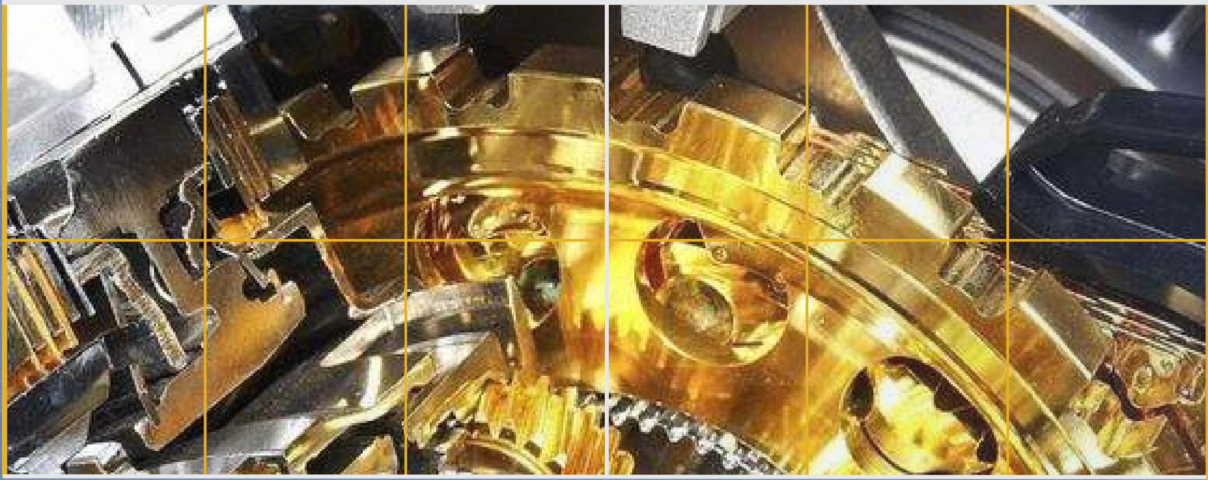
No. 1/2016

# HIDRAULICA

HYDRAULICS-PNEUMATICS-TRIBOLOGY-ECOLOGY-SENSORICS-MECHATRONICS



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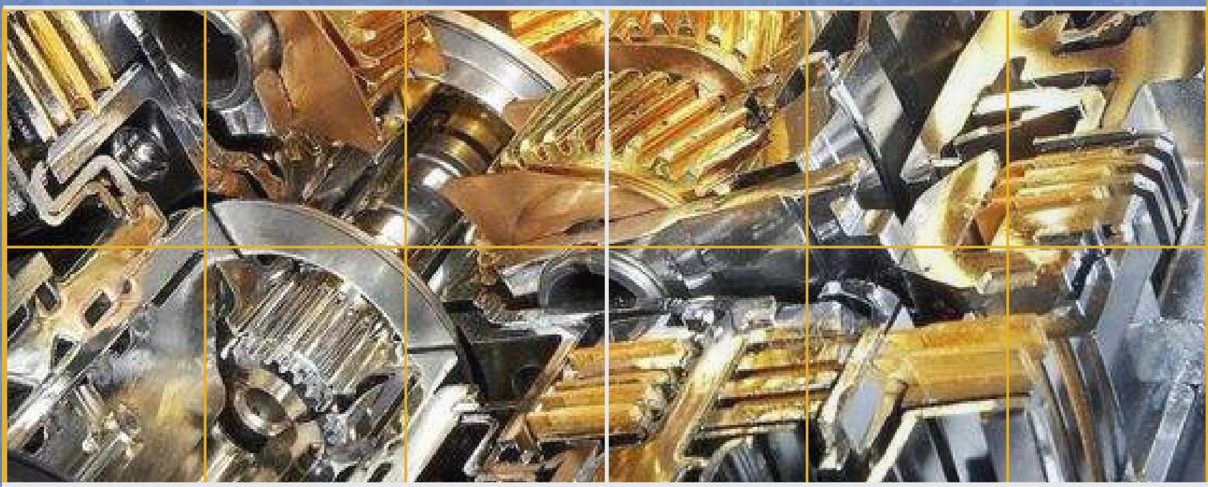

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## CONTENTS

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# EDITORIAL

## Intamplari

Dr. ing. Petrin DRUMEA
DIRECTOR DE PUBLICATIE

In ultimul timp s-au intamplat atat de multe lucruri rele, incat ma simt obligat sa ies din aceasta logica si sa va prezint cateva situatii hazlii spre penibil cu care s-a confruntat revista noastra in relatia cu persoane care au considerat la un moment dat ca activitatea de cercetare se refera doar la cariera, imbunatatirea artificiala a CV-ului si mobilitati, care sa ii sprijine in urcarea unor trepte administrative.

Prima intamplare se refera la comportamentul unui "specialist" care tinea cu tot dinadinsul sa intre in contact cu un profesor renumit dintr-o tara indepartata, nu pentru realizarea unor contacte de cercetare, ci pentru a putea sa se laude cu prieteniile lui la "nivel inalt". Acest specialist a respins o invitatie la o intrunire internationala desfasurata in tara pe domeniul sau de activitate pe motiv ca exact in aceeasi perioada are un invitat care cunoaste pe cineva, care este prieten cu cineva care poate avea acces la un colaborator al profesorului. Specialistul nostru a aflat mai tarziu ca la manifestarea la care fusese invitat a participat si profesorul din tara indepartata si ca astfel a ratat ocazia intalnirii idolului sau.

O alta intamplare este legata de un alt specialist care la aparitia revistei HIDRAULICA ne-a cerut, destul de vehement, sa fie numit director al publicatiei, intrucat el este cel mai mare expert din tara pe domeniile de interes ale revistei. In final lucrurile s-au lamurit colegial si respectivul specialist a devenit prietenul si sustinatorul publicatiei, fara a mai dori sa fie director sau membru al colectivului de redactie.

Un caz interesant l-a constituit cel al unui specialist care a insistat si ne-a tot rugat sa il cooptam in colectivul de redactie pe post de redactor sef sau director, macar pentru o scurta perioda, intrucat acest lucru il ajuta foarte mult la obtinerea unui post de conducere in domeniul cercetarii. Raspunsul nostru negativ l-a facut sa denigreze revista vreme destul de indelungata, pana cand, fiind pus in situatia sa participe la conducerea unei alte reviste, si-a dat seama ca aceasta activitate nu e nici simpla, nici usoara, si ca nu poate fi facuta de persoane care sa nu dispuna de ceva experienta de lucru intr-o redactie.

