CONSTRUCTIVE-FUNCTIONAL ANALYSIS OF SINGLE-ROD DOUBLE-ACTING HYDRAULIC CYLINDERS

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Abstract: *Hydraulic cylinders are subassemblies of hydrostatic equipment which convert the hydrostatic energy to mechanical energy. This is possible by transforming the hydraulic parameters such as pressure and flow of the incoming fluid to parameters associated with translation, force and speed. The hydraulic cylinders usually have, as active element, a subassembly composed from a rod and one or more pistons which translates inside a working cylinder.*

This paper presents the functioning analysis at parallel hydraulic coupling of 3 single-rod double-acting hydraulic cylinders with different diameters and strokes. This is made in order to calculate the corresponding forces and speeds for both displacement directions of rod-piston subassembly.

Keywords: *single-rod, hydraulic cylinder, forces and speeds, functional analysis*

1. Introduction

The wide use of hydraulic drives and automations is due to the perspective regarding the increase in equipment, machinery and installations productivity, the static and dynamic skills, the global reliability and efficiency [Axinti,2008], [Ciocan,2008].

The spread of the use of hydraulic drives is also explained by their special ability, meaning the easiness and simplicity of the synthesis performed for any equipment or installation, as well as the modifications and transition from one technical structure to another [Stan, 2014].

Mainly, the role of a hydraulic system is to transfer the mechanical energy from one place to another. This is performed by assembling the functions of the parts and spreading a hydraulic environment inside them [Tița, 2009].

The hydrostatic systems, figure 1, have a structure comprised of a hydraulic pump *PH* and a rotary engine *MHR* or linear engine *MHL* which are the parts of the *TT* converter, the command and adjustment equipment *ACR* for the value of the hydrostatic parameters, as well as auxiliary elements.

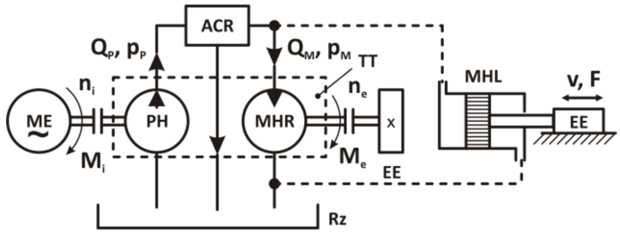


Figure 1: *The structure of a hydrostatic system [Ciocan, 2008]: ME – electric engine; PH – hydraulic pump; ACR – command and adjustment equipment; MHR – rotary hydraulic engine, MHL – Linear hydraulic engine; TT – energy transformer and transmitter block; EE – execution element.*

The HP pump driven by an electric engine EE transforms the received parameters of the mechanical power, n_i – number of rotations and M_i - moment, in parameters of the hydraulic power, Q_P – debit and p_P – pressure. The hydraulic engine receives the hydraulic agent from the pump. With its help, it collects the two hydraulic parameters Q_M, p_M and transforms them in mechanical parameters n_e, M_e or v - speed, F – force and it transfers them to an execution element *EE*.

The hydraulic agent leaked from the hydro-engine can be returned to the pump directly or with the help of a collector *Rz*.

2. Linear hydraulic engines

The hydraulic cylinders, also known as linear hydraulic engines are elements of the hydraulic systems, generally for execution.

They regularly have as active elements a subassembly formed of a rod with one or more pistons which move inside work cylinders, figure 2.

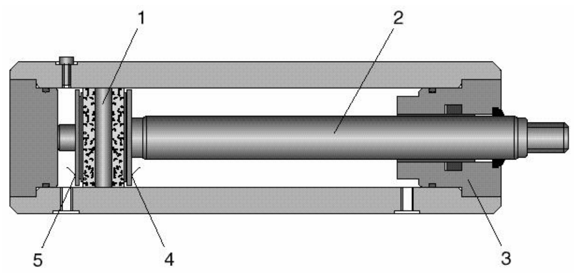


Figure 2: *Linear hydraulic engine with rod and piston [7]: 1 – piston; 2 – rod; 3 – guiding element; 4, 5 – piston surfaces.*

Generally, the individual powering of a hydraulic cylinder is performed from a pump by distributors with two positions – solution which allows the rod-piston subassembly to stop at the end of the stroke in the extreme positions, figure 3 or distributors with three positions – solution which allows the blocking of the rod-piston subassembly in any position – figure 4.

