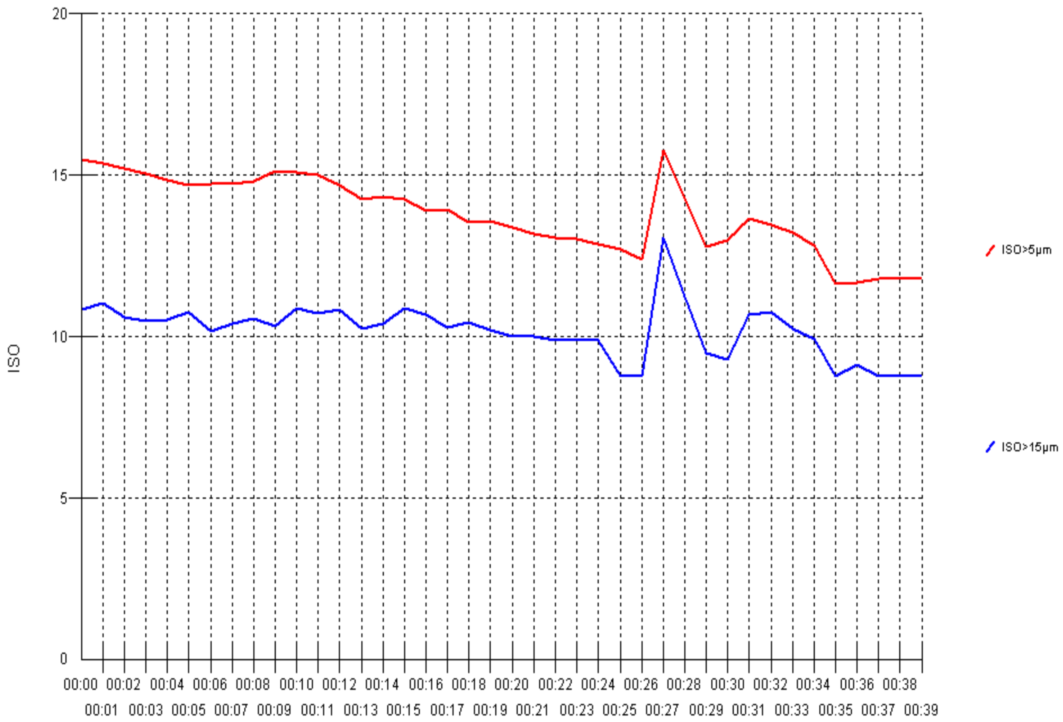

FIG. 2 Document contaminant level

The continued running at the test pump shaft speed 2000 rpm for 15 min at pressure 150 bar, was followed by 15 min at the pressure 300 bar. The system pressure was reduced to minimum (the relief valve RV1 was opened), the filters were switched out of the circuit and the flow rate was recorded. This is the reference flow rate Q_{ref} [l/min].

The system pressure was increased to 300 bar and the following parameters were recorded:

a) the pump flow rate Q_p [l/min]
b) the pump inlet temperature t1 [$^\circ$C]
c) the pump outlet temperature t2 [$^\circ$C]
d) the pump casing temperature t3 [$^\circ$C]

While maintaining following test conditions

a) the test pump inlet pressure: 1 bar to 1,5 bar
b) the fluid temperature: 50° C ± 2° C
c) the test pressure 300 bar ± 2% (6 bar)

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d) the test pump shaft speed: 2000 rpm ± 2% (40 rpm)

One of 0 to 20 μm sample slurries was injected over a period of 5 min. The record parameters Q_p, t1, t2, t3 were recorded continually. Two above mentioned procedures were repeated every 30 min five times (0 to 20 μm sample slurries) without cleaning of the system between injections. After completion of the sixth 0 to 20 μm injections the above mentioned two procedures were repeated using successive injections of the 0 to 30 μm, 0 to 40 μm, 0 to 50 μm, 0 to 60 μm, 0 to 70 μm, 0 to 70 μm, 0 to 80 μm sample slurries.

TAB.1 Presentation of contamination - sensitivity test results

Date tested	:	Test location	: KHS, SjF STU, Bratislava ....
Pump	:...		: System volume 150 $dm^3.min^{-1}$ ....
Speed	: 2000 .... rpm	Grams/injection	: in accordance ISO 9632 .... g
Temperature	: 50.. °C	Reference flow Q_{ref}	: 135, 43 .... $dm^3.min^{-1}$
Pressure	: 300.................................... [bar 1)]	Comments	: ....
Fluid identification	: OH HM- 32, Slovnaft Bratislava .		....
Viscosity at test temperature	: .... cSt 2)		....

Time	Time from start [min]	Size of contaminant added [μm]	Temperature [°C]			Pump speed [r.p.m.]	Flow rate [$dm^3.min^{-1}$]	Volumetric efficiency
			Inlet	Casing	Outlet			
10:40	0	0 to 20	49,82	63,48	54,05	2015,71	135,43	1,0000
10:50	10		49,81	65,07	55,57	2011,01	129,53	0,9564
11:00	20		51,44	66,58	57,20	2004,06	128,21	0,9467
11:10	30		51,22	66,37	57,01	2006,38	126,89	0,9369
11:20	40		49,97	65,28	55,82	1997,12	125,30	0,9252
11:40	60		51,88	70,46	57,76	2004,06	134,09	0,9901
11:50	70		50,75	68,26	56,54	2001,75	133,13	0,9830
12:00	80		49,59	66,36	55,37	1962,41	129,62	0,9571
12:10	90		49,73	66,18	54,86	1860,58	126,92	0,9372
12:20	100		49,33	67,99	57,50	1987,86	130,88	0,9664
12:30	110		48,02	65,01	53,64	1990,18	130,11	0,9607
12:40	120		49,96	65,69	56,00	2004,06	129,85	0,9588
12:50	130		49,88	65,58	55,87	2001,75	128,98	0,9524
13:00	140		50,42	66,39	56,26	1992,49	128,36	0,9478
13:10	150		50,50	66,48	56,61	1999,43	129,30	0,9548
13:20	160		49,74	65,89	56,16	2004,06	128,61	0,9497
13:30	170		49,56	66,04	55,69	1992,49	129,46	0,9559
13:40	180	0 to 30	50,32	66,69	56,56	1994,81	130,25	0,9617
13:50	190		51,29	67,36	57,36	1990,18	126,06	0,9308
14:00	200		50,28	66,39	56,40	1994,81	126,77	0,9360
14:10	210	0 to 40	49,85	66,33	55,83	2011,01	124,59	0,9199
14:20	220		48,99	64,69	55,09	2004,06	123,26	0,9101
14:30	230		50,27	65,12	56,54	2001,75	118,98	0,8785
14:40	240	0 to 50	49,59	64,96	55,86	1983,24	120,27	0,8880
14:50	250		51,21	65,94	57,59	2020,26	120,48	0,8896
15:00	260		49,17	64,87	55,22	1997,12	114,77	0,8474
15:10	270	0 to 60	46,40	52,38	50,03	1784,22	114,99	0,8491
15:20	280		49,27	64,11	57,70	1948,52	115,66	0,8541
15:30	290		50,93	63,57	56,50	1970,51	114,48	0,8453
15:40	300	0 to 70	49,50	61,67	55,12	1995,19	116,49	0,8601
15:50	310		51,66	63,48	56,69	2054,59	110,56	0,8164
16:00	320		48,69	62,09	54,21	2076,19	108,80	0,8034
16:10	330	0 to 80	49,70	62,80	55,05	2049,19	110,38	0,8150

16:20	340		49,79	63,34	55,32	2024,89	109,91	0,8116
16:30	350		49,24	62,80	54,71	2039,74	106,81	0,7887
16:40	360		50,71	63,81	56,04	2030,29	107,93	0,7970
Flow degradation ratio after min								

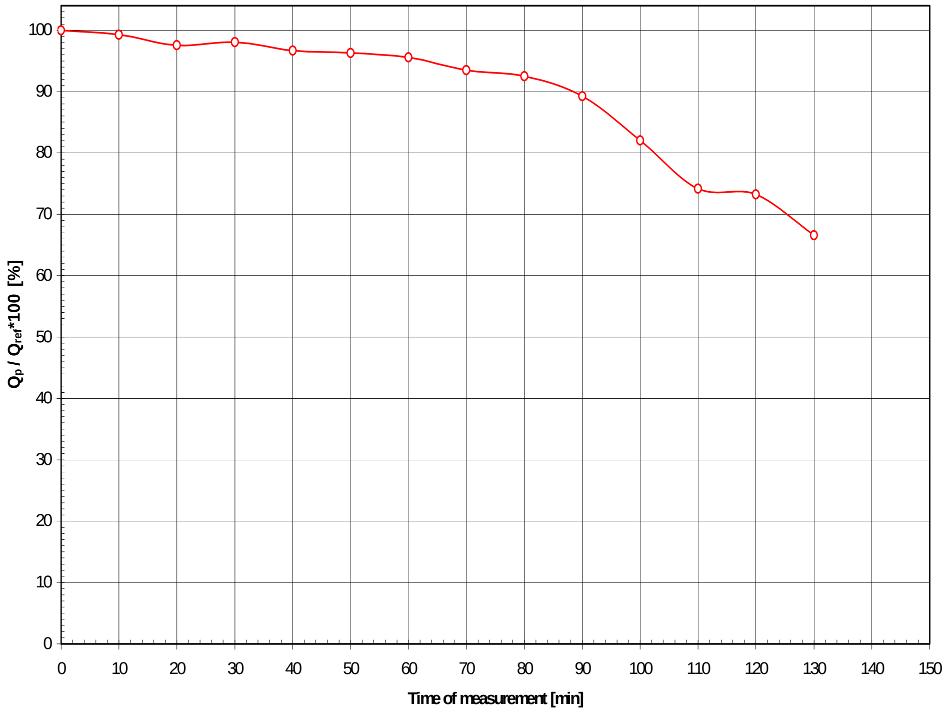

FIG. 3 Degradation of flow – pump No. 1

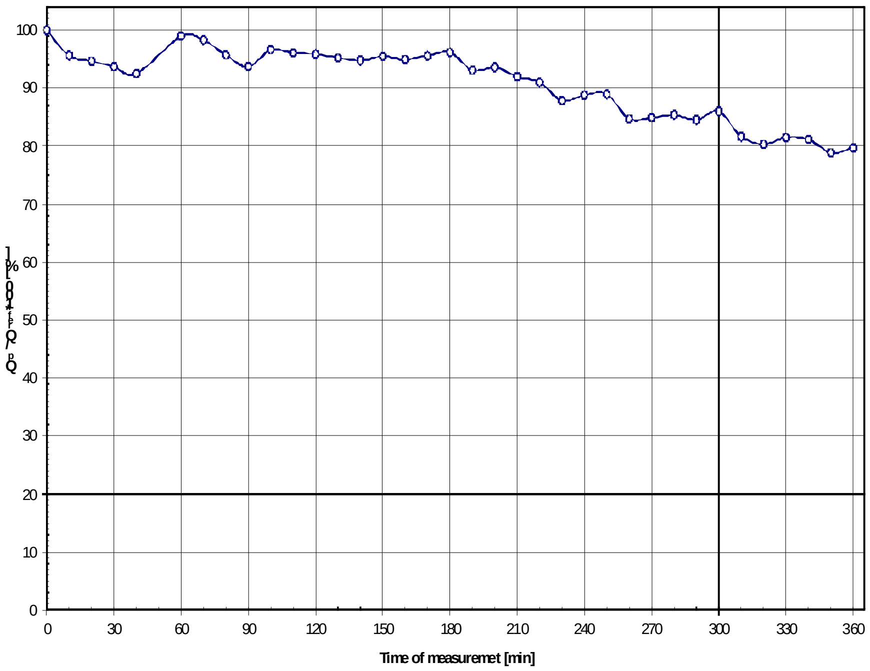

FIG. 4 Degradation of flow – pump No. 2

4. Conclusion

From figures 1 and 2 a faster decrease of second pump’s output flow rate is observed. The test was finished when the flow rate decreased to less than 70 % of its original reference value Q_{ref}
Despite a thorough chain of testing and cleaning procedures, the influence of the sequence of the tested pumps cannot be avoided. Also it would be necessary to examine the possibility of carrying out the mentioned tests at lower parameters (pressure, flow rate, speed). That would lower power consumption during testing high-power pumps.

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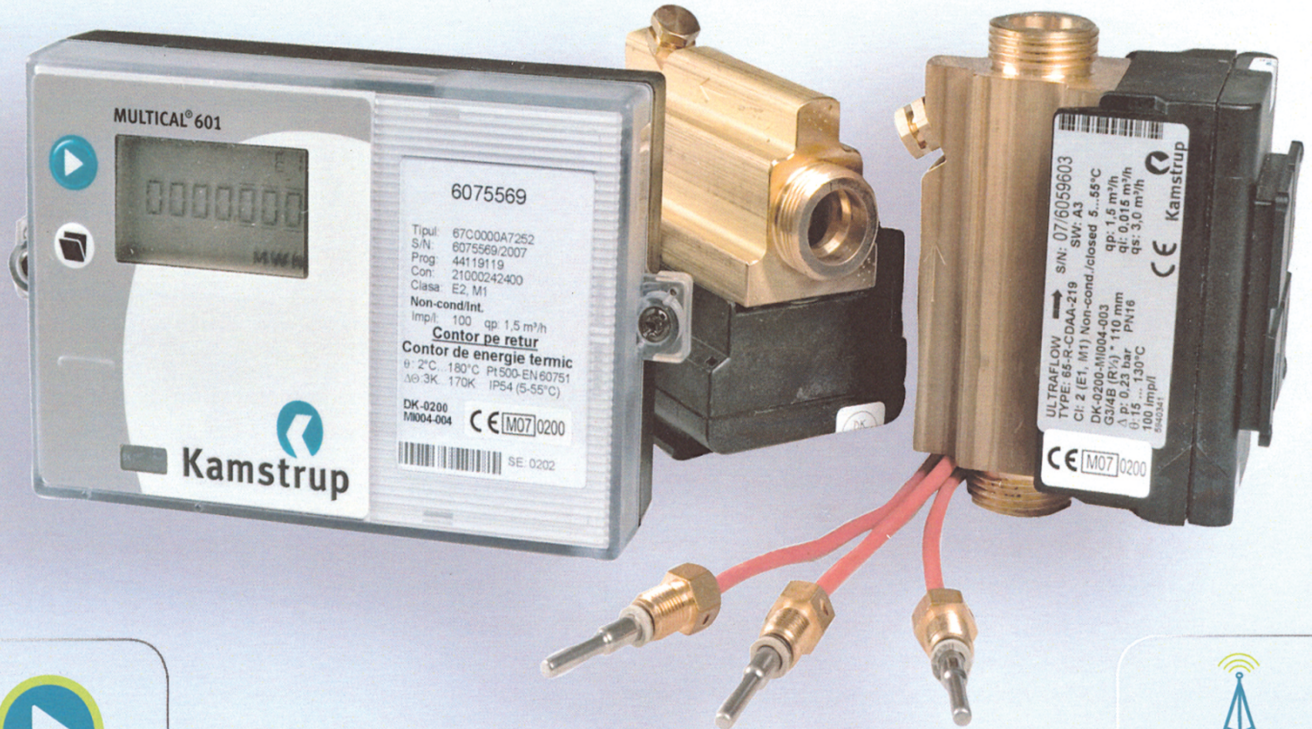

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THE CAVITATION IN ZONE BODY-SPOOL VALVE FOR HYDRAULIC DISTRIBUTOR

BY

CALIN RASZGA[1]; VICTOR BALASOIU [1];

Abstract. *The cavitation phenomena which occurring in hydraulic machinery and hydraulic equipment, is for many years a research topic, which lead to great running improvements of the hydraulic systems. Cavitation occurs in some elements of hydraulic drive devices as a result of great increases of the flow velocity. In distributors it represents the major limits the possibility to reduce the flow capacity beyond some values. The initiation and the development of the cavitation in hydraulic drive systems present some particularities originated both in the liquid employed (especially mineral oil) and the severe running conditions. The cavitation erosion is generally not a very stressing factor. On the other hand, as a result of important modifications in the characteristics of the working fluid, when the pressure decreases (increasing of the gas and steam content) the running of the system is heavily disturbed.*

Having as a start point in general the definition of cavitation coefficients for the directional valves with cylindrical spools. For this definition, there were taken into account all the important aspects of the flow geometry and the elements depending on the fluid nature. By attentive examinations of the flow through the specific hydraulic resistance it was established a relation to obtain the cavitation reserve of the system.

Key words : cavitation erosion, electrohydraulic distributors, hydraulic systems,

1. The cavitation phenomenon in distribution section distributor body - spool valve

The hydraulic resistance describe hydraulic system elements with diverse functional role, and the diversity of type construction assures the purpose for their role. The majority of hydraulic resistance works by strangle the flow vein . From this reason the cavitation phenomena can occurs during with increasing speed and the different pressure values in different points of the hydraulic resistance. In hydraulic distribution apparatus, the cavitation phenomenon occurs in case of command resistance with small aperture, where the speed will be increased, and from Bernoulli equation results that the pressure drops in the case when exists a big drop of pressure on the command resistance.

The distributor hydraulic track is characterized by the distributor body-spool valve geometry, and different type of spool valves geometry. For studying the cavitation regime in hydraulic resistance distributor body-spool valve , the research had focus to determine energetic spectrum of analyse cavitation effect among the hydraulic distribution apparatus [6].