O intamplare "cu repetitie" se refera la doua cazuri de specialisti destul de tineri care ne-au cerut sa ii cuprindem in colegiul de redactie. Argumentele erau destul de puternice in opinia lor, intrucat aveau la nici 35 de ani CV-uri intinse pe zeci de pagini. Numai apartenenta la colective de redactie si la comitete stiintifice de conferinte nationale si internationale era prezentata pe mai multe pagini, incat ne-am intrebat cum au ajuns acolo si cand aveau timp sa faca si ei ceva. Spre deosebire de aceste doua persoane, marii profesori din domeniu au acceptat sa fie inclusi in astfel de comitete numai dupa ce s-au convins ca pot sprijini cat de cat buna desfasurare a activitatii revistei sau conferintei respective.

Este de remarcat ca am adus in discutie numai intamplari cu persoane foarte calificate, cu bune realizari in domeniile hidraulicii si pneumaticii, dar care au avut si momente pe care in timp au vrut sa le uite, iar dupa o mai lunga perioada au trecut in anecdotica.

## EDITORIAL

### Past Situations

Ph.D.Eng. Petrin DRUMEA
DIRECTOR OF PUBLICATION

So many bad things have happened lately that I feel as an imperative to get out of this logic and present you some funny to embarrassing situations which our journal had to face in relation with people who thought at a given moment that research activity refers only to individual career, artificial improvement of personal CV and mobilities, helping them in climbing hierarchical steps.

The first such situation refers to the behavior of a 'specialist' who wanted by all means to get in contact with a famous professor from a far-away country, not to establish research contacts but to be able to boast on his 'high level' friendships. This expert has rejected an invitation to an international event held in the country in his field of activity arguing that precisely in the same period he has a guest who knows someone, who is a friend of someone who may have access to a collaborator of the Professor. Our specialist found out later that the event he has been invited to has also been attended by the far-away country Professor, and thus he missed the opportunity of meeting his idol.

Another past situation relates to another specialist who on first issuing the HIDRAULICA journal asked us, quite vehemently, to appoint him as director of publication, since he was the greatest expert in the country on the journal areas of interest. In the end things were set by a comradely manner, and the expert in question became a friend and supporter of our journal, no longer wanting to be the director or a member of the editorial board.

An interesting case was that of a specialist who has insisted on and kept asking us to invite him adhere to the editorial board as the journal editor-in-chief or director, even for a short period of time, since this could help him a lot in getting a management position in research area. Our negative reply made him denigrate the journal for quite a long period until, once being put in the situation to get involved in the management board of another journal, he realized that this activity is neither simple nor easy, and it cannot be done by people who lack work experience in an editorial board.

A situation “on repeat” refers to the cases of two quite young specialists who have asked us to include them in the editorial board. Arguments were rather strong according to them, since they, aged not even 35, had CVs spread across dozens of pages. Their affiliation to editorial boards and scientific committees of national and international committees alone spread on multiple pages, making us wonder how they got there and when they had time to do something. Unlike these two individuals, great professors in the field have agreed to be included in such committees only after they were convinced that they can support somehow the proper development of activities relating to the journal or conference in question.

It is noteworthy that I have brought into discussion only cases of highly-qualified individuals, who had good achievements in the fields of hydraulics and pneumatics, but had also moments that they wanted to forget about over time, moments which after a longer period turned into funny stories.

# Numerical Investigation of Unsteady Flows Using OpenFOAM

PhD.Eng. **Alin ANTON**[1]

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***Abstract:*** *In this paper OpenFOAM is used for the numerical investigation of the unsteady flow in a well known numerical benchmark. Von Karman vortex street is obtained in a flow of water past a square cylinder. The results are validated against experimental and numerical investigations from the literature using the drag coefficient. The bazaar-style development of the OpenFOAM toolkit and the principles of Free Software are confirmed to be valid even for the industrial requirements of the computational fluid dynamics (CFD) community. The bazaar-like costs of running OpenFOAM in an established high performance computer center are reduced to the cost of studying the software and understanding how it works.*

***Keywords***: *OpenFOAM, square cylinder, vortex shedding, Karman vortex street*

## 1. Introduction

Unsteady flows present one of the most complex natural phenomena and govern most engineering and technological processes involving fluids [23], [24]. Milk factories and plastic factories use vortex dynamics in order to prevent the fluid from clogging the pipes [18]. Lift-induced wingtip vortices and wake turbulence can have hazardous consequences to airliners [19] just as well as vehicle aerodynamics has a direct implication on passenger safety and gas consumption [20]. Turbulence causes bridge structural failures, insects flight and comet tails [17]. Unlike steady-state flows, unsteady flows are described by partial differential equations and quantities that depend on time. Von Karman vortex street is presented in Fig.1.
Numerical investigation of the unsteady flow with von Karman vortex street behind a bluff body is presented. A bazaar-like Free Software [9] system is used in order to model the phenomenon: an opensource computational fluid dynamics (CFD) toolkit called OpenFOAM [5]. The results are compared and validated with numerical and experimental data from literature.
The software system used belongs to a software paradigm contrasted to the classic hierarchical approach. OpenFOAM is free software as defined by [7] and has been created in the spirit of [8] with a bazaar-style development process [9] without any warranties for the validity of the numerical solutions. The software is used in the CFD industry as a substitute for highly expensive solutions which propose a cost model that depends on the number of computer processors in use.
A software solution is Free Software if it enables the following freedoms [7]:

- The freedom to run the program as anyone wishes, for any purpose (freedom 0).
- The freedom to study how the program works, and change it so it does the computing as anyone wishes (freedom 1). Access to the source code is a precondition for this.
- The freedom to redistribute copies so anyone can help their neighbor (freedom 2).
- The freedom to distribute copies of one’s modified versions to others (freedom 3). By doing this one can give the whole community a chance to benefit from the changes. Access to the source code is a precondition for this.

The rest of the paper is organized as follows. Section 2 presents the test case used for comparison. Section 3 shows the results obtained and discusses them. In Section 4 conclusions are presented.