If the cylinder is powered from a pump by a distributor with three positions, but shock valves are also fixed between the chambers of the cylinders, the excess pressure from the pipes connecting the distributors and the cylinder is avoided.

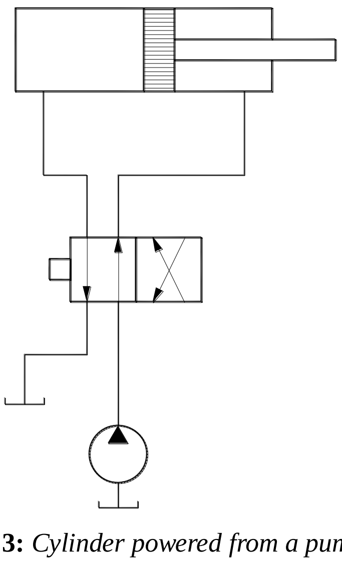


Figure 3: *Cylinder powered from a pump by a distributor with two positions.*

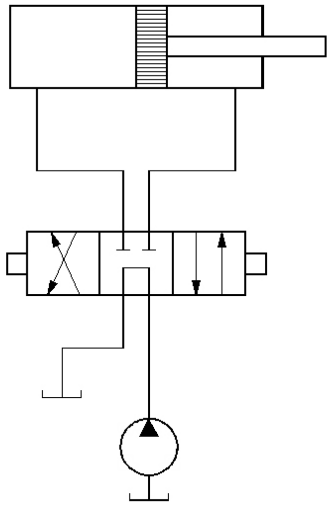


Figure 4: *Cylinder powered from a pump by a distributor with three positions.*

By exceeding the pressure limit in one of the cylinder chambers, the shock valve opens towards the other chamber which presents the contrary, exhaustion, phenomenon which also must be avoided, figure 5. In terms of the number of rods, the construction influences the movement speed of the work body.

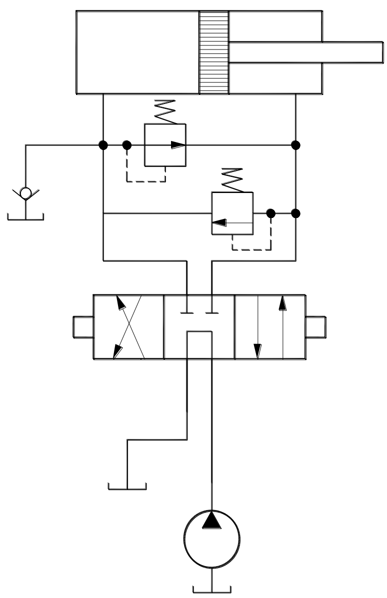


Figure 5: *Cylinder powered from a pump by a distributor with three positions with shock valves.*

3. Stand for constructive-functional analysis of hydraulic cylinders

The stand for the functional analysis of linear hydro-engines, when parallel hydraulic coupling of three linear hydraulic engines is performed, with double effect and unilateral rod parallel hydraulically coupled in range *D/d* (63/40, 50/32, 40/25) mm, in which: *D* is the piston diameter; *d* – rod diameter, figure 6.