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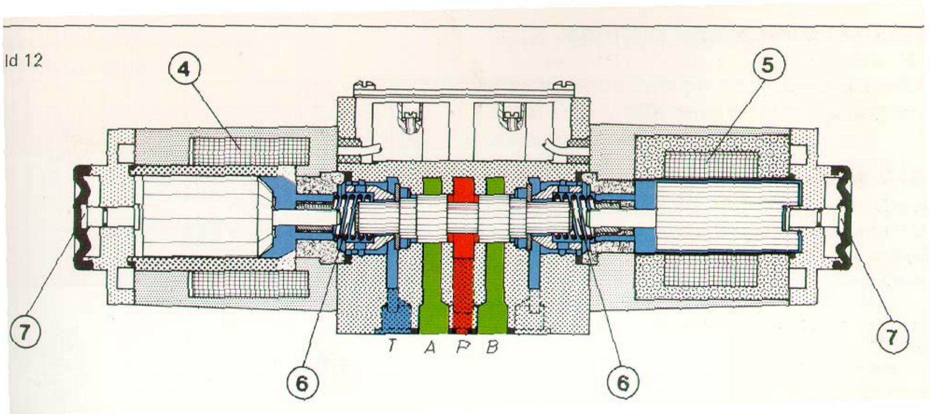


Fig.1. Electrohydraulic distributor

In a series of anatomization Numachi broach the problem of cavitation inside Venturi tubes and inside the diaphragm to determine the cavitation start moment ,by using the relation:

$$\sigma_{inst} = \frac{p_i - p_d}{\Delta p} = \frac{p_i - p_d}{\rho \frac{v_0^2}{2}} \qquad (1)$$

The initiation and the development of the cavitation inside the hydraulic servo valve with cylindrical spool valve is possible due to local flowing conditions ,which can occurs by increasing the speed and lowering the pressure and it is characterized through the cavitation coefficient [1, 2,3,5,6,7,8].

Where: p_2 pressure on the strangled section , p_v the vaporization pressure of the liquid, v_0 the reference medium speed to input. Numachi ,starting from the definition relation of the flow rate coefficient ,he determine by experimental the coefficient :

$$C_k = \frac{Q_k \sqrt{1 - m^2}}{\frac{\pi}{4} d_2^2 \sqrt{\frac{2g(p_{1k} - p_{2k})}{\gamma}}} \qquad (2)$$

Where: C_k - flow coefficient in cavitation regime Q_k - flow of the cavitation regime ,p_{1k} ,p_{2k} input and output pressure in cavitation regime. In figure 1. its shown the flow coefficient in the commission with the value of the cavitation coefficient .

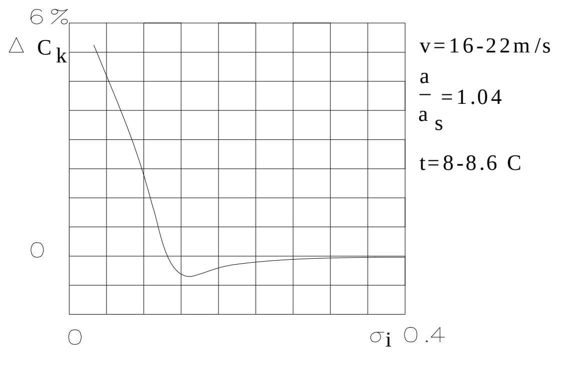
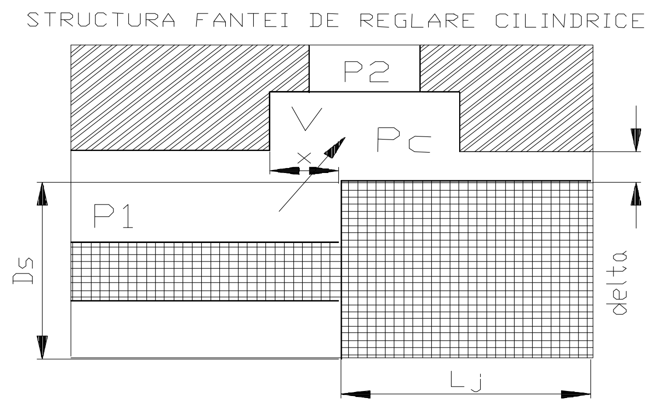


Fig. 2.



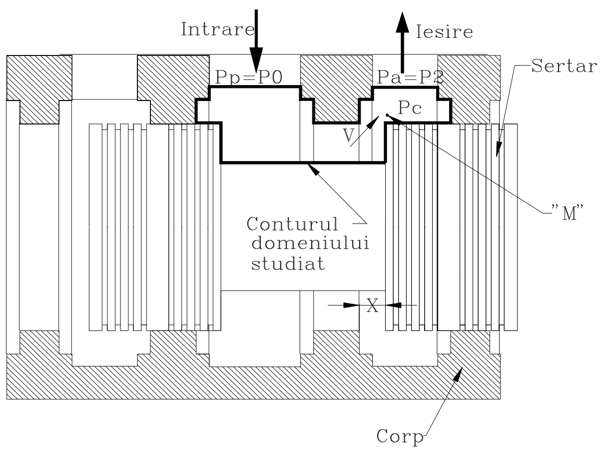


Fig. 3.

The spool-valve distributor its characterized by Martin and Wiggert [2] as being made of several hydraulic resistance, which works at high pressure and increasing speed. The most important is high speed jet with circular form, which appears at the exit of adjusting slit through cylindrical room and the exit circuit of apparatus.(Fig 2). If the jet is free for small apertures and two-dimensional flow model, it will reach to a value for flow coefficient Cd of 0.673, and a 69° in ratio with spool-valve axis, calculated by McCloy and Martin in 1973 [3] and Raszga [6].

In distributors with cylinder spool-valves (fig.1)can be notice two types of incipient cavitation correlated to free or attached jet. In case of attached jet can be notice a cavitation development along wall and the cavitation bubbles appears due to increased friction tension near to wall, that leads to a liquid break and appearance of new liquid-gas interface.

For free jet cavitation appears in case when the pressure from the exit room drops below a limit value, but it can be also higher than the value of vaporisation pressure of the employed liquid.

The development of cavitation in distributors with cylindrical spool-valve can be identified by recording dynamical values, which are pulsating pressure in the exit room of circular jet, which represent a different noise from the turbulent jet, to be distinguished in energetic spectrum of measured values.

Martin and Wiggert [4]. Due to development conditions of phenomenal and assumed measurements for experimental appliance. The type of cavitation that appear is developed cavitation, not a gas cavitation that appears in other author's works incriminated. Point out the presence of dissolved air in 10% has a reduced effect due to short time of existence of cavity in case of cavitation implosion. But they are responsible whit forming of air bubble with high dimensions which can affect system dynamics.

2. Determination for general expression of cavitation coefficient in case for a distributor with cylindric spool-valve.

If we consider the structure of hydraulic trace for a distributor with cylindric spool-valve, in maximum speed assumption, respective lower pressure, appears near to minimal section of trace from the adjusting slit area. (Point M from fig.3)

Making use by cavitation coefficient of distribution section, ratio to input section we obtain:

$$\sigma_{corp0} = \sigma_{D0} = \left(\frac{V_{\max}^2}{V_0^2} - 1\right) + \frac{h_{p0M}}{\frac{V_0^2}{2g}} \qquad (3)$$

The energetic loses therm can be write starting from the equation of energy transfer, to take into account that the hydraulic loses are preponderant due to perturbation of speed field.
Reference to entrance section we obtain:

$$h_{p0M} = \varsigma_{0M} \frac{V_0^2}{2g} \qquad (4.1)$$

The distributor cavitation coefficient became:

$$\sigma_{D0} = \left(\frac{V_{\max}^2}{V_0^2} - 1\right) + \frac{h_{p0M}}{\frac{V_0^2}{2g}} = \left(\frac{V_{\max}^2}{V_0^2} - 1\right) + \varsigma_{0M} \qquad (4.2)$$

Reporting the difference of pressure between in and out, we take input section as reference:

$$\sigma_{inst0}^{*} = \frac{p_0 - p_V}{\Delta p_{02}} \qquad (5)$$

$$\sigma_{d0}^{*} = \frac{\rho V_0^2}{2\Delta p_{02}}\left[\left(\frac{V_{\max}}{V_0}\right) - 1 + \zeta_{0M}\right] \qquad (6)$$

for output section considered as reference:

$$\sigma_{inst2}^{*} = \frac{p_2 - p_V}{\Delta p_{02}} \qquad (7)$$

$$\sigma_{D2}^{*} = \frac{\rho V_2^2}{2\Delta p_{02}}\left[\left(\frac{V_{\max}}{V_2}\right) - 1 + \zeta_{M2}\right] \qquad (8)$$

Reporting to section "x" :

$$\sigma_{instx}^{*} = \frac{p_x - p_V}{\Delta p_{02}} \qquad (9)$$

$$\sigma_{Dx}^{*} = \frac{\rho V_x^2}{2\Delta p_{02}}\left[\left(\frac{V_{\max}}{V_x}\right) - 1 + \zeta_{xM}\right] \qquad (10)$$

Similar to cavitation coefficient [1,6,7,8] ,the local loosing coefficient signify the energetic looses which emerge on trace of the hydraulic distributor with cylindrical spool valve, due to strong perturbation geometrical structure of the speed field of the inner hydraulic trace.

The major loose it take place in adjusting slit area with x opening. At the slit output appears a drown annular jet in the exit room ("A" or "2") .In fig. 4 is represented the spectrum of current lines on the all considered domain, and in fig. 5 we have represented a detailed flow structure in

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adjusting split area. Inquiring the energy transfer equation between section "0" and "f" the local energetic losses coefficient ζ_{0M} that appear on hydraulic trace with cylindrical spool valve becomes:

$$\sigma_{M2/0} = \frac{p_f - p_2}{\rho \frac{V_0^2}{2}} - \frac{K_0^2}{K_2^2} + \frac{1}{C_c^2} \cdot \frac{K_0^2}{x^2} \qquad (11)$$

The reserved cavitation coefficient in all matters of connote, marking out the difference between cavitation coefficient of installation and the cavitation coefficient of device (apparatus) – hydraulic distributor . Long time as the value of reserved cavitation coefficient is positive , the regime of work remain cavitation free.

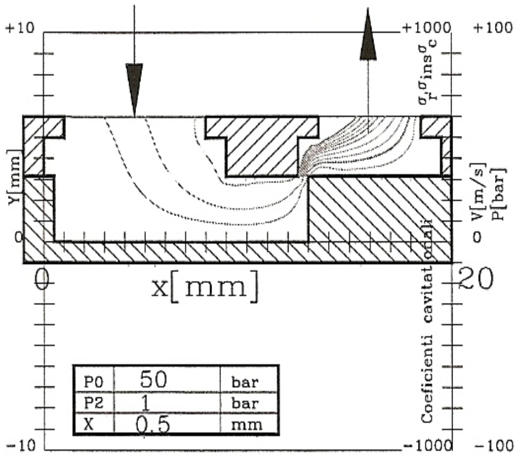

Fig. 4

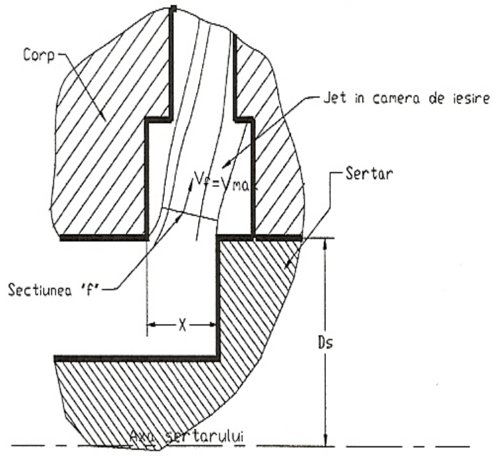

Fig.5

In the moment when the value of reserved cavitation coefficient is null , the working cavitation regime is incipient , and in one point from the hydraulic trace the pressure drops enough to make possible the air bubble emerge ,and the liquid break and the gas vapour development is high enough . In this working regime and only now ,the value of cavitation coefficient of device is equally numeric with value of cavitation coefficient of the distributor , although they differ as analytic relations .From analytic point of view , the characterization of free cavitation regime of incipient, developed and industrial cavitation and super cavitation it is realised. Practical knowledge of coefficients values is difficult by using local values characteristic flowing field through distributor spool valve .The all definition for cavitation coefficient ,in particular for distributor with cylindrical spool valve ,was started from rigorous definition of cavitation regime and cavitation coefficient.

3. Conclusion

Elaborated documentation about studied phenomenon – the cavitation phenomenon in the hydraulic distributor – spool valve ,allow knowledge from theoretical research and experimental ,and offer the based elements for extend knowledge of phenomenon. To accentuate the existence of cavitation phenomenon inside the acting hydraulic installation ,which works usually with mineral oil ,and especially in case of distributor with spool valve, it was accomplished by experimental way and deduced with numerical way by using calculus software with method of finite element. Correlation of experimental result by Anton I [1], Martin and Wiggert [4, 5] with indicated numerical results by Th. Gaurer [3, 5],Toshiyuki [7], Bernd and Muntean [2, 7]. To determine an analytic numerical relation even approximated it is necessary if we want to describe the initiation and cavitation phenomenon effect in operation of the distributors with spool valve.

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The presented work has been supported from the National University Research Council Grant CNCSIS-IDEI nr.35/68/2007 and CNMP 1467/21047/2007

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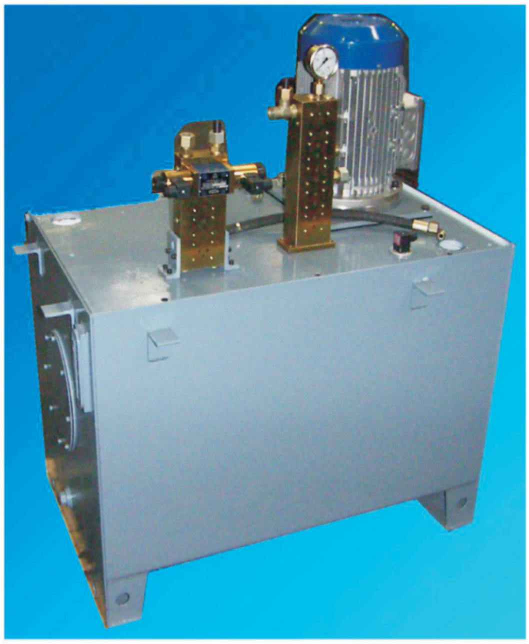

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MATHEMATICAL MODEL FOR A SERVOVALVE – LINEAR MOTOR

VICTOR BALASOIU[1], MIRCEA POPOVICIU[1], ILARE BORDEASU[1],

Abstract. *The system piston-cylinder supplied by servo directional control valve and used for motion/positioning is indispensable as translation module in various applications of hydraulic automations. A mathematical model of the phenomena produced during the running process is necessary for the rigorous analysis and synthesis of such devices.*

In agreement with its structure, such a mathematical model corresponds to a servo mechanism that works like an open control chain, for which the electro hydraulic servo valve adjust the position of the piston for a cylinder loaded with external forces. An electronic control and driving circuit assures the feedback. A correct choice both for the electro hydraulic servo valve and the measuring technique can improve the positioning precision but also a quick response and the stability of the system.

The developed model is used to establish the transfer functions, for various computing hypothesis, of a previously defined hydraulic system. The transfer functions for the cylinder (in correlation with its load) and for the assembly, as a whole, must be established. By assembling the mathematical models for servo valve, cylinder and load it result the circuit scheme presented in the paper. The frequency characteristics of the system are determined considering the servo valve as an ideal system but also by taking into account the delay introduced by the servo valve. Finally there is obtained the transfer function for the open or closed circuit of the hydraulic system.

In the second part of the work, appealing the developed mathematical model and the transfer functions it is established a program for analyzing the dynamic behavior for the particular hydraulic cylinder "DYNAMIC 1". For the geometric and running parameters of a given translation module, there have been computed and represented graphically the following characteristics: amplitude-phase-frequency, the hodograph of the transfer function, the answer function for a step signal taking into account an ideal and real system, in an ideal and real circuit, for open and closed circuits. Analyzing the frequency and motion characteristics, the pressure and flow rate and the place of roots it has been concluded that the studied systems are stable dynamically, conclusion confirmed also by analyzing the third order transfer function with the program DYNAMIC 1.

The analyzing method and the frequency and step signal characteristics put into evidence the influence of the geometric and running parameters upon the stable condition of running for the modules taken into considerations. The mathematical model developed and the computing procedure represents an efficient tool for an optimized design of the translation modules used in the drive/automation hydraulic system.

Key words, mathematical models, electrohydraulic distributors, servovalve electro hydraulic, hydraulic cylinder, axes hydraulic linear, transfer function, hydraulic, transfer function,

1.Structure of the positioning system

In conformity with his structure, the model of the system correspond to a servo transmission working as an open loop chain, in which, the electro-hydraulic servo directional valve (EHSDV) adjust the piston position Z_C of a cylinder loaded with external forces. The reaction is assured by a positioning transducer and worked out by the electronic control circuit (fig. 1). Choosing an adequate EHSDV and an adequate measuring technique improves the positioning precision, the velocity and the stability of the system.

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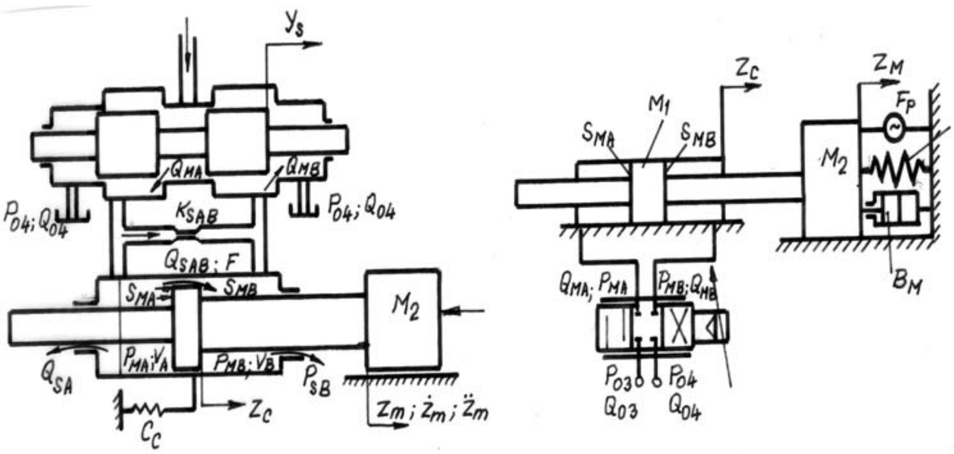
As it can be seen in fig. 2 the value F_p is the force acting on the piston and Δi_C is the EHSDV command current for which is obtained the displacement Ys for the spool valve and the position Z_C for the hydraulic cylinder piston. In order to define the assembly EHSDV-cylinder (fig. 2) and to obtain the transfer function, some simplifying hypothesis presented in [1] have been made.