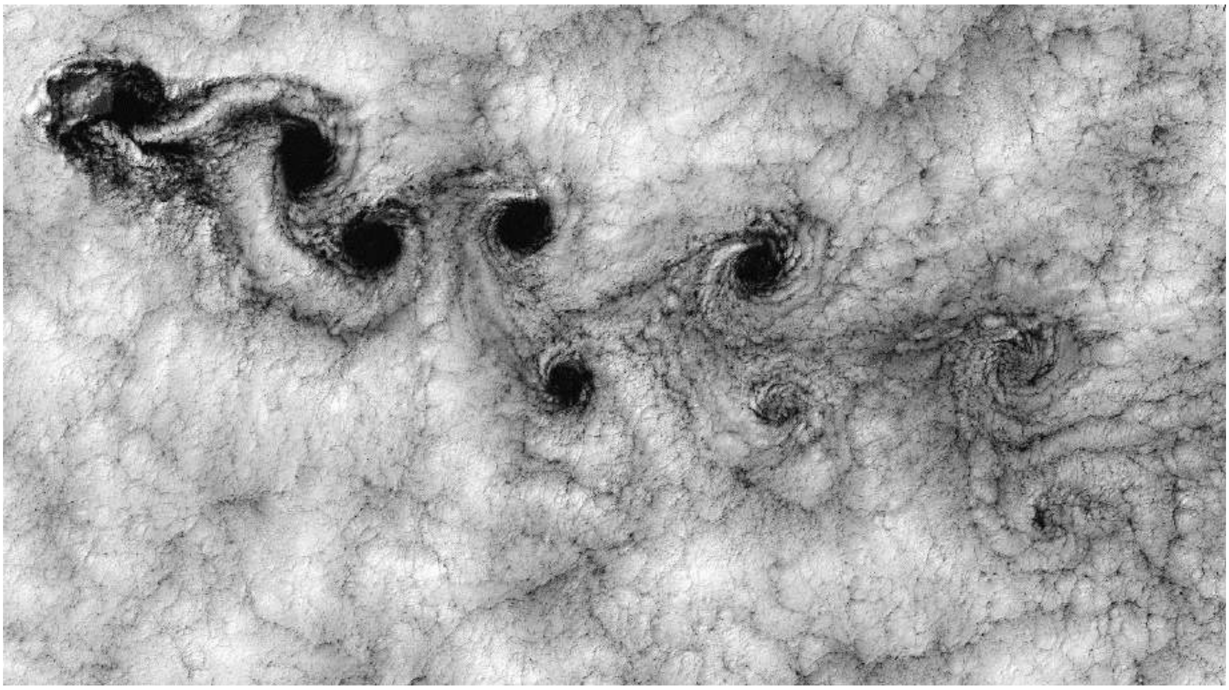

**Fig. 1.** Karman Vortex street formed by clouds near the Robinson Crusoe Island, courtesy of Landsat [21].

## 2. The OpenFOAM Test Case Setup

The European Research Community On Flow, Turbulence And Combustion (ERCOFTAC) created the square cylinder test case which is a well known CFD benchmark used for the validation of numerical solvers. The geometry is shown in Fig. 2.

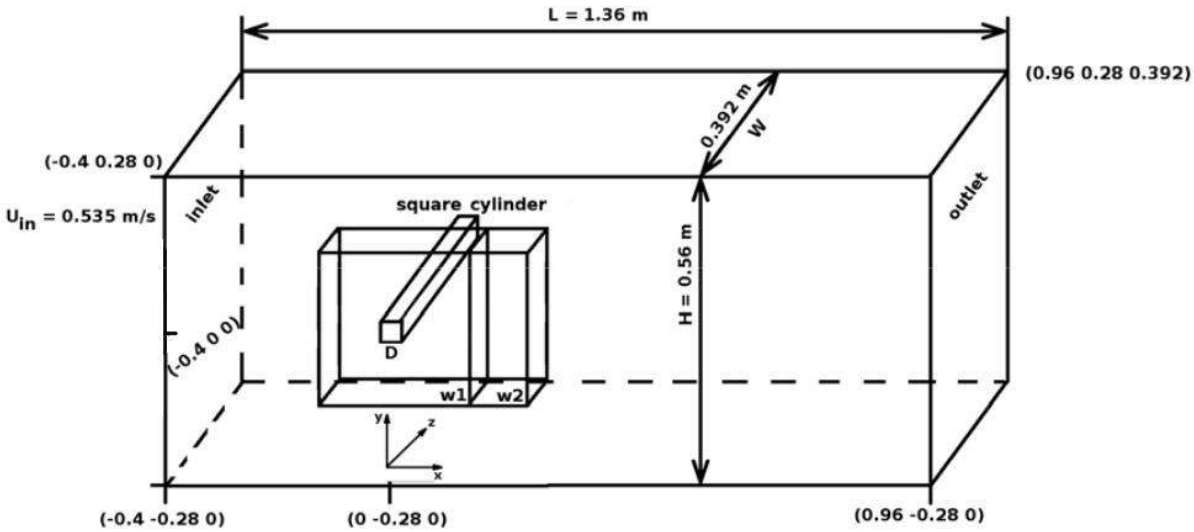


**Fig. 2.** The ERCOFTAC Square Cylinder Test Case

The governing equations are given in terms of continuity (1) and momentum (2) where v is flow velocity, $\rho$ is fluid density, p is pressure,T is the stress tensor and f are body forces.

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho v) = 0 \tag{1}$$

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho v v) = \rho f + \nabla \cdot T \tag{2}$$

A square cylinder is positioned in a channel at 0.38 m from the inlet. The side of the cylinder is D = 0.04 m and the wideness is W = 0.392 m. The length of the enclosing channel is L = 1.36 m and the height is H = 0.56 m. The origins of the coordinate system are located in the center of the square cylinder on the surface of the viewpoint.

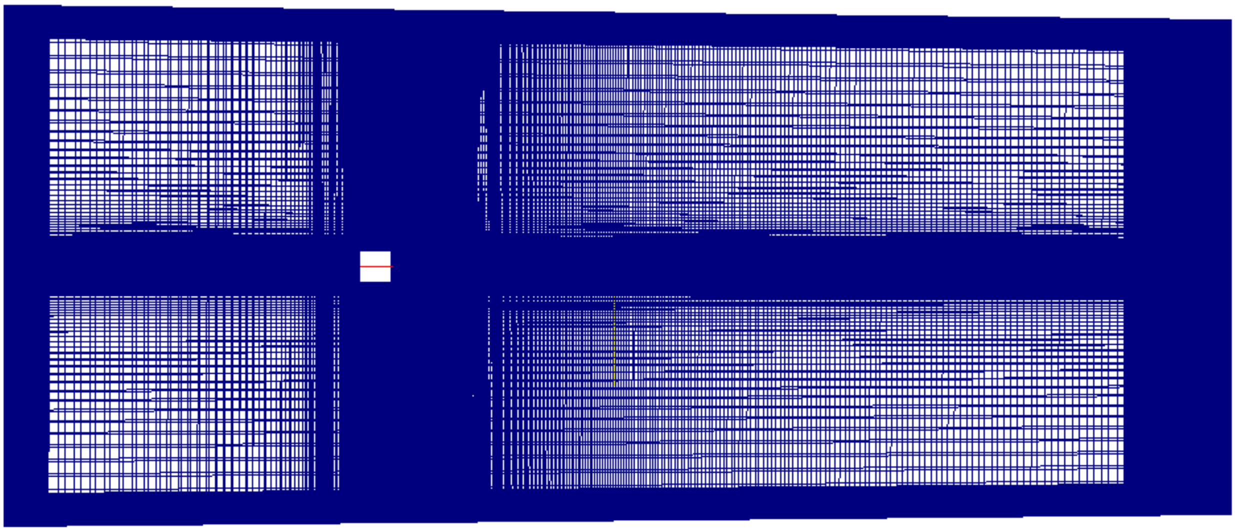

**Fig. 3.** The mesh generated with the blockMesh tool in OpenFOAM, rendered by ParaView [22]

Water enters the channel at inlet with $U_{in}$ = 0.535 m/s and 2% turbulence forming a von Karman vortex street behind the cylinder. The Reynolds number is computed with the side of the square cross section of the cylinder as Re = 21400. The boundary conditions for the walls of the channel are modelled using slip conditions. Only the 4 surfaces of the square cylinder are considered to be walls. Constant pressure is imposed at zero value on the outlet.
The mesh generated with the blockMesh tool from the OpenFOAM toolkit is shown in Fig. 3. It contains 3 million hexahedral cells and is denser around the bluff body in order to capture the vortex street.
Large Eddy Simulation (LES) [11] is used with a sub grid scale (SGS) model [12]. The pressure-velocity coupling algorithm is a blend between Pressure Implicit with Splitting of Operators (PISO) and the Semi-Implicit Method for Pressure-Linked Equations (SIMPLE). The test case is solved using the pimpleFoam application [13].