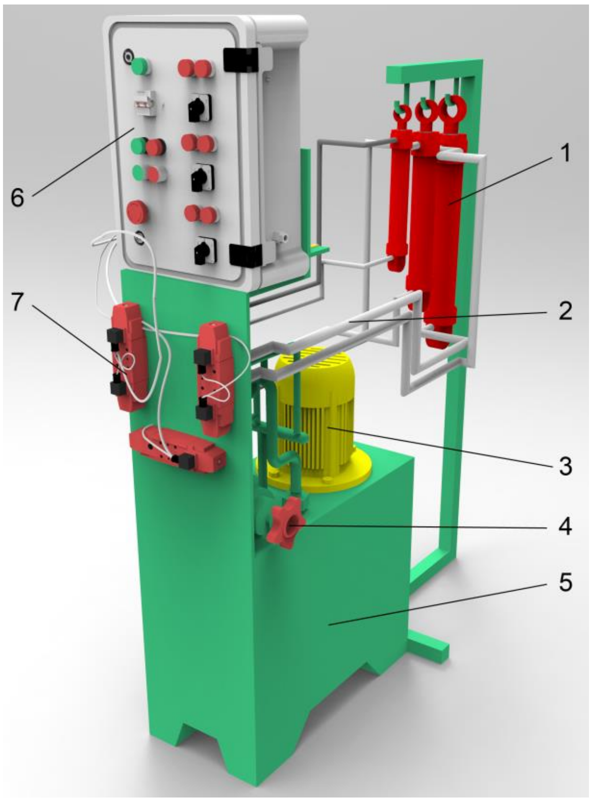


Figure 6: *Stand for constructive-functional analysis of hydraulic cylinders: 1 – hydraulic cylinder; 2 – pipes; 3 – electric engine; 4 – valve; 5 – collector; 6 – switch panel; 7 – distributor.*

A volume pump powers all three hydro-engines with the duty cycle is *AL* (slow advancement), *RR* (quick withdrawal), choosing the type of pump estimating the functional version with constant or variable capacity; setting the constructive version, with sprocket, pallets or axial piston; setting the maximum level of the pressure; learning the financial capacity of the user. A volume unit with constant capacity with sprockets was selected in this case.

The type of pump must have the level for maximum pressure $p_{\max} \geq p_{AL}$.

The closest unit, commercially and financially available is selected from a catalogue of the manufacturing unit, which has $p_{\max} = 5\text{-}250\, bar$.

The character of the rotation drive, fixed or variable and its value must be taken in consideration. A fixed rotation is selected from the range of typical rotations $n = 1500 rot/\min$. The capacity of the pump must insure the maximum debit which is Q_{RR}. In tables 1÷3, the maximum force values of the parameters are centralized in terms of the pressure value and the values of typical parameters of linear hydraulic engines.

Table 1: *Maximum force values performed by the linear hydraulic engine from range D/d (63/40)*

D [mm]	*d [mm]*	*p [bar]*	L_r *[mm]*	F_{max} *[daN]*
63	40	50	2062,6	1559
63	40	100	1458,7	3117
63	40	210	1006,6	6546

in which L_r is the maximum length between the fixing points of the hydro-engine (the length of the reference column).

Table 2: *Maximum force values performed by the linear hydraulic engine from range D/d (50/32)*

D [mm]	*d [mm]*	*p [bar]*	L_r *[mm]*	F_{max} *[daN]*
50	32	50	1663,5	982
50	32	100	1176,2	1963
50	32	210	811,7	4123

Table 3: *Maximum force values performed by the linear hydraulic engine from range D/d (40/25)*

D [mm]	*d [mm]*	*p [bar]*	L_r *[mm]*	F_{max} *[daN]*
40	25	50	1269,4	628
40	25	100	897,6	1256
40	25	210	619,3	2639

The linear hydro-engine is powered with a debit $Q = 6\, l/\min$ rejected by the hydraulic pump. The mobile subassembly of the hydro-engine must overcome a resistant charge and perform a certain stroke.

The capacity of the hydro-engine is considered $\eta_m = 0{,}9$. Moreover, the dimensional features of the pipe which powers the hydro-engine are:

inside diameter $d = 20\,mm$, length from pump to hydro-engine $l = 2\,m$; the features of the hydraulic agent: kinematic viscosity $\nu = 0{,}32\,cSt$; density $\rho = 0{,}9\ g/cm^3$.