2. The transfer function of the execution hydraulic cylinder

For the execution hydraulic cylinder in correlation with the load (fig. 1 and 2) the transfer function can be written:

a) as the ratio_ between the actual displacement and the liquid flow capacity, with the condition $F_p = 0$.



a) b)

Fig. 1 The structure of the system ehsdv-cylinder-load

$$H_{C1}(s)=\frac{Z_c(s)}{Q_M(s)}=\frac{\dfrac{1}{S_M}}{\dfrac{V_M M_S}{4E_S S_M}S^3+\left[\dfrac{B_S+F_C}{4E_U S_M}V_M+K_{sie}\dfrac{M_S}{S_M^2}\right]S^2+\left[1+K_{sie}\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]S+\dfrac{C_S K_{sie}}{S_M^2}} \qquad (1)$$

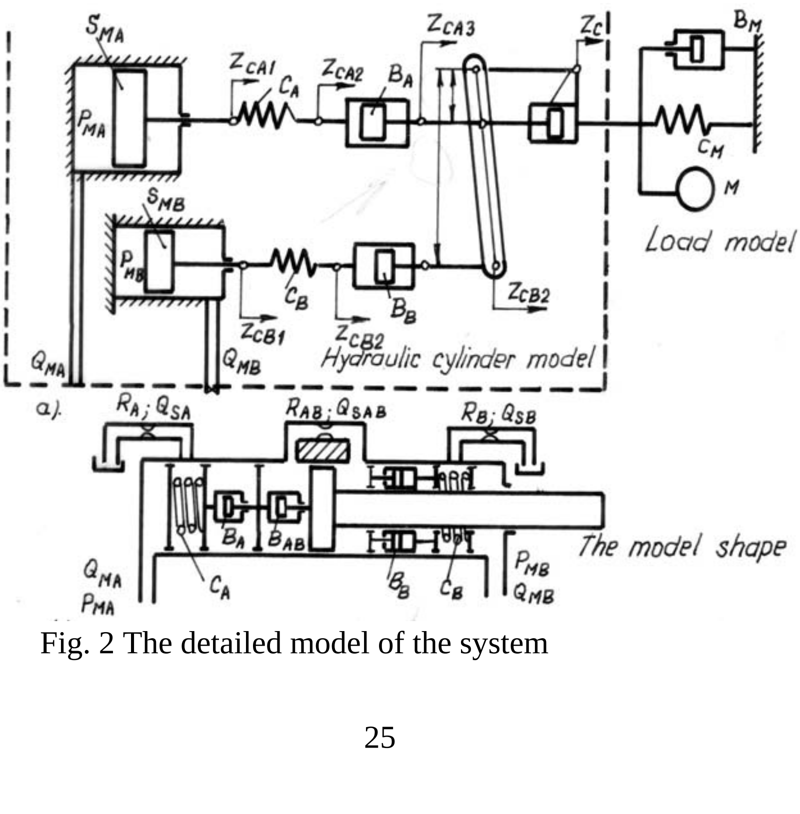


Fig. 2 The detailed model of the system

or in the form:

$$H_{C1}(s)=\frac{\dfrac{1}{S_M\left[1+K_{sie}\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]}}{\dfrac{V_M M_S}{4E_S S_M\left[1+K_{sie}\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]}S^3+\dfrac{\left[\dfrac{B_S+F_C}{4E_U S_M}V_M+K_{sie}\dfrac{M_S}{S_M^2}\right]}{\left[1+K_{sie}\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]}S^2+S+\dfrac{\dfrac{C_S K_{sie}}{S_M^2}}{\left[1+K_{sie}\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]}} \quad (2)$$

taking into account the nomenclature defined in [1] for K_M, ω_n and ξ results:

$$H_{C1}(s)=\frac{K_M}{S\left[\dfrac{S^2}{\omega_n^2}+\dfrac{2\xi}{\omega_n}S+1\right]+\dfrac{C_s K_{sie}}{S_M}K_M} \quad (3)$$

b) as the rate between the actual displacement of the piston and the acting force, with the condition $Q_M = 0$:

$$H_{C1}(s)=\frac{\dfrac{1}{S_M\left[1+K_{sie}\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]}\left[K_{sie}+\dfrac{V_M}{4E_U}S\right]}{\dfrac{V_M M_S}{4E_S S_M\left[1+K_{sie}\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]}S^3+\dfrac{\left[\dfrac{B_S+F_C}{4E_U S_M}V_M+K_{sie}\dfrac{M_S}{S_M^2}\right]}{\left[1+K_{sie}\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]}S^2+S+\dfrac{\dfrac{C_S K_{sie}}{S_M^2}}{\left[1+K_{sie}\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]}} \quad (4)$$

and the transfer function becomes:

$$H_{C1}(s)=\frac{K_A(T_A S+1)}{S\left[\dfrac{S^2}{\omega_n^2}+\dfrac{2\xi}{\omega_n}S+1\right]+\dfrac{C_s K_{sie}}{S_M}K_M} \quad (5)$$

The complete transfer function of the hydraulic cylinder will be:

$$H_{C1}(s)=\frac{K_M Q_M(s)-K_A(T_A S+1)F_p(s)}{S\left[\dfrac{S^2}{\omega_n^2}+\dfrac{2\xi}{\omega_n}S+1\right]+\dfrac{C_s K_{sie}}{S_M}K_M} \quad (6)$$

in comparison with the relations given in [2, 3], the relation (6) take into account the influence of the damping constant B_M, the elasticity of the system C_M, and the Coulombian friction. By introducing the linear equation of the flow capacity $Q_M = K_{Qy}\cdot\Delta Y_s(s)-K_{Qp}\cdot\Delta p_{MAB}$, in the vicinity of the considered point, and equalizing it with the flow capacity demanded by the hydraulic cylinder Q_M [1], it will be obtained**:**

$$H_{Zc}=\frac{\dfrac{K_{Qy}}{S_M}Y(s)-\dfrac{K_{Qpy}-K_{sie}}{S_M}\left[1+\dfrac{V_M}{4E_S\left(K_{Qpy}+K_{sie}\right)}S\right]F_p(s)}{\dfrac{V_M M_S}{4E_U.S_M^2}S_M^2+\left[\dfrac{K_{Qpy}+K_{sie}}{S_M^2}M_S+\dfrac{B_S+F_C}{4E_U S_M^2}V_M\right]S^2+\left[1+(K_{sie}+K_{Qpy})\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]S+\dfrac{C_S(K_{Qpy}+K_{sie}}{S_M^2}} \quad (7)$$

Neglecting $F_p(s)$ and applying the Laplace transform it results the transfer function for the hydraulic cylinder positioning systm:

$$H_{Zc}=\frac{\dfrac{K_{Qy}}{S_M}Y(s)}{\dfrac{V_M M_S}{4E_U.S_M^2}S_M^2+\left[\dfrac{K_{Qpy}+K_{sie}}{S_M^2}M_S+\dfrac{B_S+F_C}{4E_U S_M^2}V_M\right]S^2+\left[1+(K_{sie}+K_{Qpy})\dfrac{B_S+F_C}{S_M^2}+\dfrac{C_S+V_M}{4E_U S_M^2}\right]S+\dfrac{C_S(K_{Qpy}+K_{sie})}{S_M^2}} \quad (8)$$

that can be written:

$$H_{Zc}=\frac{Zc(s)}{Y_S}=\frac{F_5}{F_1 S^3+F_2 S^2+F_3 S+F_4} \quad (9)$$

with the nomenclature used in [1], or in the normalized form, (9) turns in:

$$J_0 = \frac{F_5}{F_4};\; J_1 = \frac{F_1}{F_4};\; J_2 = \frac{F_2}{F_4};\; J_3 = \frac{F_3}{F_4};\; \omega_n = \sqrt{\frac{F_3}{F_1}};\; \xi = \frac{F_2}{2.\sqrt{F_1.F_3}} \qquad (10)$$

where:

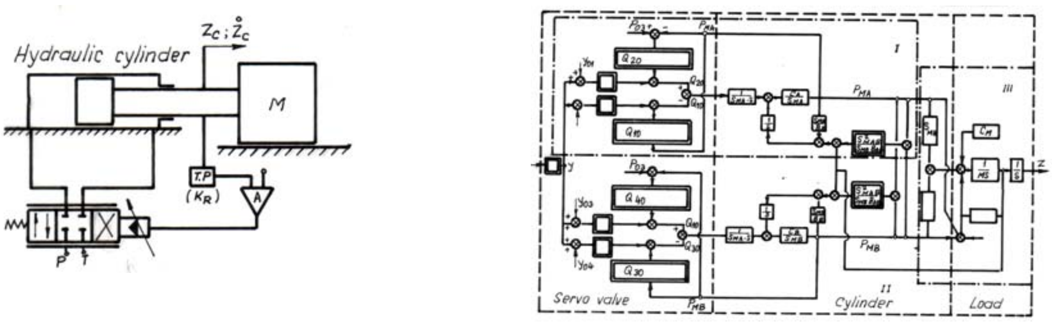
$$\omega_n = \sqrt{\frac{F_3}{F_1}} = \sqrt{\frac{4E_S S_M^2 + 4E_S(B_S + F_C)(K_{Qp} + K_{sie}) + C_S V_M}{V_M.M_S}}$$ -is the natural pulsation of the load cylinder

$$\xi = \frac{F_2}{2\sqrt{F_1.F_3}} = \frac{4E_S M_S(K_{Qp} + K_{sie}) + (B_S + F_C)V_M}{2\sqrt{V_M M_S\left[4E_S S_M^2 + 4E_S(B_S + F_C)(K_{Qp} + K_{sie}) + C_S V_M\right]}}$$ -

is the damping factor of the hydraulic load cylinder.

3.The model of the system EHSDV-cylinder-load

Assembling EHSDV, cylinder and load results the scheme presented in fig. 2. Resorting to the models of block schemes presented in [1, 10, 11, 12] it was worked out the unit scheme of the system (fig. 3) from which came out the numerous parameters involved and their nonlinearely, in conformity with the relations presented in [1, 2, 3]. The motion equations determined on the basis of the energetic balance of the subassemblies are presented in tab. 1. The equations have been obtained through analyze of the block scheme (fig. 3) of the assembly cylinder-load.



a) b)

Fig. 3

4. Frequency characteristics of the EHSDV system ideal and real

In order to analyze the characteristics of the system ehsdv-cylinder-load in conformity with model presented in [1, 10] there will be introduced the following transfer functions, the input parameter being the spool valve stroke y_s.

the transfer function of the piston stroke z_c is:

$$H_{Zc}(s) = \frac{Z_C(s)}{Y_S(s)} = \frac{M_0 + M_1.s}{M_6 S^4 + M_5 S^3 + M_4 S^2 + M_3 S + M_2} \qquad (11.a)$$

therefore, the piston stroke becomes:

$$Z_C(s) = H_{SV}(s).H_{ZC}(s).\Delta i_C \qquad (11.b)$$

maintaining the nomenclature given in [1], the transfer functions of pressures p_{ma} and p_{mb} are:

$$H_{pMA}(s)=\frac{p_{MA}(s)}{Y_s(s)}=\frac{H_{41}s^3+H_{31}s^2H_{21}s\ \ H_{11}s}{M_6S^4+M_5S^3+M_4S^2+M_3S\ \ +M_2}$$
$$H_{pMA}(s)=\frac{p_{MA}(s)}{Y_s(s)}=\frac{H_{42}s^3+H_{32}s^2H_{22}s\ \ H_{12}s}{M_6S^4+M_5S^3+M_4S^2+M_3S\ \ +M_2} \qquad (12.a)$$

Therefore, the working pressures in the hydraulic cylinder chambers becomes:

$$p_{MA}(t)=Y_s(t)\left|H_{pMA}(j\omega)\right|$$
$$p_{MA}(t)=Y_s(t)\left|H_{pMA}(j\omega)\right| \qquad (12.b)$$

The transfer functions for the flows q_{ma} and q_{mb} are obtained in the same way as for the relations (11 and 12):

$$H_{QMA}(s)=\frac{Q_{MA}(s)}{Y_s(s)}=K_{Qy}-K_{Qp}.H_{pMA}(s)=K_{Qy}-K_{Qp}\cdot\frac{H_{41}s^3+H_{31}s^2H_{21}s\ \ H_{11}s}{M_6S^4+M_5S^3+M_4S^2+M_3S\ \ +M_2}$$
$$H_{QMB}(s)=\frac{Q_{MB}(s)}{Y_s(s)}=K_{Qy}+K_{Qp}.H_{pMB}(s)=K_{Qy}+K_{Qp}\cdot\frac{H_{42}s^3+H_{32}s^2H_{22}s\ \ H_{12}s}{M_6S^4+M_5S^3+M_4S^2+M_3S\ \ +M_2}$$

(13.a)

Therefore, the flows in the hydraulic cylinder control chambers are:

$$Q_{MA}(t)=Y_s(t)\left|H_{QMA}(j\omega)\right|$$
$$Q_{MB}(t)=Y_s(t)\left|H_{QMB}(j\omega)\right| \qquad (13.b)$$

The transfer functions computed with the relations (11, 12, 13) have as input parameter the value y_s, considering that the motion of the spool valve as being rigidly connected to the command, in other words as if the spool valve stroke follows immediately and identically the amplitude of the control current δi_c. in practice, the stroke y_s has a certain delay in comparison with the control current δi_c (the ehsdv properties can be described by a delay of order i-iii). the actual characteristics of the system ehsdv-cylinder-load, having as input element δi_c, are easily described if the transfer functions (11, 12, 13) are multiplied with the ehsdv transfer function given in [1, 3, 4]. in such conditions the transfer functions for the real system become:

The transfer function for the piston stroke, z_{crs}:

$$H_{ZCR}(s)=H_{SV}(s).H_{ZC}(s) \qquad (14.a)$$

Respective:

$$Z_{CR}=\left|H_{ZCR}\right|.Y_{SN}\cdot\frac{\Delta i_C}{\Delta i_{C\,N}} \qquad (14.b)$$

The transfer functions for the pressures p_{ma} and p_{mb}:

$$H_{pMAR}(s)=H_{SV}(s).H_{pMA}(s)$$
$$H_{pMAR}(s)=H_{SV}(s).H_{pMA}(s) \qquad (15.a)$$

respective:

$$p_{MAR}(t)=\left|H_{pMA}(j\omega)\right|.Y_{SN}\cdot\frac{\Delta i_C}{\Delta i_{CN}}$$
$$p_{MBR}(t)=\left|H_{pMB}(j\omega)\right|.Y_{SN}\cdot\frac{\Delta i_C}{\Delta i_{CN}} \qquad (15.b)$$

the transfer functions for the flows q_{ma} and q_{mb} of the hydraulic cylinder chambers:

$$H_{QMAR}(s)=H_{SV}(s).H_{QMA}(s)$$
$$H_{QMBR}(s)=H_{SV}(s).H_{QMB}(s) \qquad (16.a)$$

respective:

$$Q_{MAR}(t) = \left|H_{QMAR}(j\omega)\right|.Y_{SN} \cdot \frac{\Delta i_C}{\Delta i_{CN}}$$
$$Q_{MBR}(t) = \left|H_{QMBR}(j\omega)\right|.Y_{SN} \cdot \frac{\Delta i_C}{\Delta i_{CN}} \tag{16.b}$$

In order to compute and plot the amplitude-phase-frequency characteristics, for both systems (with real and ideal ehsdv) it was worked out a computation program symulink-dynamic-system, included in the package shap [1]. for controlling the displacement z_c of the hydraulic cylinder piston an inductive transducer was used. the transfer function of the measuring system is:

$$H_{RC}(s) = \frac{\Delta U_{RC}(s)}{Z_C(s)} = k_{rc} \tag{17}$$

where k_{cb} is the constant of the reaction inductive transducer, and δu_{rc} is the output value of the transducer.

5. Transfer function for the hydraulic system with open and closed loop

Both the transfer functions for the direct branch of the real system (13) and for the reaction branch (16) being available, it can be written the transfer function for the open loop system:

$$H_D(s) = H_{ZCR}(s).H_{RC}(s) \tag{18}$$

which put into evidence the total amplifying factor of the system. the transfer function for the closed loop system (fig. 3) is:

$$H_{1ZC}(s) = \frac{H_{ZCR}(s)}{1 + H_D(s)} = \frac{H_{ZCR}(s)}{1 + H_{ZCR}(s).H_{RC}(s)} = \frac{H_{ZCR}(s)}{1 + K_{RC}(s).H_{ZCR}(s)} \tag{19}$$

The amplitude and the phase of the transfer function is [1]:

$$\begin{cases} \left|H_{1ZC}(j\omega)\right|_{dB} = \dfrac{H1ZC(j\omega)}{1 + K^2{}_{RC}.H^2{}_{1ZC}(s) + 2KRC.H_{1ZC}(s).\cos\varphi} \\ \varphi^0_{H1ZC}(j\omega) = arctg\dfrac{\sin\varphi}{\cos\varphi + K_{RC}.H_{1RC}(j\omega)} \end{cases} \tag{20}$$

In order to determine the performance parameters of the closed loop system, as a general rule, it is used the transfer function for the of the open loop system which, on the one hand has the preponderant weight and on the other hand is easier to be computed.