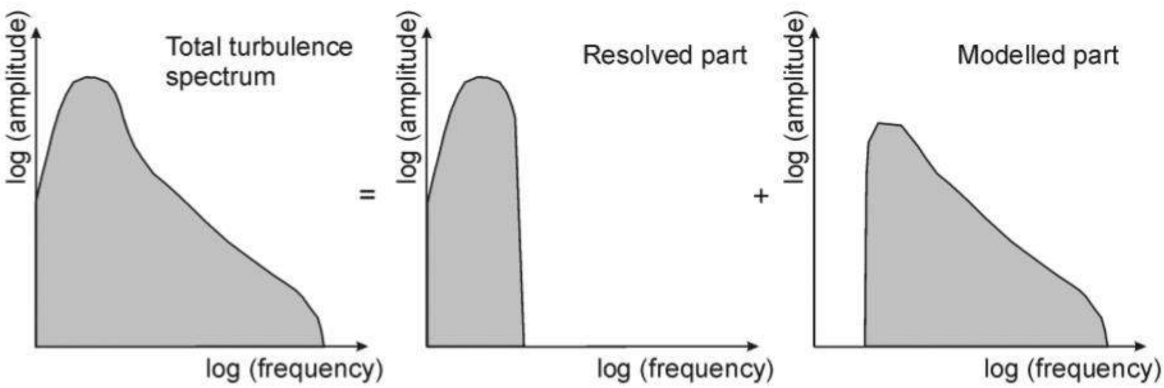


**Fig. 4.** Modelling approach in Very Large Eddy Simulation (VLES) [2]

Fig. 4 shows the modelling approach in VLES. The total turbulence spectrum is obtained by combining the resolved part with the unresolved or modelled part. The tendency to solve complex unsteady turbulent flows at high Reynolds number is to apply the principle "solve less – model more" [2].

**TABLE 1:** Computational requirements for turbulence modelling [14]

| Approach | Reynolds dependency | Empirical | Required mesh size | Required time steps | Availability |
|---|---|---|---|---|---|
| RANS | Weak | High | $10^7$ | $10^{3.5}$ | 1995 |
| VLES | Weak | High | $10^8$ | $10^4$ | 2000 |
| LES | Weak | Small | $10^{11.5}$ | $10^{6.7}$ | 2045 |
| DNS | strong | Zero | $10^{16}$ | $10^{7.7}$ | 2080 |

Table 1 presents the computational requirements for turbulence modelling. The Smagorinsky model is used [15].

### 3. Results and discussion

The simulation is performed on a cluster of 256 processors with a time step $\Delta t = 10^{-3}$ seconds and a fine mesh of 3 million hexahedral cells. The Courant number is monitored to vary between CFL = 0.8 and CFL = 1. Fig. 5 shows the von Karman vortex street forming behind the cylinder as captured by the simulation.

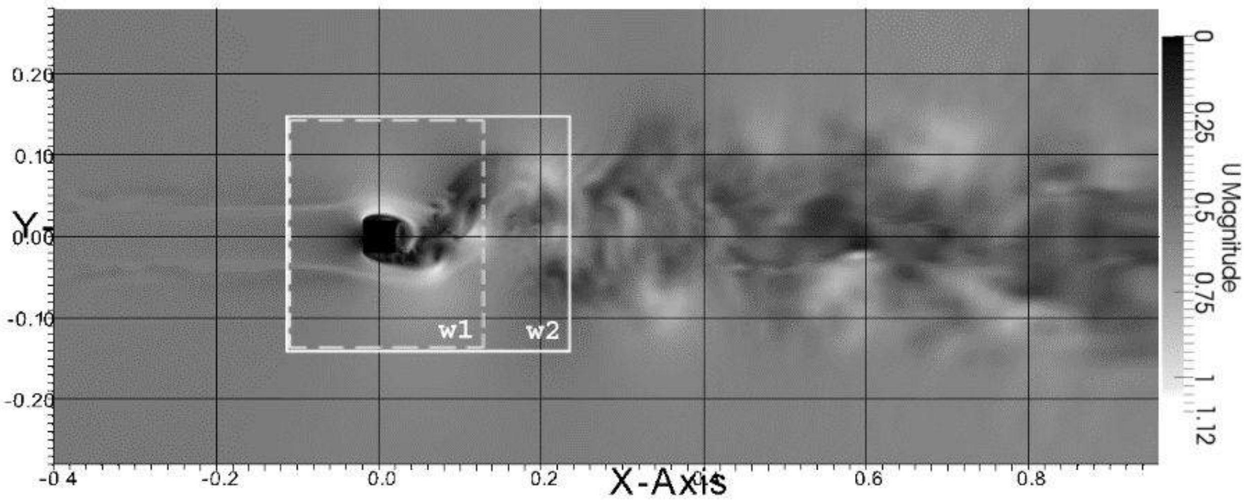


**Fig. 5.** Von Karman vortex street behind cylinder

Fig. 6 shows the drag coefficient for the square cylinder computed using a pressure probe installed downstream the cylinder, on its surface. The drag coefficient $C_d$ varies around 2.2 in agreement with the numerical results of [1] and the experimental data of [3] and [4] where the same test case is analysed at Re = 21400.

**TABLE 2:** Comparison with results from the literature

| Results | Drag coefficient *Cd* | Reynolds *Re* | Investigation type |
|---|---|---|---|
| Reference [1] | 2.03 – 2.78 | 21400 | Numerical |
| Reference [3] | 2.1 | 21400 | Experimental |
| Reference [4] | 2.1 | 21400 | Experimental |
| OpenFOAM results | 2.2 | 21400 | Numerical |

Table 2 compares the results obtained with OpenFOAM with well known literature investigations, both numerical and experimental. At Re = 21400 the obtained drag coefficient of $C_d$ = 2.2 matches the values from the experimental results of [3], [4] and the numerical investigations of [1].Given that the Strouhal number in (3) is

$$St = \frac{fL}{U} \tag{3}$$

the frequency *f* obtained in Fig. 6 also confirms the results previously present in the literature.