4. The calculation of the work parameters of the hydraulic cylinders

4.1 Calculation of rod-piston subassembly speed

The rate of travel of the rod-piston subassembly is in terms of its direction. For the same administered debit, if the hydro-engine has unilateral rod, then the travel rates in the two directions are different, figure 7.

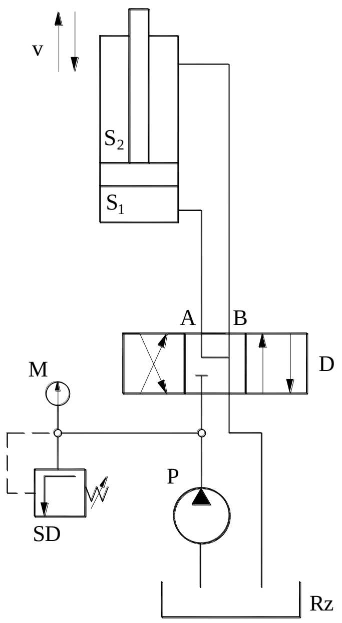


Figure 7: *Hydraulic drive diagram of a linear hydraulic engine: D – distributor; SD – discharge valve; M – manometer; P – pump with constant debit; Rz – collector.*

The speed of the rod-piston subassembly is set with proportion

$$v = \frac{Q}{S}\ \text{[m/min]} \tag{1}$$

in which: S is the active surface of the piston; Q - the value of the oil debit which powers the hydraulic cylinder. S may have the following values:

$$S_1 = \frac{\pi \cdot D^2}{4}\ [\text{mm}^2] \tag{2}$$

or

$$S_2 = \frac{\pi \cdot \left(D^2 - d^2\right)}{4}\ [\text{mm}^2] \tag{3}$$

in which D is the piston diameter, and d - the rod diameter. In this way,

$$v_1 = \frac{4 \cdot Q}{\pi \cdot D^2}; \quad v_2 = \frac{4 \cdot Q}{\pi \cdot \left(D^2 - d^2\right)} \tag{4}$$

4.2 Calculation of axial force developed by the rod-piston subassembly

The axial force is also set in terms of the direction of the rod-piston subassembly and is calculated with proportion:

$$F = p \cdot S\ \text{[daN]} \tag{5}$$

in which: p is the necessary pressure in the surface chamber of the hydraulic cylinder.

Taking into account the two surfaces of the hydro-engine, S_1 and S_2 the axial forces developed by the rod-piston subassembly become:

$$F_1 = p \cdot \frac{\pi \cdot D^2}{4}; \quad F_2 = p \cdot \frac{\pi \cdot \left(D^2 - d^2\right)}{4} \tag{6}$$

In tables 4÷6 the values of the forces and the speeds related to the movement of the linear hydraulic engines range *D/d* (63/40, 50/32, 40/25) are presented, for debit $Q = 6\,l/\min$.

Table 4: *Values of forces and speeds of linear hydraulic engine from range D/d (63/40), Q=6 l/min.*

p *[bar]*	*F_1* *[daN]*	*F_2* *[daN]*	*v_1* *[m/min]*	*v_2* *[m/min]*
50	1559	930	1,92	3,22
100	3117	1861	1,92	3,22
210	6546	3907	1,92	3,22

Table 5: *Values of forces and speeds of linear hydraulic engine from range D/d (50/32), Q=6 l/min.*

p *[bar]*	*F_1* *[daN]*	*F_2* *[daN]*	*v_1* *[m/min]*	*v_2* *[m/min]*
50	982	580	3,06	5,18
100	1963	1159	3,06	5,18
210	4123	2434	3,06	5,18

Table 6: *Values of forces and speeds of linear hydraulic engine from range D/d (40/25), Q=6 l/min.*

p [bar]	*F*$_1$ [daN]	*F*$_2$ [daN]	*v*$_1$ [m/min]	*v*$_2$ [m/min]
50	628	383	4,77	7,84
100	1256	766	4,77	7,84
210	2639	1608	4,77	7,84