6. Dynamic 1- analyzing program for the dynamic behavior of the execution cylinder characteristic curves

Taking into account the mathematical model defined in cap. 2 and appealing to the analyzing mode of characteristic parameters, respective to the running performances introduced in [1, 5, 8, 13] it was realized the numerical simulation program dynamic 1. this program allows to study the response both for frequency and step signal at transitional regimes. the stability conditions are obtained by applying to the iii order transfer equations (8, 9) the criteria routh-hurwitz, bode-nyquist, in matlab-symulink, taking into consideration the indices responses. the program dynamic 1 is organized on several stages:

computation of the transfer functions,
computation of the transfer function poles and the time constants,
computation of the module $h_{zc}(j\omega)$, the argument $\varphi^0_{HZC}(j\omega)$, the real part R_{eZC} and the imaginary one I_{mZC}.

for plotting the frequency characteristics and the transfer place (response in frequencies) as well as for computing the variation as response to the step signal (response in indices), it was used the baistrow method of the polynomial equation (8, 9) roots, attached to the transfer function.

the computations are effected for two peculiar cases: a translation module with differential cylinder d50/32-800 and a laboratory translation module, with cylinder having symmetric rods d50/32-200 and the following parameters: pressure p_0 = 5.5...10 mpa and the flow q_0 = 2.5...20 l/min (fig. 3) for the nominal current of the servo valve ehsdv-2t-7.5 (δi_c = 10 ma). in table 2 there are presented the computing parameters for both the translation modules taken into account. applying the method presented in [2, 3, 4], for the parameters shown in tab. 2 and using the program dynamic 1 it was possible to plot the characteristics frequency amplitude (fig. 4.a, and 4.b), the transfer function hodograph (fig. 5.a and 5.b) and the response function to the step signal (fig. 6.a and 6.b).

from the frequency characteristics (fig. 4) and index responses (fig. 4, fig. 5) established in the conditions of variation and character of the load, the following conclusions can be obtained:

the running frequencies are between the normal limits till in the immediate vicinity of the resonance frequency (ω = 10...20 hz), but under the limit of the ehsdv dominant frequency (fig. 4);

the resonance frequency increases with the increase of the piston diameter, for the same load and working pressure (fig. 4);

the oscillation amplitude increases with the increase of pressure and load, especially for elastic systems (fig. 6);

the resonance frequency is variable in large limits with the modifications of the load, the pressure having an negligible effect (fig. 4);

the transfer function is characteristic for a III order system (fig. 5), with the weight of the running frequencies in the quadrants II and III;

7. The dynamic behavior of the system ehsdv-cylinder-load

In the computing program mathlab-dynamic there have been obtained the following parameters: amplitude-phase-frequency for the transfer function of the frequency-displacement positioning system h_{zc} (10, 13), frequency-pressure h_{pm} (11, 14), and frequency-flow (12,15) for the ideal and real system, in closed and open loop. in the program the frequencies are modified between i = 1...200, respective for ω = 1...400 hz; for each frequency are computed the amplitude and the phase difference. the plot in frequency characteristics requires the presentation of data in normalized system (the rate to normal parameters) and to express them in db.

Considering the geometric and running parameters given in tab. 2 for both experimental modules (fig. 2) and taking into account the mathematical model developed in cap. 4, 5and 6 there have been obtained the characteristics amplitude-phase-frequency for the piston stroke z_c (in open and closed loop), the pressures p_{ma} and p_{mb} and the flows q_{ma} and q_{mb}. for the same parameters have been obtained also the stability conditions.

tab. 2

Geometrical and running parameters		Translation module with symmetrical cylinder		Translation module with asymmetrical cylinder	
EHSDV	Spool valve sroke Ys [mm]	0.54	0.67	0.84	1.5
	Grad acoperire Y_{0i}	-	-	-	-
parameters	Pressure p_0 [MPa]	5.5	7.0	10.0	10.0
	Flow Q_0 cm^3/sec	140	200	300	1000
	Great area S_{MA}	11.52		19.63	

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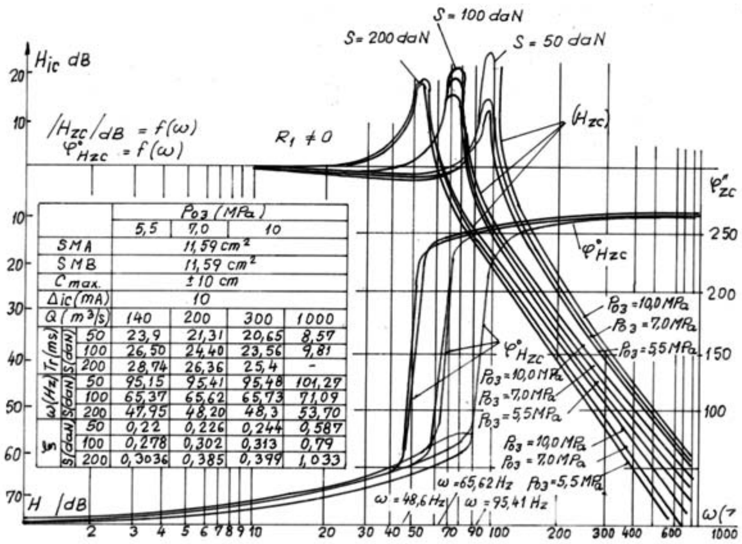
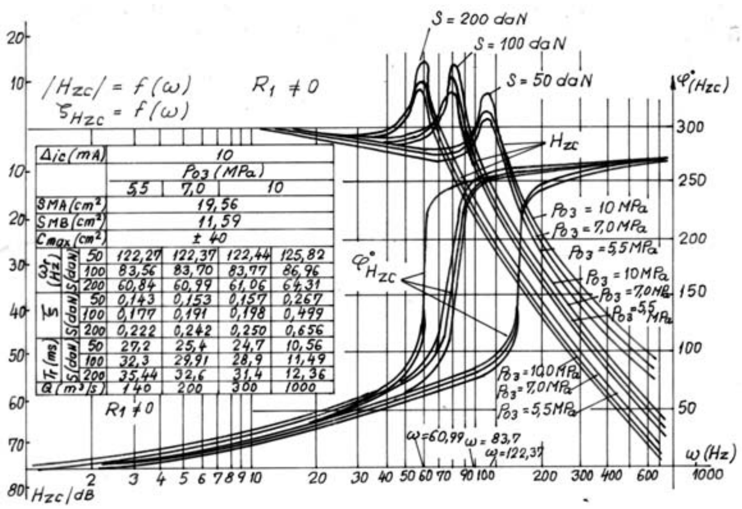
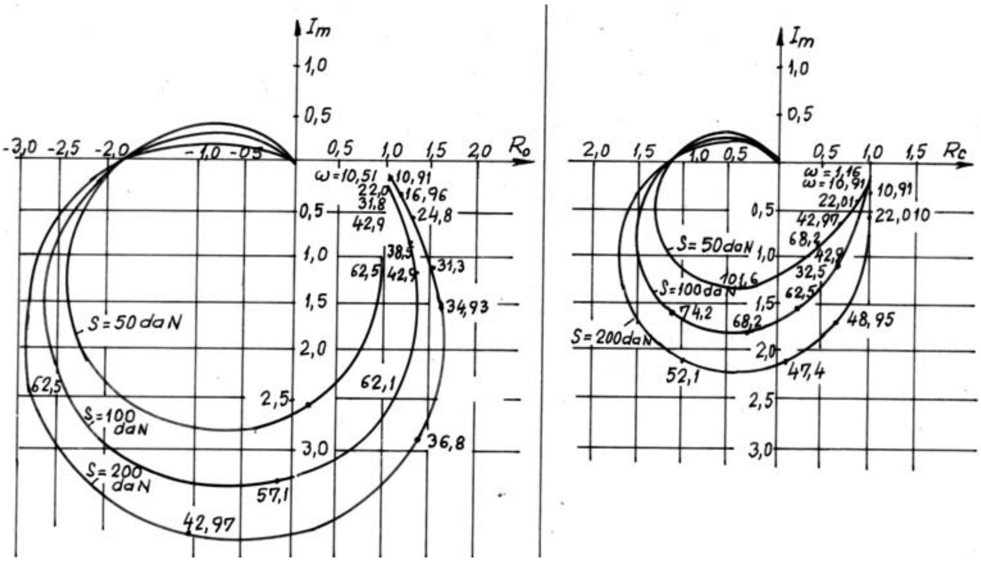
	[cm^2]; D = 50 mm		
	Small area S_{MB}[cm^2]; d = 32 mm	11.52	11.52
Hydraulic	Piston stroke C [mm]	20	80
cylinder	Elastic const. R_1 [daN/cm]	10	0
parameters	Friction coef. F_f [daNs/cm]	5 ; 7 ; 10	
	Piston velocity V_p [m/sec]	0.1207 ... 0.86	0.072 ... 0.50
	Load S [daN]	50.0	100.0 ; 200.0



a) laboratory translation model (ltm)

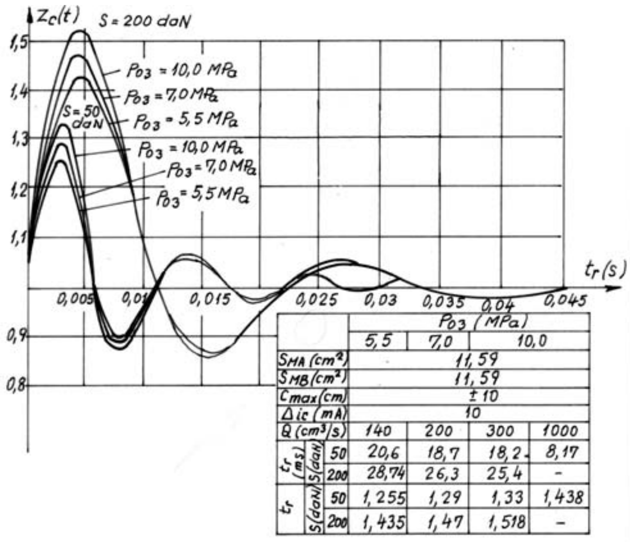


b) module with differential cylinders (mdc)
fig. 4. amplitude-phase-frequency characteristic



a)LTM b) MDF

fig. 5. Transfer function hodograph



			Po3 (MPa)			
			5,5	7,0	10,0	
SMA (cm²)			11,59			
SMB (cm²)			11,59			
Cmax (cm)			± 10			
Δic (mA)			10			
Q (cm³/s)			140	200	300	1000
tr (ms)	S (daN)	50	20,6	18,7	18,2	8,17
		200	28,74	26,3	25,4	-
tr	S (daN)	50	1,255	1,29	1,33	1,438
		200	1,435	1,47	1,518	-

A) LTM

8. Conclusions regarding the behavior of the electro-hydraulic positioning systems ehsdv-cylinder-load

Analyzing the fundamental equilibrium equations of the automatic hydraulic systems and taking into account also the synthesis presented in [1, 3, 4,10,11,12] the following results have been obtained:

it was determined the transfer function equation, for the considered system (a system with delay of iii order), in the normalized form (1, 8, 9);

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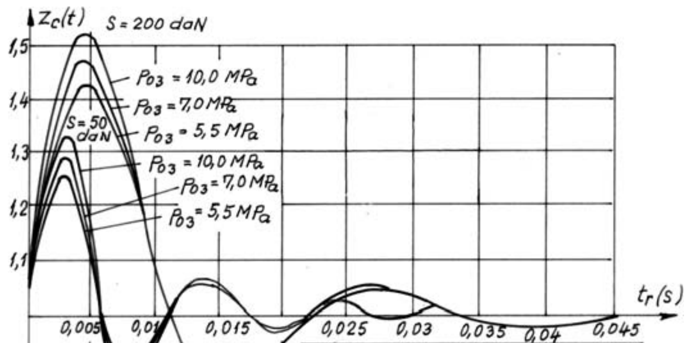

			Po3 (MPa)			
			5,5	7,0	10,0	
SMA (cm²)			11,59			
SMB (cm²)			11,59			
Cmax (cm)			± 10			
Δic (mA)			10			
Q (cm³/s)			140	200	300	1000
tr (ms)	S (daN)	50	20,6	18,7	18,2	8,17
		200	28,74	26,3	25,4	-
tr	S (daN)	50	1,255	1,29	1,33	1,438
		200	1,435	1,47	1,518	-

B) MDF
fig. 6. The characteristic zc(t) = f(t)

it was solved the transfer function obtained for the geometrical and running parameters for the two translation modules analyzed and were determined the characteristics of frequency and response to the step signal, putting into evidence the influence on the stability degree of the following parameters: the pressure p_0 and the value of the load s taking into account his character (elastic or rigid load);

it was determined the mathematical model for the assembly ehsdv-cylinder-load for piston displacement, pressures and flows in the load cylinder for the systems with ideal and real ehsdv. the obtained characteristics put into evidence the usual frequencies as a function of the dominant frequency of ehsdv (ω_{-3db});

in order to an operative use of the mathematical model for analyzes and syntheses of the system ehsdv-cylinder-load in every concrete situation it was worked out a package of computing programs dynamic 1 and dynamic system.

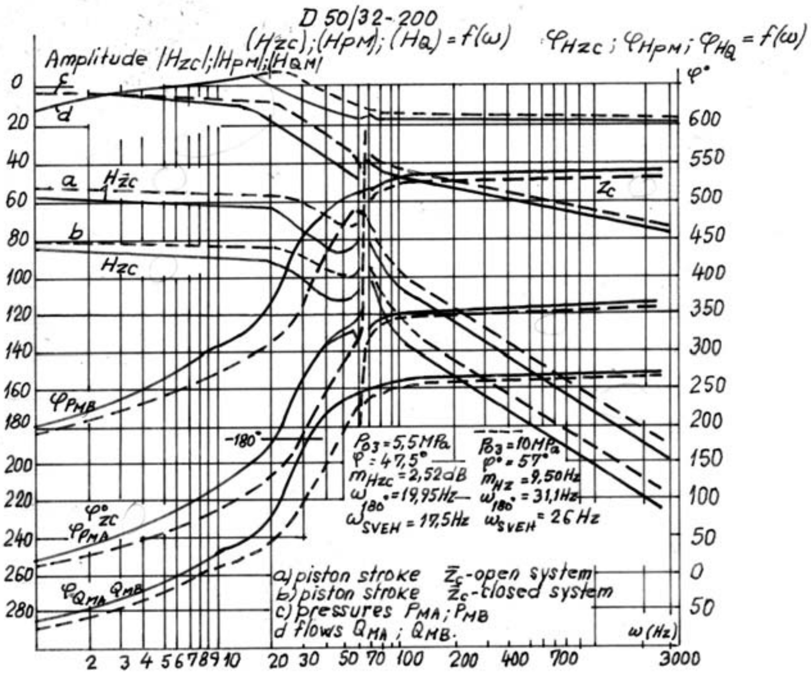

fig. 7a. The frequency characteristics of the system ehsdv-cylinder-load (MLT)

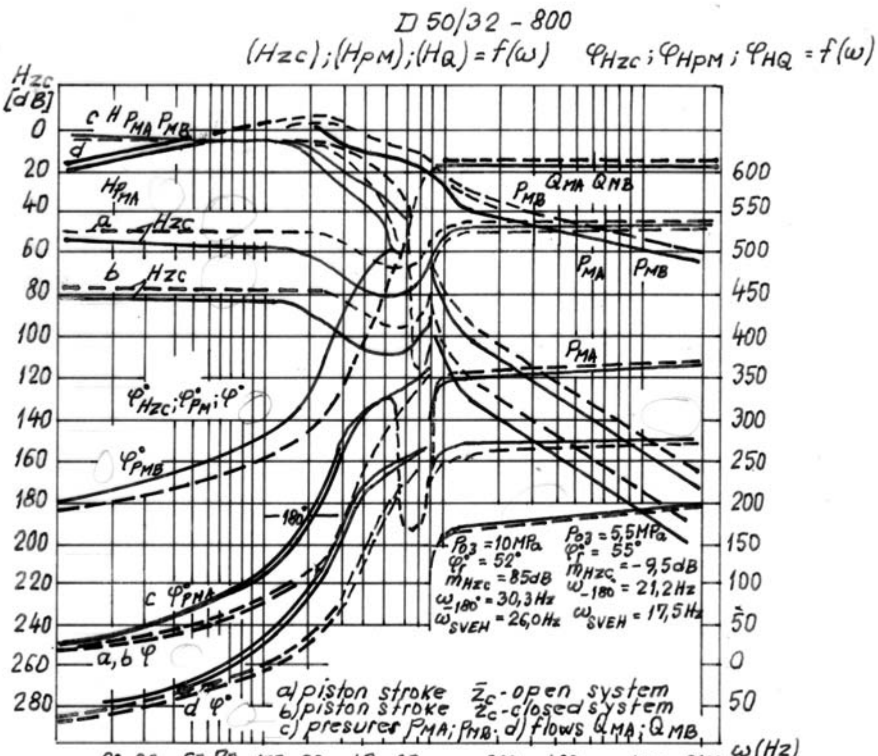

fig. 7b. The frequency characteristics of the system ehsdv-cylinder-load (MDF)

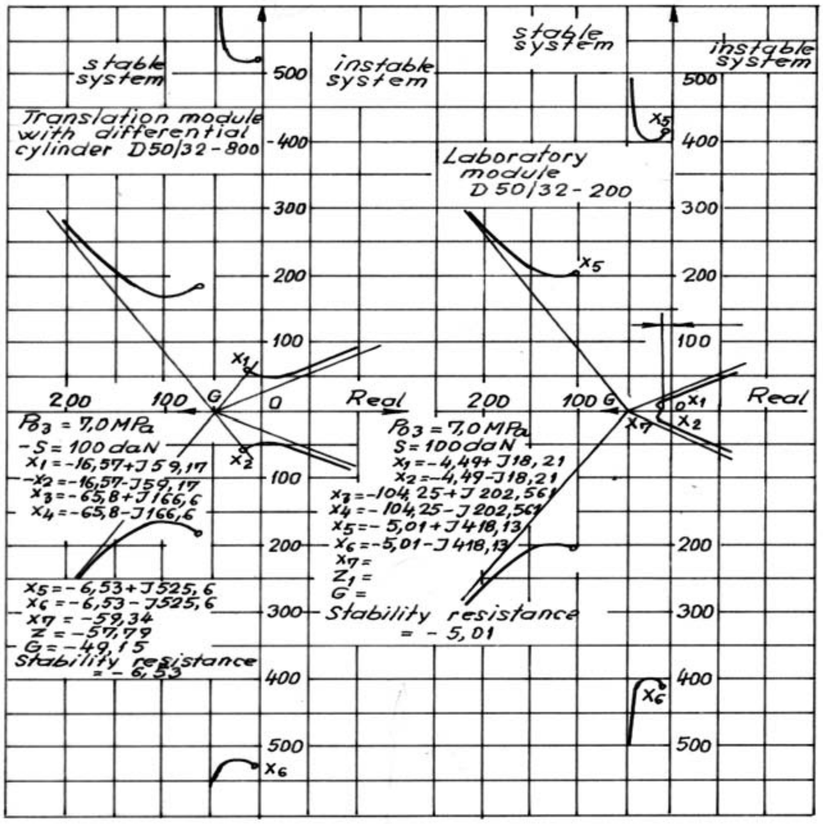

fig. 8. The place for the transfer function roots

The proposed method for the analysis of the frequency and step signal put into evidence the influence of the geometric and running parameters upon the stable dynamic running conditions for the analyzed translation modules.