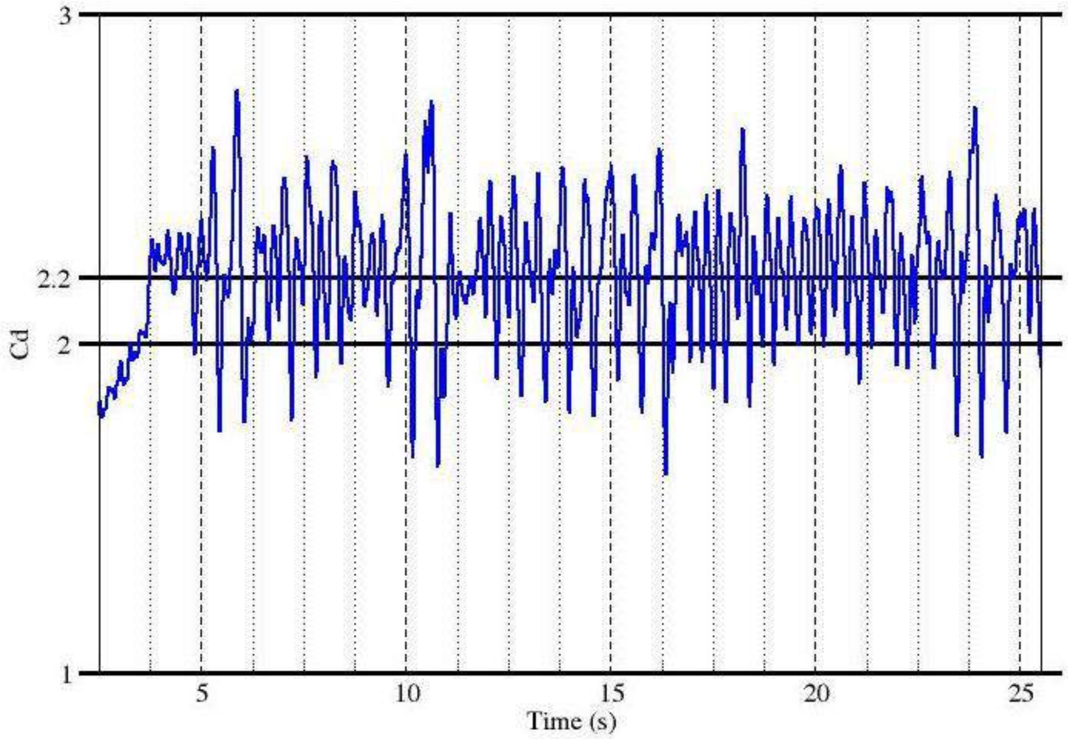


**Fig. 6.** The drag coefficient Cd for the square cylinder

Expensive and proprietary solutions can always be replaced by affordable Free Software with competitive results, which produce the same numerical data much faster and free of charge.

## 4. Conclusion

Numerical investigation of unsteady flow past a square cylinder has been performed. The numerical results obtained with OpenFOAM, a Free Software developed in bazaar-style process have been validated with data from the literature, both numerical and experimental. The drag coefficient has been obtained to be in agreement with the referenced results.
It can be concluded that even though OpenFOAM is a bazaar software with warrantless implementations of the numerical schemes, it provides valid results as compared to the benchmark test cases from the literature. The advantage of OpenFOAM is that of being Free Software and the fact that it also provides free solvers without embedding any additional costs for running it in parallel.

**Acknowledgements**

The author acknowledges Dr. Niklas Nordin for the square cylinder test case. Satellite imagery is credited to NASA Earth Observatory [21]. This work was partially supported by the strategic grant POSDRU/159/1.5/S/137070 (2014) of the Ministry of National Education, Romania, co-financed by the European Social Fund – Investing in People, within the Sectorial Operational Programme Human Resources Development 2007-2013. The author would like to acknowledge the use of computing resources provided by the Black Forest Grid Initiative [16].

**References**

[1] Pourquié, M., Breuer, M., Rodi, W.: Computed Test Case: Square Cylinder, Proc. of the 2nd ERCOFTAC Workshop on Direct and Large-Eddy Simulation, Grenoble, France, Sept. 16-19, 1996, ERCOFTAC Series, vol. 5, pp. 375-379, Direct and Large-Eddy Simulation II, eds. J.P. Chollet, P.R. Voke and L. Kleiser, Kluwer Academic Publishers, Dordrecht, (1997);
[2] Muntean S., Buntic I., Ruprecht A., Susan-Resiga R., A numerical investigation of the 3D swirling flow in a pipe with constant diameter. Part 2: Turbulent computation, Scientific Bulletin of the "Politehnica" Univerity of Timisoara, Transactions on Mechanics 01/2005; 50(64):87-96;
[3] Lyn, D.A., Rodi, W., „The applying shear layer formed by flow separation from the forward corner of a square cylinder”, Journal of Fluid Mechanics, 267, pp. 353–376, 1994;
[4] Lyn, D.A., Einav, S., Rodi, W., Park, J.H., „A laser-doppler velocimetry study of ensemble-averaged characteristics of the turbulent near wake of a square cylinder”, Journal of Fluid Mechanics, 304, pp. 285–319, 1995;
[5] Weller, H.G., Tabor, G., Jasak, H., Fureby, C., „A tensorial approach to computational continuum mechanics using object-oriented techniques”, Comput. Phys., 12, 6, pp. 620–631, 1998;
[6] Penttinen, O., „A pimplefoam tutorial for channel flow, with respect to different LES models”, Technical Report, Chalmers University of Technology, November 2011;
[7] Richard Stallman. 2009. Viewpoint: Why "open source" misses the point of free software. Commun. ACM 52, 6 (June 2009), 31-33. DOI=http://dx.doi.org/10.1145/1516046.1516058;
[8] Stallman R (2005) Free Community Science and the Free Development of Science. PLoS Med 2(2): e47. doi:10.1371/journal.pmed.0020047.
[9] Eric S. Raymond. 2001. The Cathedral and the Bazaar: Musings on Linux and Open Source by an Accidental Revolutionary. O'Reilly & Associates, Inc., Sebastopol, CA, USA;
[10] U. Ayachit, B. Geveci, K. Moreland, J. Patchett, and J. Ahrens. The ParaView visualization application. In E. W. Bethel, H. Childs, and C. Hansen, editors, High Performance Visualization, chapter 18, pages 383–400. CRC Press, 2012;
[11] Ruprecht, A., Helmrich, T., Buntic, I., Very large eddy simulation for the prediction of unsteady vortex motion, Proceedings of the 12th International Conference on Fluid Flow Technologies, Budapest, 2003;
[12] De Villiers, E., The Potential of Large Eddy Simulation for the Modeling of Wall Bounded Flows, PhD Thesis, Imperial College of Science, Technology and Medicine, 2006;
[13] Penttinen, O., A pimplefoam tutorial for channel flow, with respect to different LES models, Technical Report, Chalmers University of Technology, November 2011;
[14] Ruprecht A., Institute of Fluid Mechanics and Hydraulic Machinery University of Stuttgart,”Turbulence modelling for hydraulic machinery”, 2015;
[15] Smagorinsky J., General circulation experiments with the primitive equations I. the basic experiment Mon. Weather Rev., 91 (3) (1963), pp. 99–164;
[16] *** bwGRiD: Member of the German D-Grid initiative, funded by the Ministry of Education and Research (Bundesministerium für Bildung und Forschung) and the Ministry for Science, Research and Arts Baden-Württemberg (Ministerium für Wissenschaft, Forschung und Kunst Baden-Württemberg); http://www.bwgrid.de/;
[17] Jenkins, R.M. (June 2004). The Star of Bethlehem and the comet of AD 66, Journal of the British Astronomy Association (114) No. 6. pp. 336–343;
[18] Lelievre C, Faille C, Benezech T 2001. Removal kinetics of Bacillus cereus spores from stainless steel pipes under CIP procedure: influence of soiling and cleaning conditions. J Food Process Eng 24(6):359–79;
[19] P. Henningsson, F. T. Muijres, A. Hedenström, Time-resolved vortex wake of a common swift flying over a range of flight speeds, J. R. Soc. Interface 2011 8 807-816; DOI: 10.1098/rsif.2010.0533. Published 19 April 2011;
[20] J. Keogh, G. Doig and S. Diasinos, Flow compressibility effects around an open-wheel racing car, Aeronautical Journal, Vol. 118, No. 1210, December 2014, ISSN 0001-9240;
[21] Earth Observatory, NASA, http://earthobservatory.nasa.gov/, accessed 2015;
[22] Utkarsh Ayachit. The ParaView Guide (Full Color Version): A Parallel Visualization Application. Kitware, Incorporated, paraview 4.3 edition, January 2015;
[23] Barglazan M., Bordeasu I., Contribution to hydraulic turbines draft tube design, “Hidraulica” (No. 2/2014) Magazine of Hydraulics, Pneumatics, Tribology, Ecology, Sensorics, Mechatronics, ISSN 1453 – 7303, pp.32-38;
[24] Micu L. M., Bordeasu D., Bordeasu I., Popescu M., Oanca O. V., Duma S. T., Influence of nitriding thermochemical treatments upon cavitation erosion resistance of duplex X2CrNiMoN22-5-3 stainless steels, “Hidraulica” (No. 4/2014) Magazine of Hydraulics, Pneumatics, Tribology, Ecology, Sensorics, Mechatronics, ISSN 1453 – 7303, pp.27-33.