5. Finite element analysis of the requirements of the linear hydraulic engines

The linear hydraulic engine from range *D/d* (63/40), figure 8 is taken in consideration in the finite element analysis [Beznea,2012]

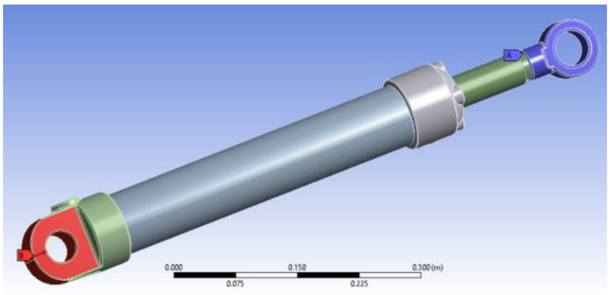


Figure 8: *Linear hydraulic engine from range D/d (63/40).*

The power of the hydraulic drive engine is given by proportion from figure 9:

$$p \cdot S = \sum R = F_a + R_f^{'} + R_f^{''} + R_f^{'''} + \\ + G_1 + F_c + F_i \qquad (7)$$

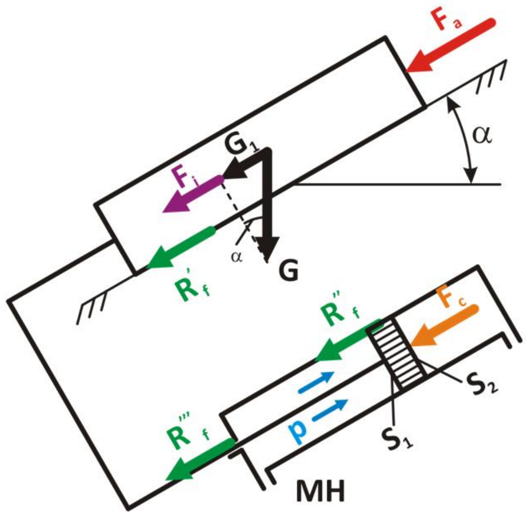
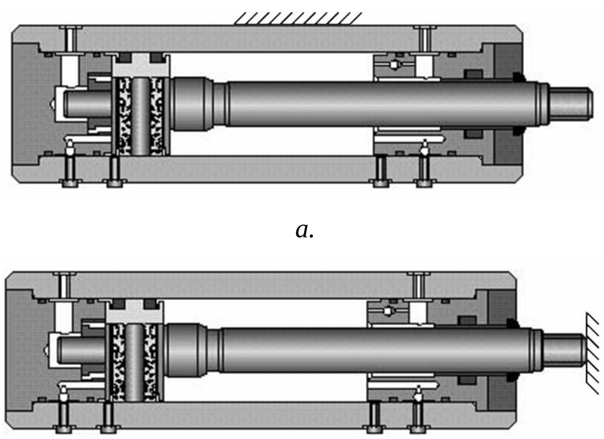


Figure 9: *Forces resistant to drive with straight-line engines.*

in which:
- S active surface of hydro-engine;
- F_a – axial force;
- R'_f – frictional force in the guides of the work body;
- R''_f - frictional force in the sealing elements of the piston;
- R'''_f - frictional force in the sealing elements of the rod;
- G_1 – weight of work body in the movement direction;
- F_c – the counter-pressure force from the inactive chamber of the hydro-engine;
- F_i – the inertia from the starting and breaking periods.

Two work situations are considered, which are defined in terms of the mobility of the hydro-engine elements: the hydro-engine has fixed cylinder and mobile piston, figure 10a and the hydro-engine has mobile cylinder and fixed piston, figure 10b.



a.

b.