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MATHEMATICAL MODEL ANALYSIS ON HYDRAULIC ENERGY DISSIPATION DEVICES

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Abstract: This paper proposes a mathematical model for hydraulic seismic energy-dissipation devices. The model allows dynamic behavior analysis on device dimensions, pressure controller influence and characteristics of hydraulic fluid. A case study was performed to analyze influence of pressure controller rigidity on dynamic characteristics of hydraulic device.

Keywords: energy, earthquake, dissipation, hydraulic, model.

1. Introduction

A hydraulic energy dissipation device can be used for earthquake-induced energy dissipation in a certain structure required by inertial forces applied to structure. The device is achieved as a hydraulic cylinder (1) with two chambers separated by a piston (2), filled with a viscous medium (synthetic oil, silicone oil, suspended particles, grease, suspended solids, etc.). The piston head is in solidarity with a piston rod (3) and a compensation tube crossing the two chambers on both sides of the piston. A third chamber (4) is located inside the compensation tube and is designed to compensate expansion or contraction of viscous fluid environment under viscous thermal effect. The piston rod is connected to the construction body and the building foundation by means of planar or spatial joints (5). The reversible work drive alternative traction-compression movements and dynamic behavior depends on the frequency (velocity) of instantly produced earthquake excitation, mechanical shock or vibration. (Figure 1). This is achieved by a system of pressure control devices (energy regulator) (7, 8) made for various ranges of dissipation, devices placed on external circuit (6).

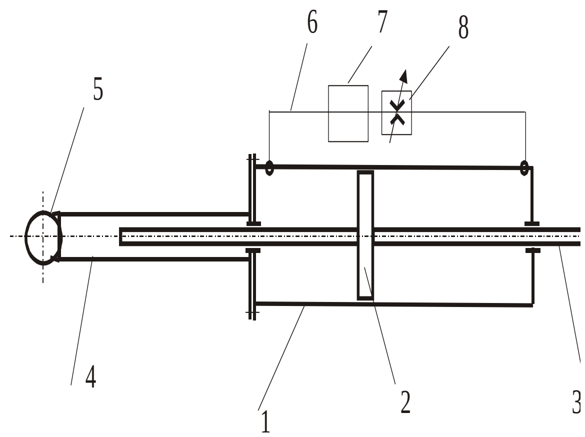

Figure 1. *Hydraulic device principle*

2. Necessary characteristic of the hydraulic dissipation device

Necessary characteristic of the hydraulic energy dissipation device is considered as resistant force variation law according to displacement of piston rod. The theoretical characteristics of strong energy dissipation force, shown in figure 2, presents real amount of dissipated energy by hydraulic dissipation device, at complete displacement according to maximum piston stroke.

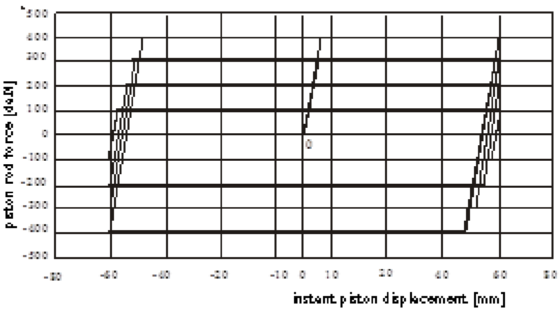

Figure 2. *Theoretical characteristic of hydraulic device*

3. Mathematical model of the dissipation device

To achieve dynamic behavior analysis according to various constructive and functional device dimensions is useful for implementation of a mathematical model. The mathematical model of an energy dissipation device result from the equation of hydraulic flow rate necessary for the piston translation and from the equation of forces applied to the piston. Nonlinear mathematical model is described by the equations:

$$\begin{aligned} \frac{dy}{dt} &= \frac{D}{A}\sqrt{p^3} - \frac{H}{A}\sqrt{p} + \frac{V_o}{A.E}\frac{dp}{dt} \\ \frac{d^2y}{dt^2} &+ \frac{C}{M}\frac{dy}{dt} + \frac{K}{M}y = \frac{g}{10} - \frac{A}{M}p \end{aligned} \quad ; \qquad (1)$$

where:

$$D = \frac{\pi^2.d^3}{4.k_r}\sqrt{\frac{2}{\xi.\rho}}; \quad H = \pi.d.\delta\sqrt{\frac{2}{\xi.\rho}}; \quad (2)$$

represent hydraulic device constant,

d - pressure controller slide diameter; δ – passive stroke for opening the pressure controller;
k_r - pressure controller rigidity;
ξ - local loss coefficient of hydraulic fluid load circulation by pressure regulator;
$a = \pi d^2/4$ – front surface area of the pressure regulator;
K - hydraulic device stiffness;
C – dissipation device damping factor;
M - suspended mass on hydraulic dissipation device;
A - frontal surface area of piston head;
V_0 - volume of hydraulic fluid between the piston and pressure regulator;
$\rho; E$ – density, elasticity of the hydraulic fluid;

Model variables are: y - piston momentary displacement; p- momentary set pressure. Are required to be studied in principle the following variables:

$$\begin{aligned} &y = y(t); p = p(t); p = p(y); \\ &F = F(t) = A \cdot p(t); F = F(y); \end{aligned} \qquad (3)$$

The mathematical model is achieved to assumptions:

- has not been taken account of flow rate losses through the gaps, as the piston seals not allow flow rate losses, being self-strain able on cylinder bore;
- flow rate losses on pressure controller, between locations of high and low pressure are considered negligible given the small size of regulator compared with the dissipation device cylinder.

$$d/D_d \leq 10/150 = 1/15 \qquad (4)$$

where:
d - pressure controller slide diameter ;
D_d - cylinder bore diameter;
- friction forces are considered negligible in the hydraulic pressure controller, compared with earthquake-induced forces directly applicable.

4. Results of dynamic modeling

A case study was achieved for an assembly energy dissipation hydraulic device- pressure controller, with the following structural, functional characteristics and hydraulic fluid used:
$k_r = (52;104;208)daN/cm$;
$d = 0{,}45cm; \rho = 0{,}0009\, kg/cm^3; \xi = 1{,}8;$
A=115,4 cm2; Vo=3460 cm3;
$A = 115{,}4cm^2; V_0 = 3460cm^3; M = 25000kg$
$E = 16900\, daN/cm^2; g = 981\, cm/s^2; \delta = 0{,}9;$ $K = 8\, daN/cm; C = 10\, daNs/cm.$
Modeling results were achieved for three different values of pressure controller spring stiffness, (k_r) and are shown in graphical form in figures 3, 4, 5, 6 and 7.

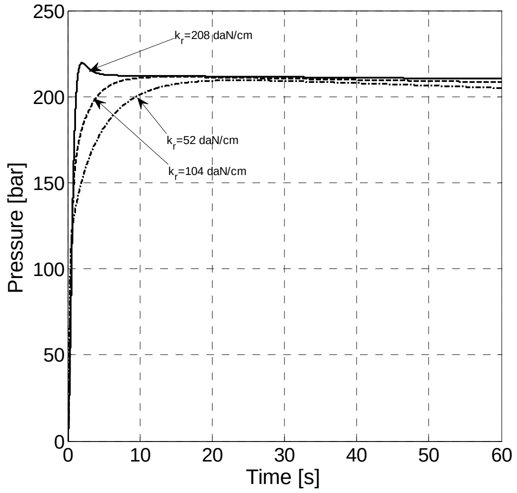

Figure 3. *Pressure variation over time, p = f (t)*

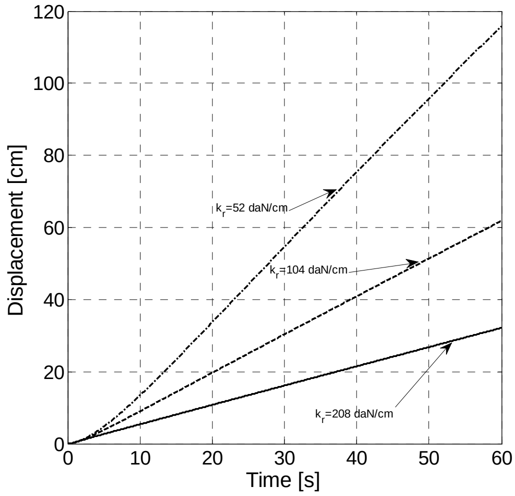

Figure 4. *Displacement variation , y =f(t)*

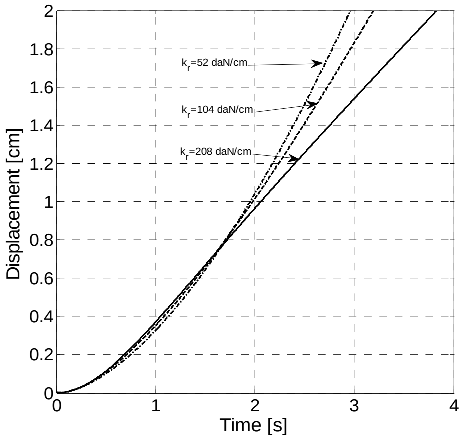

Figure 5. *Detail view on displacement variation*

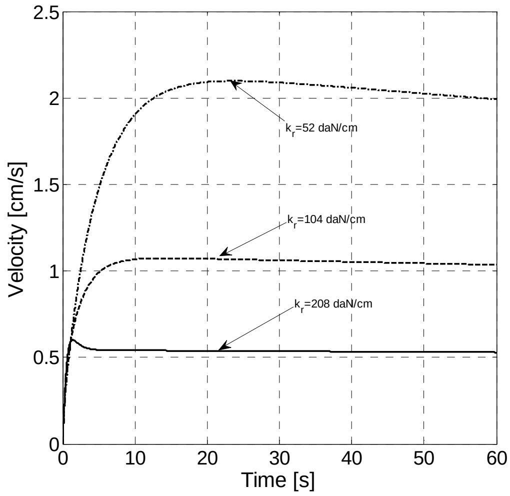

Figure 6. *Velocity variation*

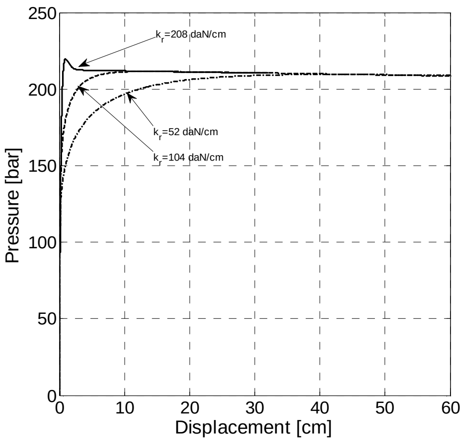

Figure 7. *Pressure variation according to piston displacement- p=f(y)*

5. Conclusion

A hydraulic energy dissipation device turn accelerated movements of the seismic excited structure base, to a uniform motion, as this device is connected to desired seismic isolated structure. Thus the earthquake induced inertial forces on structure become zero or very low for most cases.

This can be possible when undertaken structure force, depending on the base movement, is taking the form of figure 2, for alternative motion induced by earthquakes shock or vibrations.

For these general operation conditions of a hydraulic seismic energy dissipation device , from dynamic modeling process, following conclusions are appropriate:

1. Pressure variation adjusted by controller should quickly settle at constant value for lower stroke values (fig.7), which requires high stiffness for pressure controller spring.
2. Adjusted pressure by device controller should stabilize in short time and base displacement should be uniform.
3. For high stiffness values for pressure controller spring need to achieve constant and low velocity values for earthquake excited structure base (fig. 6);
4. The steady pressure and force values, provided by hydraulic energy dissipation device, opposite to excited seismic base movement, depends only on the structure mass M and less influenced by other physical values shown in the model**.**

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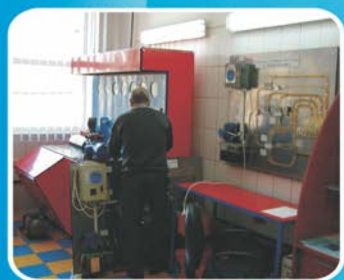

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THE STATIC AND DYNAMIC ANALYSIS OF A HYDRAULIC 3/2 VALVE WITH LINEAR DISPLACEMENT. PRESSURE-FLOW CURVES

KOZMA, C.

Abstract:
It is considered a 3/2 hydraulic flow control valve to controll a hydraulic cylinder with an A+E combination of half-bridges. Continuity equations and orifice equations are written. Based on the identified and applied relations the load flow equation is delineated for the functionality domain. It is given the graph of this function revealing the pressure-flow curves. The importance of these curves is high for the stability and dynamic analysis of the valve. The steps done in this research are explicit and detailed to facilitate researcher's work in obtaining an understanding of the concepts, acquiring the strategic steps and applying the calculus method for similar structures. Beside the chosen notations there are utilised the terms that appear in the speciality literature to highlight the similarity between the steps. The transparency of this study must respond to the research of the parameters that modify in the case of a hydraulic rotary displacement valve.

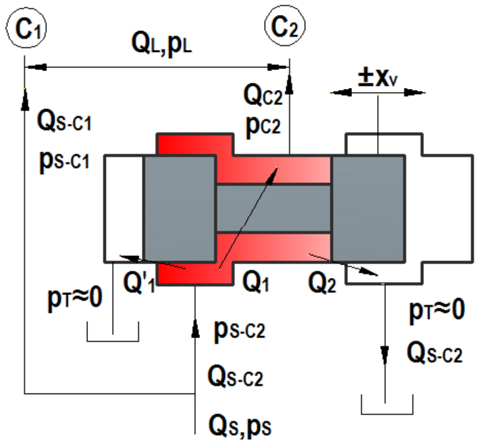
Key words: *hydraulic valve, three ways, pressure-flow curves, steady-state characteristics, valve coefficients*

Introduction

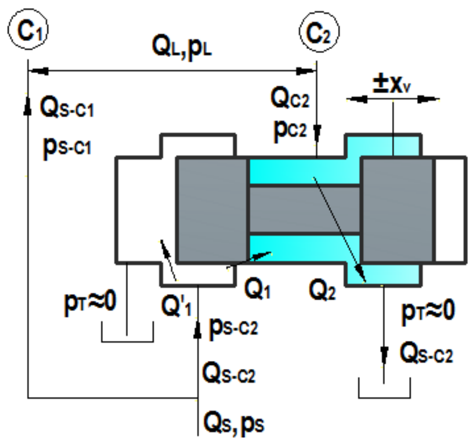
Hydraulic distributors are, from the point of view of automatic systems theory, amplifiers, so that they have an essential role in the precision assurance and the stability of the hydraulic automatic systems.

The analysis of the energetic characteristics and the losses of these elements form a fundamental in developing realistic mathematical models. [1] Valves with three ways are used in positioning hydromechanical servo systems.[2]

2. The considered hydraulic linear 3/2 valve - LHD



a)



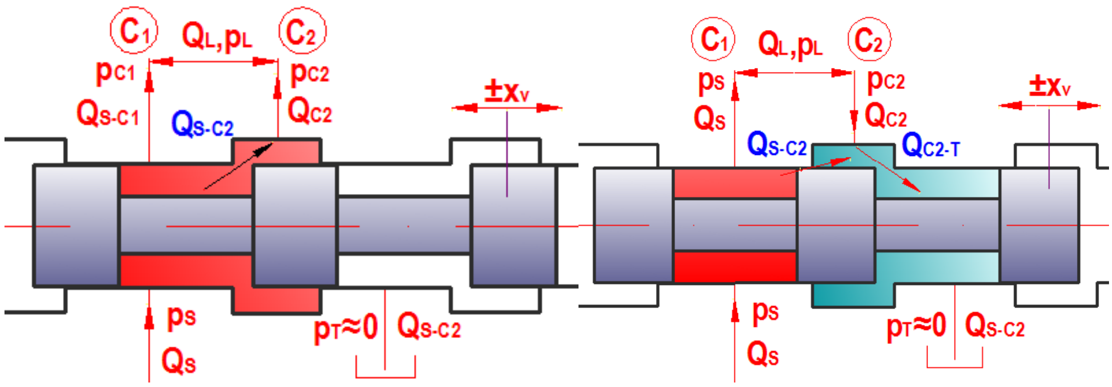
b)

Figure1. The hydraulic three way valve (LHD):
a) Supply: the connection with the source; b) Discharge: the connection with the tank.