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# Modeling and Simulation of Conical Poppet type Relief Valve with Damping Spool

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***Abstract:*** *This paper deals with the simulation of poppet type pressure relief valve with damping spool to analyze transient response. Mathematically explained the performance characteristics of this valve based on original research carrying out on the compound pressure relief valve and related to hydraulic pressure control valve. Pressure control valves allow establishing sequences in hydraulic operations via scaling the pressure in different parts of the system. The equations describing the dynamic performance the valve are deduced and simulated in the MATLAB/SIMULINK software.*

***Keywords****: Damping spool, pressure, relief valve, simulation, transient response.*

## 1. Introduction

Pressure control valves are used to select pressure levels at which particular parts of the circuit must work; they control actuator force, avoid hydraulic system damage, power wastage and circuit overheating. Relief valves are used in hydraulic circuits as safety valves; they are normally closed and open whenever circuit pressure overcomes a certain value. In order to insure fluid tightness and therefore minimize leakage, poppet element is usually of conical or spherical.
In the case of a direct-operated relief valve, the poppet is actually a spring-supported mass. This mass-spring system is subjected to very low viscous friction and spring material structural-damping forces. Then, in the steady state, this valve may suffer from sustained oscillations of the poppet, which results in observable pressure oscillations. Therefore, it is necessary to add a damping spool to the valve. Figure 1 shows a direct operated pressure relief valve with a damping spool.
Relief valves are connected with high-pressure and return low-pressure lines. They are used to limit the maximum operating pressure in the high-pressure lines. The relief valve consists mainly of a poppet, loaded by a spring. The poppet is pushed by the spring to rest against its seat in the valve seating. The spring pre-compression force is adjusted by a spring seat screw.

## 2. Valve structure and operation

Figure 1 shows a schematic of the studied direct-operated relief valve. The poppet is subjected to spring, inertia, flow reaction forces and pressure forces. The poppet rests against its seat as long as the pressure force is less than on the forces on the spring side of the poppet element. These forces are equal when the pressure reaches the cracking pressure. For further increases of pressure, the poppet is displaced and the oil flows from the high-pressure line to the return line.

The valve consists of a poppet valve (2), rigidly attached to a damping spool (1). The poppet is loaded by a spring of stiffness krv. The spring is pre-compressed by an adjustable pre-compression distance, x. When the valve is not operating, the spring pre-compression force (krv xo) pushes the poppet against its seat. The seat produces an equal seat reaction force FSR. When the input pressure Pis increased, the liquid flows to the damping chamber through the radial clearance of the damping spool, Qd. The pressure, Pd, increases and acts on the damping spool. When the valve is closed, the pressure Pat the valve inlet chamber does not produce any axial force on the moving parts.

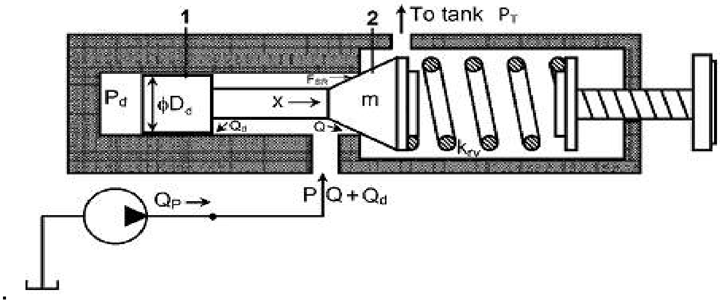


**Fig. 1.** A direct-operated relief valve connected to a pump supply

But, as the poppet valve opens, the poppet area subjected to inlet pressure $A_p$ , becomes less than the damping spool area $A_d$. The pressure Pacts on the area difference ($A_d$– $A_p$) to the left.
Neglecting the seat reaction forces, the motion of the damping spool and poppet is governed by the spring force, the jet reaction force, inertia force, damping force and the pressure forces. When the pressure force ($P_d$ $A_d$) exceeds the forces on the spring side, the poppet displaces, opening the path from the inlet port to the drain port, with a pressure $P_T$. The variation of the pressure $P_d$ is resisted by the radial clearance, which throttles the connection of the inlet port with the damping spool chamber. The relief valve is connected to the delivery line of a fixed displacement pump rotated by a constant speed. A bypass valve is connected to the pump delivery line to control the loading pressure.