Figure 10: *Mobility of hydro-engine elements.*

FEM analysis of hydro-engine with mobile cylinder and fixed piston

Following the modeling and static structural analysis with finite element for the hydro-engine with mobile cylinder and fixed piston, the values of the deformation and the maximum stress are defined in table 7, figures 11 and 12.

Table 7: *The deformations and stress – hydro-engine with mobile cylinder and fixed piston.*

	Equivalent Stress [MPa]	*Total Deformation [mm]*
Min	0	0
Max	0,35329	2,0687x10^{-4}

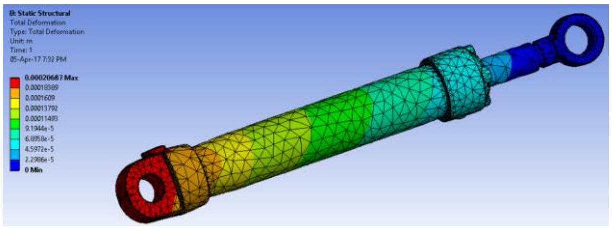


Figure 11: *Deformations – hydro-engine with mobile cylinder and fixed piston.*

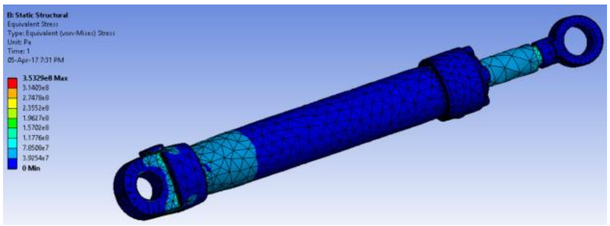


Figure 12: *Stress* **-** *hydro-engine with mobile cylinder and fixed piston.*

FEM analysis of hydro-engine with fixed cylinder and mobile piston

Following the modeling and static structural analysis with finite element for the hydro-engine with mobile cylinder and fixed piston, the values of the maximum deformation and stress are defined in table 8, figures 13 and 14.

Table 8: *Deformations and stress – hydro-engine with fixed cylinder and mobile piston.*

	Equivalent Stress [Pa]	*Total Deformation [mm]*
Min	2,1562	0
Max	$2{,}765 \times 10^{5}$	$3{,}807 \times 10^{-8}$

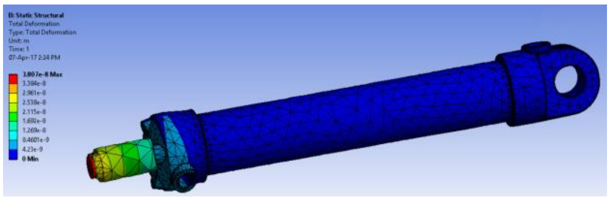


Figure 13: *Deformations – hydro-engine with fixed cylinder and mobile piston.*

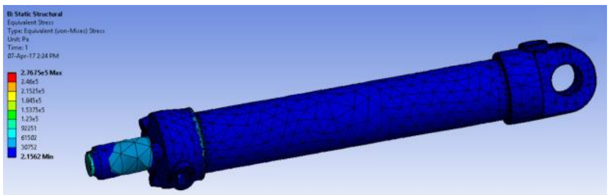


Figure 14: *Stress – hydro-engine with fixed cylinder and mobile piston.*

6 Conclusions

- Due to the fact that the two chambers of the hydro-engines have different surfaces, in $S_1 > S_2$, there is a decrease in the performed force, correlated with a speed increase. The speed will be higher if the debit enters the smaller surface S_1 of the hydro-engine and it will be smaller if the debit enters the larger surface S_2 and vice-versa in forces;

- The diameters of the rods and the pistons at the same debit value will result in force increases along with the increase of the value of the work pressure;

- Following the finite element analysis we have discovered that the total deformations are smaller in the hydro-engine with fixed cylinder and mobile piston in comparison with the hydro-engine with mobile cylinder and fixed piston, while the equivalent stress von-Mises is higher in the hydro-engine with mobile cylinder and fixed piston.

Acknowledgment

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