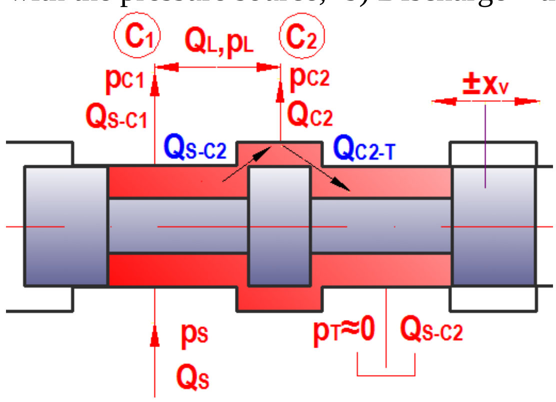
It is taken for analysis the hydraulic system from Figure 1, following the steps from the speciality literature [2], [3], [4]. The system consists, in principle, of a cylindrical slide valve with two lands, three ways and two working positions (3/2). The two spool lands form with their edges, hydraulic resistances in the case of the supply, Figure 1 a), and in the case of the discharge, Figure1 b). The directional control valve is also a flow control valve for this system because the hydraulic formed resistances are variable and adjustable. The flow that passes through the hydraulic linear distributor (LHD) controls only a chamber of the hydraulic cylinder, C2. The motor control is made by an A+E half-bridges combination. In other words, a chamber of the hydraulic cylinder, C1, is controlled by an E half-bridge, and the other chamber, C2, is controlled by an A half-bridge. This type of command must be used with a piston that has different areas to assure the reverse of the motor piston. [2], [1].

In the case of the E half-bridge, the supply pressure acts on the smaller piston area. The resistance type that E half-bridge represents is a fix hydraulic resistance so that the force developed on the piston surface is constant for any velocity if the supply flow of the chamber C1 is constant. [5]

The A half-bridge consists in two hydraulic resistances mechanically or electrically correlated so that at one command displacement applied to the valve, these resistances vary equally and oppositely. It records a rise from one resistance (it closes) and a descend from the other resistance (it opens). At the A half-bridge, having two control edges, both resistances are adjustable. [5]



a) Supply – the connection with the pressure source; b) Discharge – the connection with the tank;



c) The connection of the all three ways – the valve middle position

Figure2. The hydraulic three way valve.

The two orifices pertain to the A half-bridge and are comprised in one chamber. Hence, it can not be discussed the matched and symmetrical properties in the absence of at least one more valve chamber. The arrows from Figure 1 and Figure 2 indicate the assumed flow direction.

It is prefered critical center valve and in consequence only one orifice is active at one time, 1 or 2. The critical center valve is classsified as having either ideal or practical geometry. Ideal geometry implies that the orifice edges are perfectly square with no rounding and that there is no radial clearance between the spool and sleeve. Although these geometrical perfections are not possible in practice, it is possible to construct a valve with a relatively linear flow gain near null position. A critical center valve with practical geometry is a valve with radial clearance.

2. The analysis of LHD. The flow equations: the continuity and orifice equations, area calculi, the pressure drops and the load flow

The two piston areas must be designed so that the control static pressure acting on the piston head respect the following relation:

$$p_{C20} = \frac{p_S}{2} \tag{1}$$

Where:
- p_{C20} is the load pressure – the sent pressure through the valve to the motor chamber C2 to control it– at the null operating point;
- p_S is the supply pressure, from the source.

The control relation (1) allows the control pressure p_{C2} to rise and fall and to provide equal acceleration and deceleration. With no loads on the piston, (1) is satisfied if between the head area A_h, and the rod area A_r exists the relation:

$$A_h = 2A_r \tag{2}$$

The expression (2) is a rule generally used for piston sizing even with load forces. However, the areas should be sized to satisy (1) if there are unidirectional force load components. [2]

In the first stage it is analysed the steady-state characteristics of the LHD, hence the compressibility flows are zero [2]. The continuity equations for the valve chamber are:

$$Q_L = Q_{S-C1} + Q_{C2} = Q_{S-C1} + Q_1 \tag{3}$$

$$Q_L = Q_{C2} - Q_{S-C1} = Q_2 - Q_{S-C1} \tag{4}$$

$$Q_S = Q_{S-C1} + Q_{S-C2} \tag{5}$$

Where:
- Q_L represents the sent flow to the CH through the A+E half-bridges combination;
- Q_{C2} represents the flow through the supply/discharge line of the chamber C2;
- Q_S is the supply flow sent by the hydraulic energy source to the CH;
- Q_1 represents the rate of flow that passes from the supply source through the first created resistance in the valve;
- Q_2 represents the rate of flow that passes from the chamber C2 through the second created resistance;
- $Q_{S\text{-}C1}$ and $Q_{S\text{-}C2}$ are the supply flows sent to the CH from the energy source through the supply lines.

The rate of flow Q'_1 corresponds to the valve leakages that happen in the supply stage. Since the valve is considered to have ideal geometry, the flow Q'_1 is zero. According [2], leakages through a valve with ideal geometry are null. It is assumed that the valve has only two working positions and a supply is inherited by a complete obturation of the same orifice/way/connection, and vice versa for the other orifice/connection. Neglecting the other hydraulic resistances, the flow sent to the motor is the rate of flow that crosses the valve, the LHD.

Being a valve with a critical center – with practical geometry – it is in many respects an optimum valve because leakage flows are minimum, flow gain is linear, and design procedures are usually besed on such a valve.

A dynamic analysis demands inclusion of compressibility flows which depend on the valve chamber volumes. Such an analysis can be easier obtained if it is considered the subsystem valve-motor since lines and motor contribute semnificatively to hydraulic capacity [6], [2].

The load flow of the actuated element is notated with Q_L and defined by (3) and (4). The pressure load is p_L:

$$p_L = p_{S-C1} - p_{C2} \tag{6}$$

The flows Q_1 and Q_2 passing through the supply orifice and discharge orifice are described with the orifice equation:

- for x_v>=0:

$$Q_{C2} = Q_1 = C_d A_1 \sqrt{\frac{2}{\rho}\left(p_{S-C2} - p_{C2}\right)} \tag{7}$$

- for x_v<=0:

$$Q_{C2} = -Q_2 = -C_d A_2 \sqrt{\frac{2}{\rho} \cdot p_{C2}} \tag{8}$$

Where:

- A_1 and A_2 are the orifice areas; they depend on valve geometry and are functions of valve displacement in the limit of the initial conditions – LHD has two working positions.

To simplify the analysis there are chosen the points that measure the pressure difference more conveniently. In consequence (7) modifies:

$$Q_{C2} = Q_1 = C_d A_1 \sqrt{\frac{2}{\rho}\left(p_S - p_{C2}\right)} \tag{9}$$

In speciality literature, [2], [3], Q_{C2} appears under the Q_L notation and thereby (7) and (8) can be easier recognised as the important pressure-flow curves.

Being the orifices that compose the A half-bridge, the values of the areas A_1 and A_2 vary symmetrically. Because the sliding body of the control valve is cylindrical, it can be written the area calculation formula (10) from [4] where the hydraulic rezistance appears. The following relation is used forward in the valve evaluation.

$$A_1\left(x_v\right) = A_2\left(-x_v\right) = \pi D x_v \tag{10}$$

Where:

- D is the spool diameter;
- x_v is the command displacement of the spool.

The definition relation (10) of the orifice area differs from one researcher to another, [2]. Thus it is the product between the valve displacement and the area gradient of w:

$$A_1\left(x_v\right) = A_2\left(-x_v\right) = w x_v \tag{11}$$

The area gradient has the measure unit [m] according [3]. The passing section area through the orifice related to the variation of the valve displacement defines the area gradient (in respect to the valve critical center):

$$w = \frac{dA}{dx_v} = \frac{A}{x_v} = \frac{\pi D x_v}{x_v} = \pi D \tag{12}$$

Where:

- A is the passing section area through the active orifice.

In [2] the opening or area gradient is defined as the rate of change of orifice area with stroke. Also, w represents the width of the slot in the sleeve and is the most important parameter for a linear valve.

For a plane passing area the speciality literature offers in the theoretical base the calculus expressions for different geometry sections.

When it is considered the valve neutral position it rezults three working positions and the orifice areas are equal:

$$A_1(0) = A_2(0) = \pi D x_0 \tag{13}$$

Where:

- x_0 is the value of the type of lap (overlap, underlap, zero or critical lap) – the measure of the orifice opening – in the neutral position or in other words in the absence of the x_v command displacement.

It is sufficient to define only one orifice area because the other area follows from it. In [2] it is said that if the orifice areas are linear with valve stroke, as is usually the case, only one defining parameter is required and that is w.

With no load, the pressure inside the chamber valve from Figure1 is equal to the supply pressure of the chamber C2:

$$p_{S-C1} = p_{C2} \tag{14}$$

Relation (14) forward express that the motor is not in equilibrium at this equality for the fact that it is about a differential cylinder. To have no load pressure (p_L=0) the pressure in chamber C2 has to be half of the supply pressure ($p_{S\text{-}C1}$*1/2). As considering, for simplification, the reference pressure p_S, it is obtained:

$$p_{C2} = \frac{p_S}{2} \tag{15}$$

The cylinder piston remains in this way in the equilibrium position, the middle position and hereinafter the reference position. Consequently, to have indeed no load pressure, (6) changes in:

$$p_L = p_S - 2p_{C2} \tag{16}$$

The helpfulness of the relation (16) is observed when defining the valve null operating point. By changing the valve displacement x_v, the pressure in C2 can be controlled and implicit the force acting on the piston surface (load pressure). While the chamber C2 iteratively records an increase and a decrease in the command pressure, the chamber C1 is permanently supplied with p_s.

It is determined that a control of the motor with an A+E half-bridges combination yelds to a load flow as a function of valve position and load pressure (p_L):

$$Q_L = Q_L(x_v, p_L) \tag{17}$$

The relation (17) can be found theoretically, but the solution for the general case is tedious because the algebraic equations involved are nonlinear. In most cases, this is no serious obstacle because valves are never so complex as to have all orifice areas differently described.

The plot of (17) is achieved following several successive steps. Thus, suggesting from [2], it is selected a value for the command displacement x_v on the base of it the orifice areas are calculated. In the other step it is chosen a series of values for Q_L to solve for each Q_L,the pressures p_1 and p_2 from the continuity equations. The cycle repeats leading to the obtaining of the graph [2].

The plot of (17) is known as the pressure-flow curves and is a complete description of steady-state valve performance. All of the performance parameters, such as valve coefficients, can be obtained from these curves. [2]

3. The linearized analysis of the valve. Valve coefficients

In making a dynamic analysis it is neccesary that the nonlinear algebraic equations which describe the pressure-flow curves be linearized. The general expression for the load flow can be express as a Taylor's series about a particular operating point Q_L=Q_{L1}. [2] Accordingly:

$$Q_L = Q_L(x_v, p_L) \tag{18}$$

$$Q_L = Q_{L1} + \left.\frac{\partial Q_L}{\partial x_v}\right|_1 \Delta x_v + \left.\frac{\partial Q_L}{\partial p_L}\right|_1 \Delta p_L + ... \tag{19}$$

$$Q_L - Q_{L1} \equiv \Delta Q_L = \left.\frac{\partial Q_L}{\partial x_v}\right|_1 \Delta x_v + \left.\frac{\partial Q_L}{\partial p_L}\right|_1 \Delta p_L \tag{20}$$

It is neglected in (20) the higher order infinitesimal terms. The partial derivatives required are obtained analytically by the derive (differentiation) of the steady-state characteristic – the equation for the pressure-flow curves - or graphically from a plot of the curves. The valve coefficients are obtained exactly from this partial derivatives and are useful for the dynamic analysis of the system. [1], [2]

The partials of the equation that plots the pressure-flow curves define the two most important parameters for a valve: the flow gain and the flow-pressure coefficient. The flow gain is the most important valve coefficient and directly affects the open loop gain constant in a system, thus having a direct influence on system stability. It is defined as:

$$K_Q \equiv \frac{\partial Q_L}{\partial x_v} \tag{21}$$

The pressure-flow coefficient K_{Qp} has the defining expression (22). This coefficient is always positive because the defining ratio is negative for all type of valve configuration. The flow-pressure coefficient directly affects the damping ration of valve-motor combinations.

$$K_{Qp} \equiv -\frac{\partial Q_L}{\partial p_L} \tag{22}$$

In the dynamic analysis of the control valve it is used another useful quantity: the pressure sensitivity or pressure-displacement coefficient, K_p. The pressure sensitivity is relatively large and accounts for the ability of valve-motor combinations to breakaway large friction loads with little error. It is defined as:

$$K_p \equiv \frac{\partial p_L}{\partial x_v} \tag{23}$$

$$K_p = \frac{K_Q}{K_{Qp}} \tag{24}$$

Having these definitions, the linearized equation of the pressure-flow curves that applies to all valves, becomes:

$$\Delta Q_L = K_Q \Delta x_v - K_{Qp} \Delta p_L \tag{25}$$

4. The valve coefficients of LHD at the operating point

The coefficients K_Q, K_{Qp} and K_p are called valve coefficients and are extremely important in determining stability, frequency response, and other dynamic characteristics. The values of these coefficients vary with the operating point. The most important operating point is the origin of the pressure-flow curves (for exemple, where Q_L=p_L=x_v=0) because system operation usually occurs near this region, the valve flow gain is largest and the flow-pressure coefficient is smallest. Large flow gain gives high system gain. Small flow-pressure coefficient gives low damping ratio. The operating point is the most critical from a stability viewpoint. A system stable at the operating point is usually stable at all operating points. The valve coefficient evaluated at the operating point are called the null valve coefficients. [2]

The operating point for the analysed valve (LHD) from Figure1 is defined at:

$$Q_{C2} = Q_L = x_v = 0 \tag{26}$$

$$p_{C2} = \frac{p_S}{2} \tag{27}$$

The partial derivatives of (25) are evaluated at the operating point established by (26) and (27). In the defining relations of the valve coefficients, (21), (22), (23), there are replaced (26) and (27), [2], [7]. Therefore, the null flow gain is:

$$K_{Q0} = \left.\frac{\partial Q_{C2}}{\partial x_v}\right|_0 = \frac{\partial\left(C_d w x_v \sqrt{\dfrac{p_S}{\rho}}\right)}{\partial x_v} = C_d w \sqrt{\frac{p_S}{\rho}} \tag{28}$$

Where:

$$Q_{C2} = C_d w x_v \sqrt{\frac{2}{\rho}\left(p_S - p_{C2}\right)} = C_d w x_v \sqrt{\frac{2}{\rho}\left(p_S - \frac{p_s}{2}\right)} = C_d w x_v \sqrt{\frac{p_S}{\rho}} \tag{39}$$

$$Q_{C2} = C_d w x_v \sqrt{\frac{2}{\rho} \cdot p_{C2}} = C_d w x_v \sqrt{\frac{p_S}{\rho}} \tag{30}$$

The computed null flow gain has been amply verified by test of practical critical center valves and may be used with confidence. The null flow gain can be easily computed and controlled. The system stability depends on this quantity, null flow gain. This is one of the major reasons why hydraulic servo enjoy a reputation for dependable stability. [2] On the other way, the computed values for the null flow-pressure coefficient and the null pressure sensitivity coefficient are far from that obtained in tests of practical center valves. It is possible to compute more realistic values for these two null coefficients once the leakage characteristics for such valves have been investigated.

The computed null flow-pressure coefficient for LHD is:

$$K_{Qp0} = -\left.\frac{\partial Q_{C2}}{\partial p_{C2}}\right|_0 = -\frac{\partial\left(C_d w x_v \sqrt{\dfrac{2}{\rho} \cdot p_{C2}}\right)}{\partial p_{C2}} = -C_d w x_v \sqrt{\frac{2}{\rho}} \cdot \left(\sqrt{p_{C2}}\right)' = -\left.\frac{C_d w x_v}{\sqrt{p_S \cdot \rho}}\right|_{x_v=0} = 0 \tag{31}$$

While the flow-pressure coefficient is a function of the area gradient w, the pressure senitivity does not depend on w:

$$K_{p0} = \left.\frac{\partial p_{C2}}{\partial x_v}\right|_0 = \left.\frac{2\left(p_S - p_{C2}\right)}{x_v}\right|_{x_v=0} = \left.\frac{p_S}{x_v}\right|_{x_v=0} = \frac{K_{Q0}}{K_{Qp0}} = \frac{C_d w \sqrt{\dfrac{p_S}{\rho}}}{0} = \infty \tag{32}$$

Comparing the valve null coefficients it is observed that the flow gain for a linear three way critical center valve and the flow gain for a linear four way critical center valve are the same but the pressure sensitivity is half that of the four way valve. Therefore constant and friction load forces will cause twice the static error in systems with three-way valves. The most serious objection to the

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use of three way valves is that dynamic errors are also about doubled. Thus three way valves are best suited to hydromechanical servos which have little or no loads or can tolerate the error.

Since three way valves are usually used in hydromechanical servos, the source positioning the spool is mechanical and quite stiff compared with the force loads imposed by the spool. For this reason the force equation describing the spool motion is usually unimportant and flow forces are not of interest.

5. Steps to plot pressure-flow curves

There are distinguished several steps needed to plot the pressure-flow curves:

- the load flow (Q_{C2} is the flow sent to CH) expression is delineated or written for a positive valve displacement, $x_v>0$;
- the load flow expression is delineated or written for a negative valve displacement c, $x_v<0$;
- the load flow expressions are reunited to result a single equation integrating the valve performance for the entire working domain or valve displacement – x_v – domain.

6. The pressure-flow curves of LHD

It is regarded that the leakages are null for a valve with ideal geometry. There are used (7) and (8) and devided by x_{vmax}. Forward it is factors out p_s to obtain a function of x_v/x_{vmax} and p_{C2}/p_S (in speciality literature is p_L/p_S) :

- for $x_v>=0$:

$$\frac{x_v}{x_{v\max}} \cdot \sqrt{1-\frac{p_{C2}}{p_S}} = \frac{Q_{C2}}{C_d \cdot w \cdot x_{v\max} \cdot \sqrt{\frac{2p_S}{\rho}}} \tag{33}$$

- for $x_v<=0$:

$$\frac{x_v}{x_{v\max}} \cdot \sqrt{\frac{p_{C2}}{p_S}} = \frac{Q_{C2}}{C_d \cdot w \cdot x_{v\max} \cdot \sqrt{\frac{2p_S}{\rho}}} \tag{34}$$

Where:

- x_{vmax} the positive maximum value of the command displacement or valve displacement.