## 3. Mathematical modelling

This section deals with the deduction of a mathematical model describing the dynamic performance of a relief valve with damping spool.

<u>Assumptions</u>

- The outlet pressure is assumed to be equal to the atmospheric pressure.
- A stable supply to the valve inlet port is ensured.
- The spring stiffness is constant.
- Cd (Re) is a discharge coefficient at the valve inlet which in general depends on the Reynolds number, although this dependence will be neglected in our subsequent analytical and numerical investigation.
- The effect of the transmission lines was neglected

The dynamic behaviour of the valve is described by the following set of mathematical relations. The dynamic model can be obtained by adding the inertial force and viscous damping force into the steady state force balance equation and including an equation relating the rate of pressure change to the net dynamic flows. The system relief pressure will be higher than the preset cracking pressure due to both the mechanical and hydraulic spring forces existing when relief flow is passing through the valve. This is normally called a pressure override.

<u>The Poppet Valve Throttling Area</u>

The following mathematical expressions for the poppet area, $A_p$, subjected to the pressure P and poppet valve throttle area $A_t$, were deduced.

$$A_{po}(x) = \pi x sin\theta\{d_p - x\, sin\theta\, cos\theta\} \tag{1}$$

$$A_p = \frac{\pi}{4}\{d_p - 2x sin\theta cos\theta\}^2 \tag{2}$$

The motion of the poppet under the action of pressure, viscous friction, inertia, and external forces is described by the following equation:

$$P_d A_d + F_{SR} - P(A_d - A_p) = m\frac{d^2x}{dt^2} + f\frac{dx}{dt} + k_{rv}(x_0 + x) \tag{3}$$

Seat Reaction Force

The poppet displacement in the closure direction is limited mechanically. When reaching its seat, a seat reaction force takes place due to the action of the seat stiffness and structural damping of the seat material. These two effects are introduced by the equivalent seat stiffness ks and damping coefficient Rs.

$$F_{SR} = \begin{cases} 0 & x > 0 \\ k_S\,|x| - R_s\frac{dx}{dt} & x < 0 \end{cases} \tag{4}$$

Flow rate through the radial clearance of the damping spool:

$$Q_d = \frac{\pi D_d c^3}{12\mu l}(p - p_d) \tag{5}$$

Flow rate through the poppet valve:

$$Q = C_d A_{po}\sqrt{\frac{2p}{\rho}} \tag{6}$$

Continuity equation applied to the damping spool chamber:

$$Q_d - A_d\frac{dx}{dt} = \frac{V_o + A_d x}{\beta}\frac{dp_d}{d_t} \tag{7}$$

Pump flow rate:

$$Q_p = Q_t - \frac{p}{R_L} \tag{8}$$

Continuity equation applied to the poppet chamber:

$$Q_p - Q_d - Q_r = \frac{V}{\beta}\left(\frac{dp}{dt}\right) \tag{9}$$

In the steady state, the valve poppet reaches equilibrium under the action of the pressure forces, spring force, and jet reaction forces.

## 4. Computer simulation

A fluid power circuit mathematical model is a collection of nonlinear flow equations associated with components together with flow continuity and force equations applicable to components and actuators. The equations put together in the form of block diagram using a comprehensive library of mathematical functions in the MATLAB Simulink package.

The system has one linear differential equation for force and a nonlinear differential equation for flow-rate continuity, the nonlinear component being due to the poppet back chamber volume changes with time. The use of differential equations and linearized differential equations opens up a powerful analysis avenue to aid the understanding of the dynamic behaviour of fluid power circuits.

Equations (1) to (9) describe the dynamic behavior of the valve system. These equations were used to develop a computer simulation program i.e. MATLAB/SIMULINK, which was employed to plot the valve's dynamic characteristics. The simulation program runs for the transient response of valve pressures to step input flow rate.

This simulink block diagram contains the appropriate mathematical models, inputs, and plotting facilities, as shown in Fig. 2. The dynamics exists at the opening of the poppet, because of the sudden application of the flow rate, these effects are very fast and have more effect on the dynamic response of the valve system.