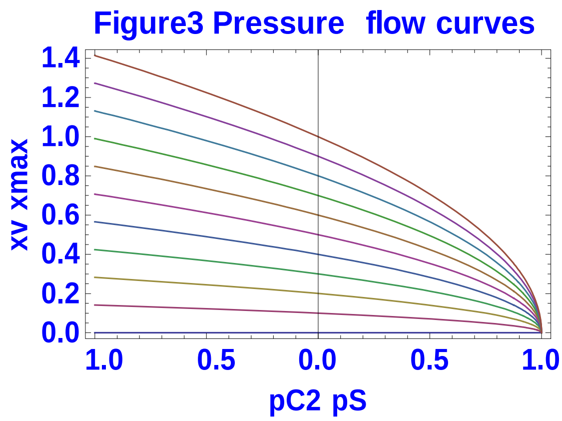

Figure3. The pressure-flow curves for a positive displacement of the three way valve.

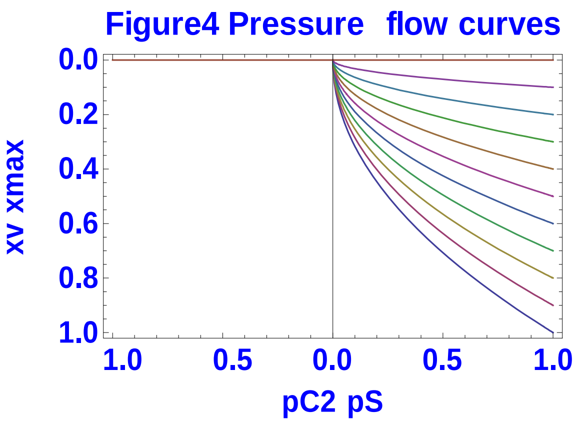

Figure4. The pressure-flow curves for a negative displacement of the three way valve.

Using (33) and (34) there are obtained the graphs for the positive displacement, Figure3, for the negative displacement, Figure4, and Figure5. For a total view and in order to a further comparison, it is revealed also the pressure-flow curves for a four way valve in Figure6. This plots

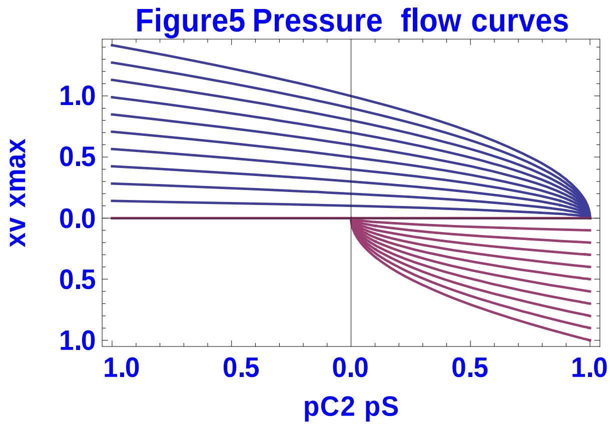

Figure5. Pressure-flow curves
for a three way valve.

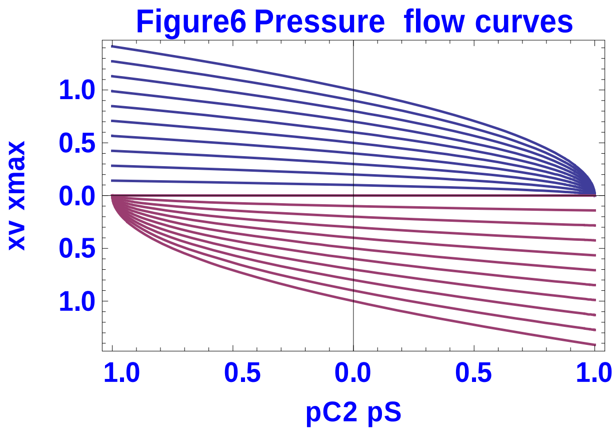

Figure6. Pressure-flow curves
for a four way valve.

describe the valve performance on the entire variation domain of the valve displacement and represent the pressure-flow curves of LHD.

Analysing the ration x_v/x_{vmax} it is determined that it varies in the domain [-1,1] likewise the ration p_{C2}/p_S.

The accomplishment of the plots was possible through the use of Mathematica, a sofware programme for engineering calculi. There were applied the functions Evaluate, Table, Sqrt and Plot. One of the written codes for plotting the flow-pressure curves is shown below, with the mention that it is similar with the others:

```
Plot[{Evaluate[Table[Evaluate[xv/10*Sqrt[(1-
pLpepS)]],{xv,0,10}]],Evaluate[Table[Evaluate[xv1/10*Sqrt[pLpepS]],{xv1,-10,0}]]},{pLpepS,-
1,1},Frame→True,FrameLabel→{"pc2/pS","xv/xvmax"},PlotLabel→Style["Figure4 Pressure-flow
curves",Blue,15],LabelStyle→Directive[Bold,Blue,FontSize→13,FontFamily→"Helvetica"] ]
```

Conclusion

The flow gain for a linear three way critical center valve and the flow gain for a linear four way critical center valve are the same but the pressure sensitivity is half that of the four way valve. Therefore constant and friction load forces will cause twice the static error in systems with three-way valves. The most serious objection to the use of three way valves is that dynamic errors are also about doubled. In consequence, three way valves are best suited to hydromechanical servos which have little or no loads or can tolerate the error. For this type of valve – used in hydromechanical servos – the source positioning the spool is mechanical and quite stiff compared with the force loads imposed by the spool. Hence, the force equation describing the spool motion is usually unimportant and flow forces are not of interest.

Contribution

It is analysed a hydraulic linear three way valve with two working positions. The valve graphs – Figure 1 – show these two working positions and by the arrows it is indicated the assumed flow direction. A three way valve with three working positions is sketched in Figure 3. The notations in this article include those used in the speciality literature to give thus an example of another way of notating and to help researchers to a better understanding of them, acquiring or familiarizing with them. The analysis methods imposed by the speciality literature are applied for the selected valve. The analysis stages are clearly presented. The continuity equations, the orifice equations, the area calculus equations and the pressure drops are written. The valve coefficients at the operating point are computed and the pressure-flow curves are plotted. The codes written to realise the plots are exemplified in this article. By understanding the code given as example, it can be applied to give the plot for the pressure-flow curves of a hydraulic linear four way valve. The manner of this valve analysis facilitates the analysis of a hydraulic three way valve with two working positions and with rotating valve displacement.

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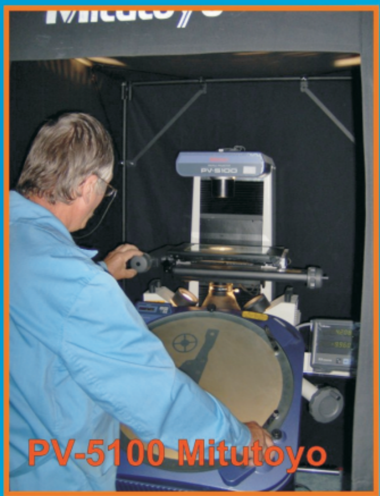

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THE HYDRAULIC DRIVE – A SOLUTION FOR THE MACHINES BENDING BIG SIZE PROFILES

ing. Florin Georgescu, ing. Ionel Nita, ing. Genoveva Vranceanu - INOE 2000-IHP Bucuresti

Abstract: The article makes a comparative analysis of various electromechanical and electrohydraulic drives, component found at cold metal forming machines The article also make reference to several types of machines to be developed in partnership by INOE 2000 IHP and SME enterprises from Romania
.

1. Introduction

In the last years it has been noticed a significant increase of the interest paid by SME enterprises for the machines performing processing operations by plastic deformation at low temperature, equipped with attached devices, capable of performing specific operations and shapes (scrolling, twisting, bending at specific angle etc.). Most of these SME enterprises belong to the civil and industrial engineering field or are small workshops with activities of mechanical processing.

Taking into account these demands existent on the market INOE 2000 IHP realized between 2010-2012 a range of machines for plastic deformation of profiles, in four sizes:

- The machine for bending profiles for round bar of 15 with electromechanical driving of rollers and manual setting;
- The machine for bending profiles for round bar of 30 with electromechanical driving of rollers and manual setting;
- The machine for bending profiles for round bar of 45 with electromechanical driving of rollers and manual setting;
- The machine for bending profiles for round bar of 60 with electrohydraulic driving of rollers and hydraulic setting.

Together with the machines were realized afferent devices purposefully made for twisting, scrolling and bending in right angle of 90^{o}.

The machines and devices were realized within a project financed from European funds during the operation 2.1.1. The project result consisted in 4 machines for bending profiles with 3 afferent devices (for twisting scrolling and bending in a 90^{o} angle.

2. The actual situation in the field of pyramidal profile bending machines

At present the manufacturers of machines for bending metallic profiles with constant or variable radius use mainly the electromechanical or electrohydraulic drive there are also versions with manual drive for small profiles being less used.

In fig 1 is shown the kinematic scheme of an electromechanical drive where:

Me= single or triple phase electric motor with 2 speeds 3000/1500 rpm or 1500/750 rpm)
Rm=gearbox
Ta=gear train
Ra=2 drive rollers
Rf=1 fixing roller

Am=manual drive (screw nut type) of the fixing roller
n=various revolutions reached due to gear units

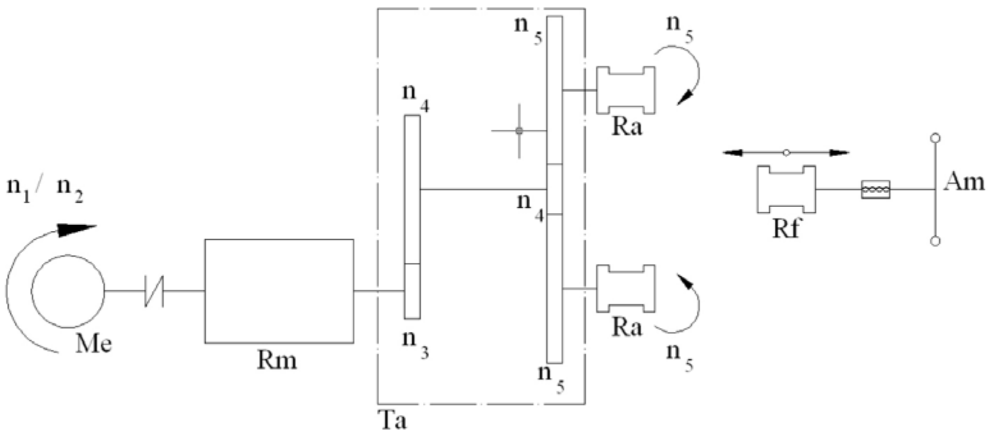

Fig. 1: The kinematic scheme of an electromechanical drive.

Figure 2 shows an overview of an equivalent profile bending machine for round bar of 15...30 with electromechanical drive of rollers and manual setting.

The main parts of an electromechanical drive machine are:

- The electromotor decelerator subassembly consisting of the electric motor supplying the drive power and a worm wheel drive which serves at reducing the revolution of this unit and providing an even transmission;

- The subassembly consisting of tooth wheels which undertake the motion from the worm wheel transmission, reduces it and transmits it to the rollers performing the deformation of profiles;

- The machine framework namely the mechanical subassembly which supports all the other operational components and devices used in the technological process;

- The operational devices ncessary for achieving the desired shapes;

- The automation subassembly by means of which are performed operational orders during the technological process.

The advantages and disadvantages of the electromechanical drive:

1. Advantages:

- The low structural complexity usually enclosing components from the current manufacturing;
- The relatively easy maintenance not requiring anz special knowledge (especially the settings) from the operator;
-Low costs involved in producing the machine allowing a fast return of the investment.

Fig.2: The equivalent profile bending machine for round bar of 15...30 with electromechanical drive of rollers and manual setting.

2. Disadvantages:

- Constant driving speed involving deformation rollers which can be adjusted at various levels required by the kind of material used at profiles (mechanical characteristics and various types of rigidity;
- Relatively small drive forces and moments;
- The subjective involvement of a worker in the initial deformation of the profile.

This type of drive it is generally used in enterprises which manufacture profiles with the same configuration (restricted number in what regards form) in big amounts using a single set of devices.

Taking into account the advantages and disadvantages of the machines with electromechanical drive a more performant solution particularly for obtaining high forces and moments is that of the hydraulic drive applied both for reaching the bending radius and for driving profiles.

Fig. 3 shows a scheme of electrohydraulic drive where vertical displacement of the upper roller and the rolling of rollers is performed by a single hydraulic unit. The main components of this type of machine are:

Uh = hydraulic drive unit
Ch = hydraulic cylinder for positioning fixing the fixing rollers
Mhr = rotary hydraulic motors
Ra = 2 drive rolls
Rf = 1 fixing roll

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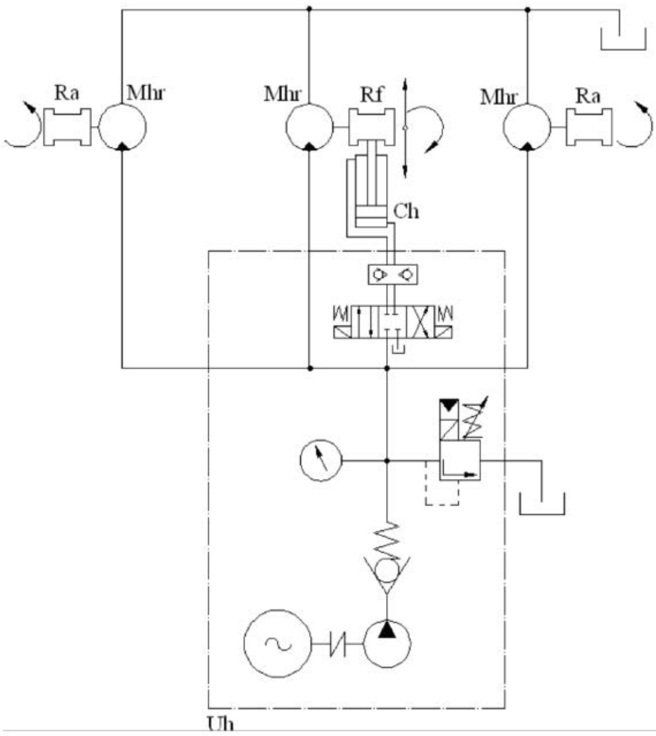

Fig. 3: Scheme of electrohydraulic drive

The advantages and disadvantages of the electrohydraulic drive:

1. Advantages:

- The hydraulic drive unit may be assemblied on machine in a convenient position, both verticall and horizontally for ensuring optimum manoeuvrability conditions for the operator and taking into account the requirements imposed by the type of deformed profile;
- The possibility of adjusting drive speed of the lateral rollers in accordance with the rigidity of the material adjustment possible bz using rotary hydraulic motors;
- The possibility of using a wide range of devices for deformation due to the advantages offered by the hydraulic drive according to the wide possibilities of adjustment of parameters for this type of drive (pressure and flow, drive force and moment).

2. Disadvantages:

- Higher costs involved in making the machines with a slower return of the investment;
- A higher training of the personnel involved in working with this type of machines implying higher costs.

3. Presentation of the solution proposed for the integral hydraulically operated profile bending machine

Taking into account that this type of hydraulic drive represents the main solution adopted almost unanimously in the field of high processing forces the company focused exactly on it for realizing the biggest machine from the category.

This machine may process metallic profiles with various sections (round square rectangular open profiles etc). The max dimension of a round profile in section which may be processed is of 60 mm corresponding to this dimension may be processed other metallic profiles as well.

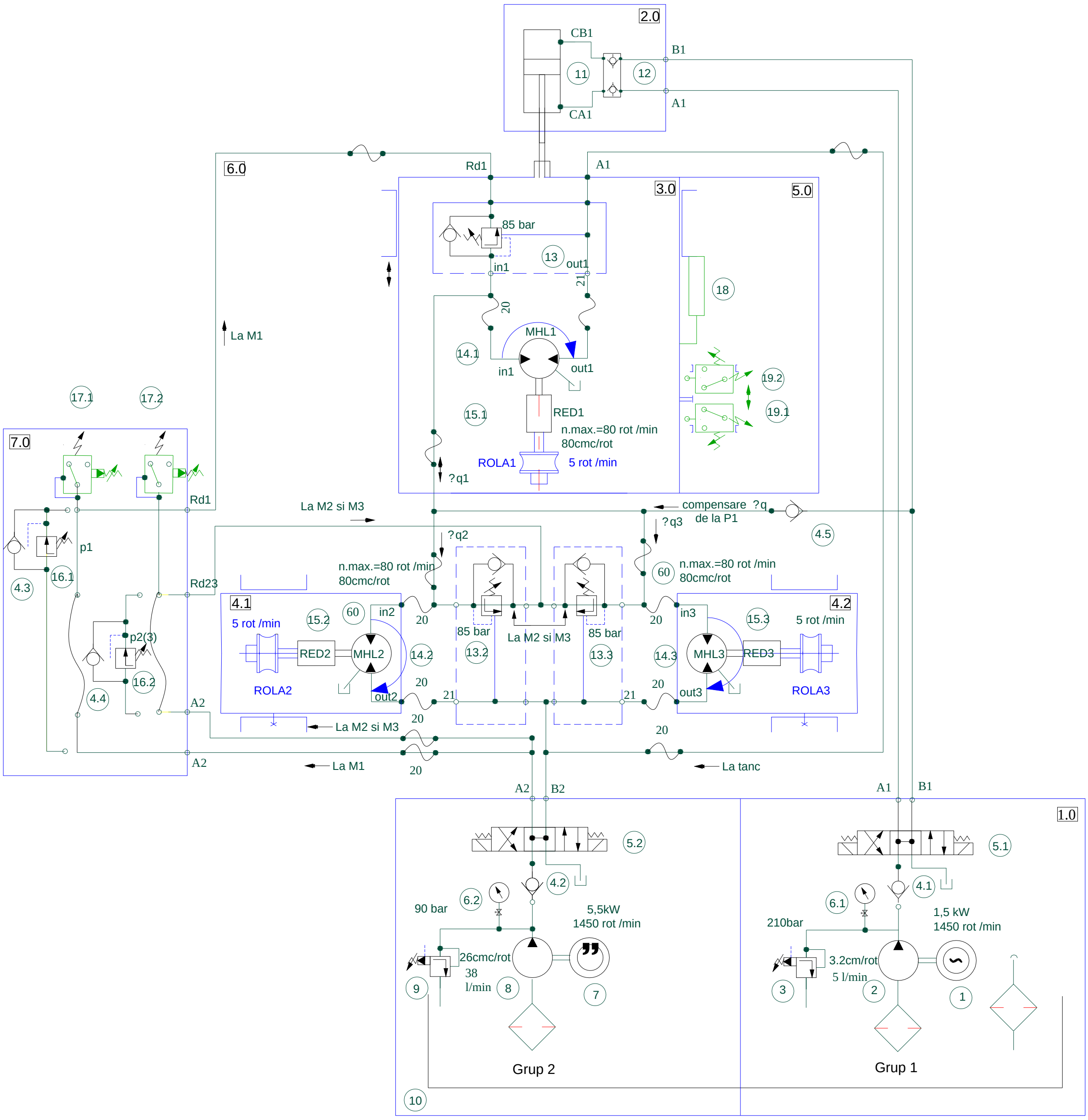

Figure 4 – Hydraulic scheme of bending machine for profiles max. Ø 60

The hydraulic scheme adopted is based on two independent pumping units with the following fucntions:

- Unit 1 ensures the vertical displacemenet of the upper roller supplying the cylinder 11; at this version for the initial deformation of the profile it is used the upper roller;

- Unit 2 ensures the entrainment of the 3 rotary hydraulic motors both for the deformation of the profiles with various radii and for using it with deformation devices.

The entrainment of rollers for rotation is peformed by the 3 slow hydraulic motors 1 motor for each roller the motors MHL1, MHL2, MHL3 are coupled with a speed decelerator, pos.15.1,15.2,15.3 which reduce th speed of the hydraulic motors and increase the entrainment moment.

The upper roller unit is maintained in the desired position by blocking the cylinder 11 by means of the unblocking double valve 12.

For the work with the additional devices (for twisting scrolling and bending in angle) it is used an arbor of the machine as follows:

- For the bending in angle it is used the upper roller unit after replacing the roller with a hob and on the arbors of the other 2 rollers is mounted a fix block;
- For the twisting and scrolling operations are used the arbors of the lower rollers on which are mounted the corresponding devices the arbors entraining the deformed profile.

.

4. Conclusions

The hydraulic drive represents the only solution with aplicability in the industrial field which offer big entrainment forces and moments. In the field of machines for deforming metallic profiles the degree of use of the hydraulic systems is proportional with the required forces so that all manufacturers use for their higher class products this type of drive.

The solution proposed within the project developed in partnership by SC Prestcom SA Focsani and INOE 2000 IHP for the biggest machine offers multiple advantages to the final beneficiary:

- The use of 2 low power hydraulic groups with separate functions;
- Low speeds and high entrainment forces by combining MHL+decelerator;
- The possibility of a separate use of the entrainment arbors for the work with devices.

This machine with a relatively simple mechanical and hydraulic structure based on a modular concept and also the others for the range realized for the project are addressing to the SME type of users with medium qualified personnel that can be easily trained how to use the machine the versions for profiles of 15 or 30 mm may be also used by individuals and small companies.

The wide range of profiles of the rollers and the combination between the machine and various devices broaden possibilities of use.

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COMPACT SYSTEM FOR UPPER ROLL DRIVE FROM BENDING PROFILES PYRAMIDAL MACHINES

Autori: Ing. IONITA Niculae Ing. DUMITRESCU Liliana

Abstract:
Hydraulic drive systems are found in the structure of most industrial machinery, which requires a large driving force, usually accomplished with hydraulic cylinders. Current trends in this field are increasing pressures of work and achieving the hydraulic units in compact structures.
Machines for bending profiles, pyramidal type, use hydraulic drive for variants carrying medium and large forces.

1. Introduction

The development of the civil engineering domain is one of the actual possibilities of economic progress of Romania. The delays in industrial and civil engineering can be turned into a positive factor which to facilitate the increase of activity in the future.

This are required two conditions:

- Availability of SME companies - which are the most dynamic and easily adaptable to market needs - willing to work in this area;
- Availability of appropriate tools, offered at affordable prices.

At present, small firms from the field of processing metallic profiles from the country use equipment with a low complexity, many of them self-made, or purchased like second hand equipment, due to the high price of the new equipment.

The specific of these Romanian firms (small staff, the need for versatile units, able to respond to many different applications) recommends for this activity simple and cheap machines, able to work in conjunction with various devices to extend their usability.

In this context, the initiative of a Romanian company (SME type) – SC PRESTCOM SA Focsani - which, after an analysis of the market and having the advantage of the current activity similar and previous experience in the manufacture of specific components, is welcomed.

2. Opportunities to achieve vertical roll displacement for the profile bending machines

The most used solution for the deformation of the metal profiles with different radius is that of the pyramidal machines with 3 rolls in the shape of a triangle with the base down.

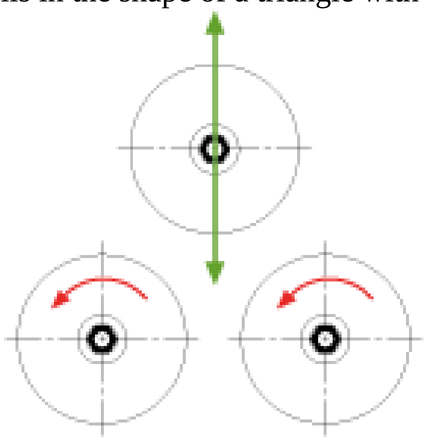

Figure 1 – Principle diagram of pyramid machines

The three rolls are used as follows:
- For small and medium machines, upper roll is to adjusts the bending radius, as non-driving roll (free); for bigger machines, this roll is also driving roll;
- Lower rolls are driving rolls in all cases.

Vertical movement and driving of lower rolls can be made:
- Manual movement and manual driving, for small machines
- Manual movement for upper roll and electromechanical driving for small and medium machines
- Hydraulic displacement for upper roll and electromechanical driving of rolls for medium and large machines
- Hydraulic displacement and hydraulic driving for 2 or 3 rolls, for big size machines

Achieving initial bending radius is the most important phase of the cycle of production; besides to achieve the quota for initial bending of the profile, a special attention should be given to the bending mechanism (whole upper roll + vertical translation mechanism).

If in the small models, this translation is done manually, hydraulic operated machines use a hydraulic unit with a classical scheme. The figure below illustrates such a scheme implemented on a hydraulic machine for the deformation profiles with equivalent section Ø 45 mm.

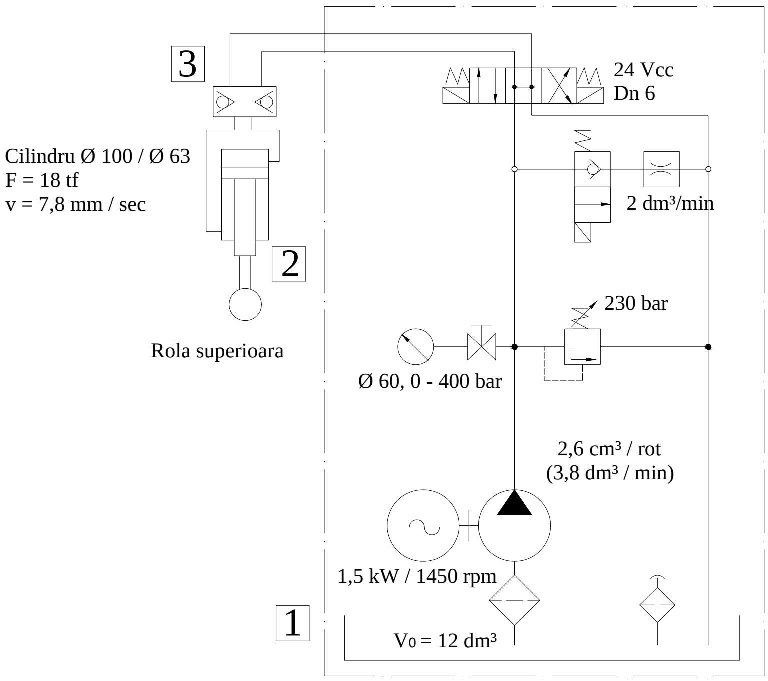

Figure 2 - Hydraulic scheme for upper roll displacement

Achieving large bending forces involves the use of large hydraulic cylinders or high pressure operation.

3. Solution using hydraulic amplifier

For eliminating these disadvantages has been developed a device for plastic deformation designed to be mounted on any cold forming machines for metal profiles, which is able to achieve very high forces, using a low hydraulic pressure; the device works in closed circuit and needs a very small amount of hydraulic oil.

As figure 1 shows, the device consists of the following main parts / components:

1 - vertical cylindrical body with support mounting by the machine;

2 - tubular rod piston;

3 - piston with a compensation of the differences of volume;
4 - spring for pressing the plunger 3;
5 - gear pump, with its own safety valve;
6 - electric motor, DC;
7 - top cover;
8 - hydraulic connection plate;
9 - check valve;
10 - safety valve for retreat;
11 - piston release;
12 - disc for separation;
13 - piston for amplification;
14 - rod exhaust;
15 - head to guide;
16 - roll press;
17 - spring return for plunger amplification 13;
18 - lower cover;
19 - external multiple connector

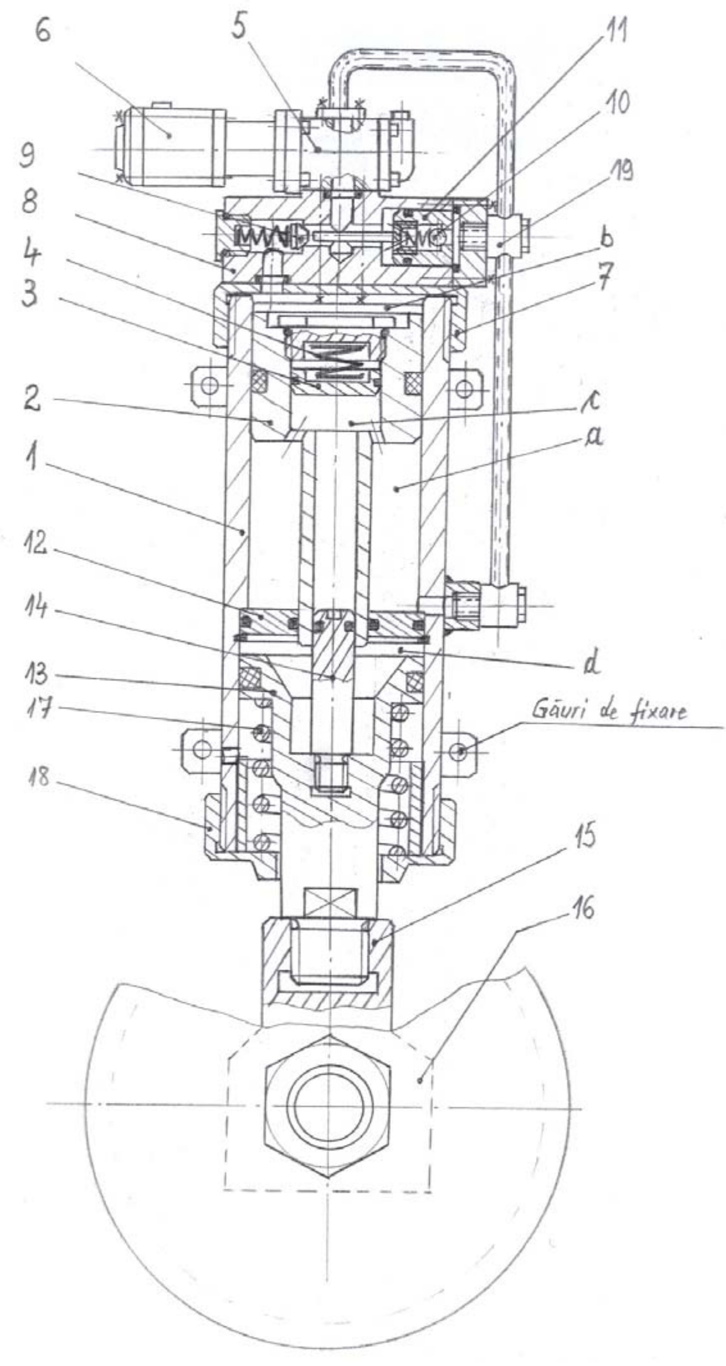

Fig. 3 - The operating device - view in section

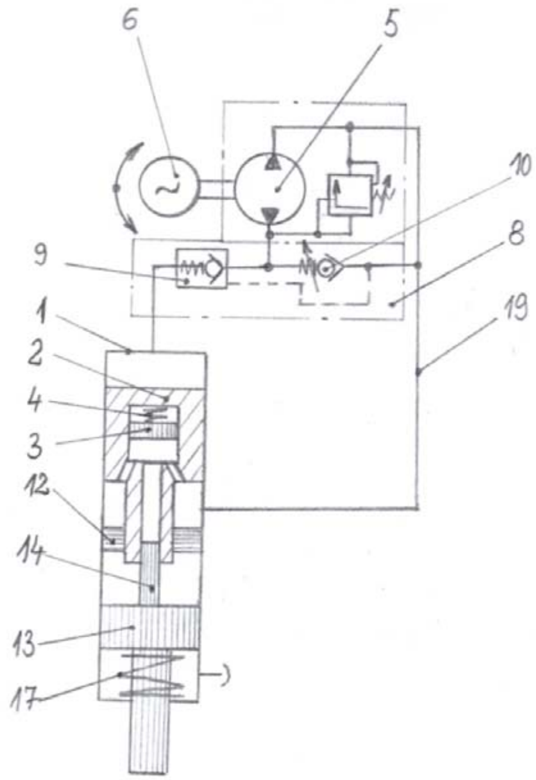

Fig. 4 - The operating device - hydraulic scheme

In figure 4 is shown the hydraulic driving scheme; the analysis shows that driving of bi-directional pump is operated by an electric motor and uses a closed hydraulic circuit and race head protection. To maintain the rate fixed to achieve the bend radius is provided a check valve in place. The hydraulic cylinder has compensation of the two volumes of oil and that also has a camera and a piston amplifier, as able to develop very large forces using hydraulic elements as pump, valve and low pressure hydraulic couplings.

Forming device is composed of a vertical cylindrical body **1**, which found in a piston at its upper part **2**, with tubular rod that containing a compensation piston **3** and a spring **4** which can be powered by a pump gear **5**, which contains its own safety valve and coupled to a DC electric motor **6**, fixed to an upper cover **7** via a connection plates **8**, found a check valve **9** and a safety valve **10**, contained a unlock piston **11**.

The vertical cylindrical body is divided into two chambers by a separate disc **12**, which makes that in a lower room to slide an amplifying piston **13**, which includes a discharge rod **14**, as coupled through a guide head **15**, at a pressing roller **16**, back to its original position by a spring return **17**, initially tight a lower cover **18**.

Between a vertical cylindrical body **1**, gear pump **5** and connections plate **8** is a hydraulic circuit externally materialized by multiple connector **19**.

Before installing the machine and operating all rooms and all hydraulic circuits of cylinder must be filled with mineral oil.

The mode of operation is:

- electric motor **6** is powered from an electrical source, in order to drive gear pump **5** clockwise, the pump will suction oil through multiple connection **19**, from **a** and **c** cavities and suppress it in the **b** cavity, by opening check valve **9**, pushing down the plunger **2**, which through its tubular rod entering in the **d** cavity, achieves a high pressure in it, which imposed the amplification plunger **13**, it transmits through the head guide **15**, by roller pressing **16** a high forming force. At this stage of work the removal of oil from the **c** cavity is helped by the entering of exhaust rod **14**, in the tubular rod of piston **2** and the pressing spring **4** on compensation piston **3**, there to amplifying piston **13** a race less than two piston stroke.

If the drive remains after the plunger **2** has reached the end of its stroke, the hydraulic circuit protection is provided by opening its valve gear pump **5**.

- The interruption of electrical power makes that gear pump **5** stop and the device remains in position under load, since the check valve **9** does not allow discharge of oil from cavity **b** and the retreat of piston **2**;

- If it reverses the polarity of electricity supply, gear pump **5** will turn rotation, suppressed in multiple connector **19**, the pressure pressing the release plunger **11**, which by its own rod opens the check valve **9**, making it possible the oil aspiration from the **b** cavity and retracting the plunger up **2**, followed in the spring force **17** by amplifying piston **13**, being performed the backstroke.

Hydraulic circuit protection at this stage of retreat is ensured by the safety valve **10** which opens at the end of the lifting stroke of the piston **2**.

The condition required for the operation of the hydraulic system in closed circuit is the volume of oil that enters the cavity **b**, in a race, is less than or equal to the volume of oil that can be discharged from the cavities **a** and **c**, this condition be followed when designing the device.

To drive the pump in both directions using a DC motor or AC motor with shift the direction of rotation.

4. Advantages of the proposed solution

The main benefit is the use of a compact unit, the group that generates flow is assembled on the cylinder with a special construction; in this way is eliminated the need for tubes linking the driving group and hydraulic motor which moves the upper roll.

In the analysed scheme is eliminated the hydraulic distributor with three positions and double unlockable valve, in this case using a single piloted check valve.

All the driving system of the upper roll is such a compact structure, low cost, moreover, except hydraulic pump (simple, gear type) and electric motor, the others components can be executed the manufacturer of the entire entity set, when there is an medium technical equipment.

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