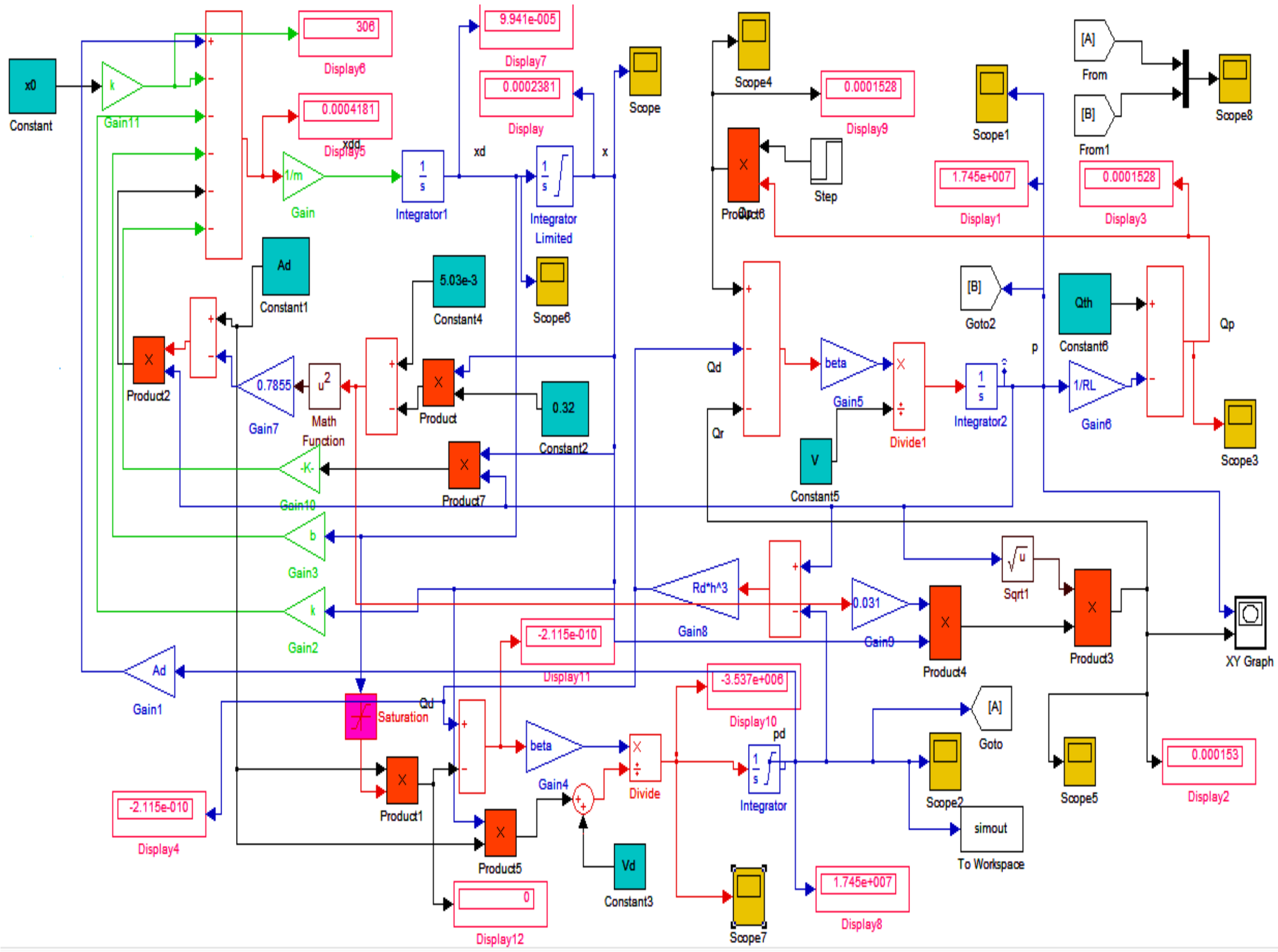


**Fig. 2.** Simulink Model of Pressure Relief Valve with damping spool

## 5. Results and discussion

The settling time is defined as the time the system takes to reach a steady state condition, while the peak time refers to the time the system takes to achieve its peak condition after the first overshoot with the same input. The transient response indicates a condition where both the settling and peak times are in the appropriate range.

The transient response of the valve was simulated for step supply flow rate. The simulink program runs for different values of the damping spool radial clearance. The transient response of valve input pressure was simulated and plotted.

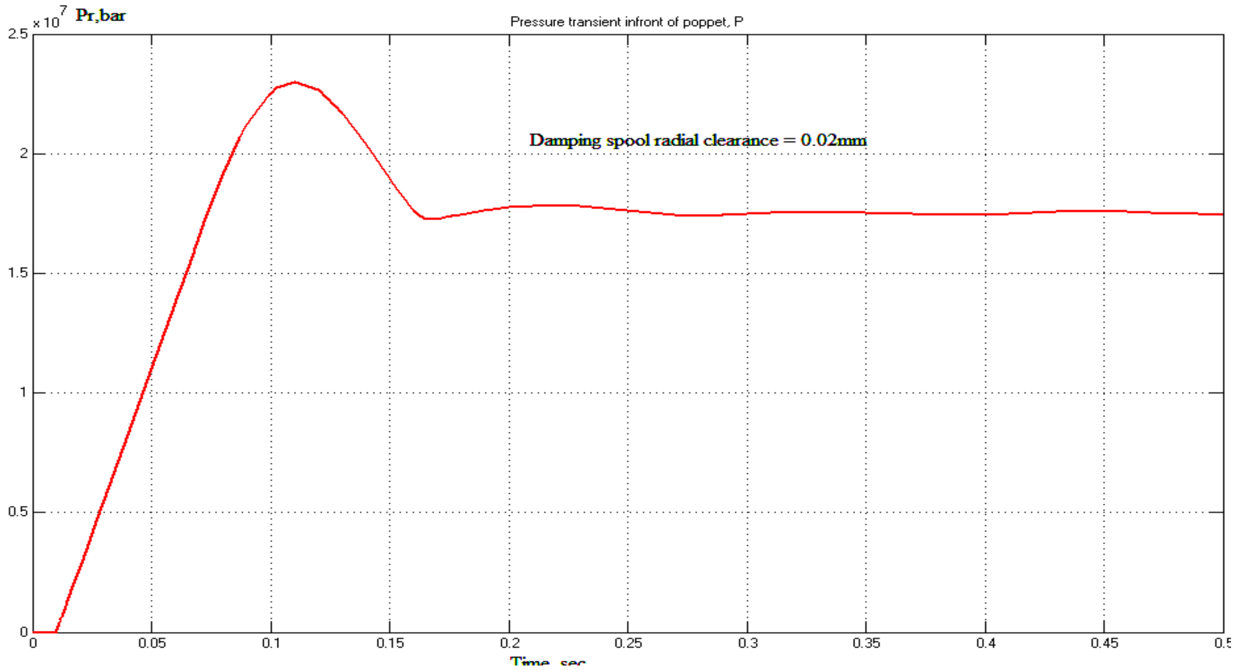


**Fig. 3.** Transient response of relief valve pressure for damping spool radial clearance of 0.02 mm

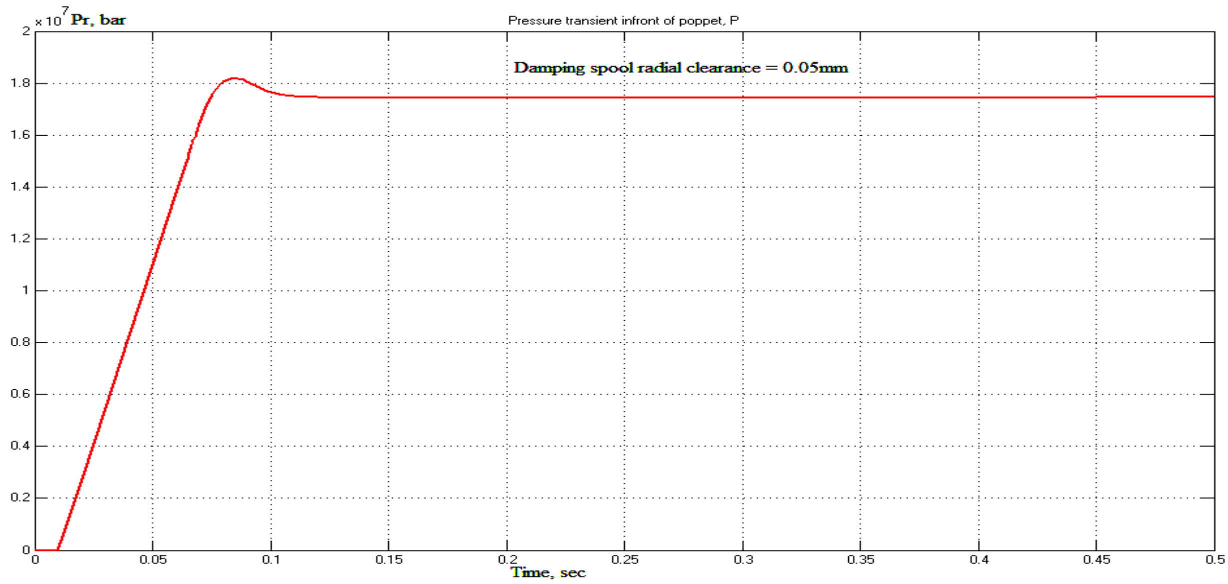


**Fig. 4.** Transient response of relief valve pressure for damping spool radial clearance of 0.05 mm

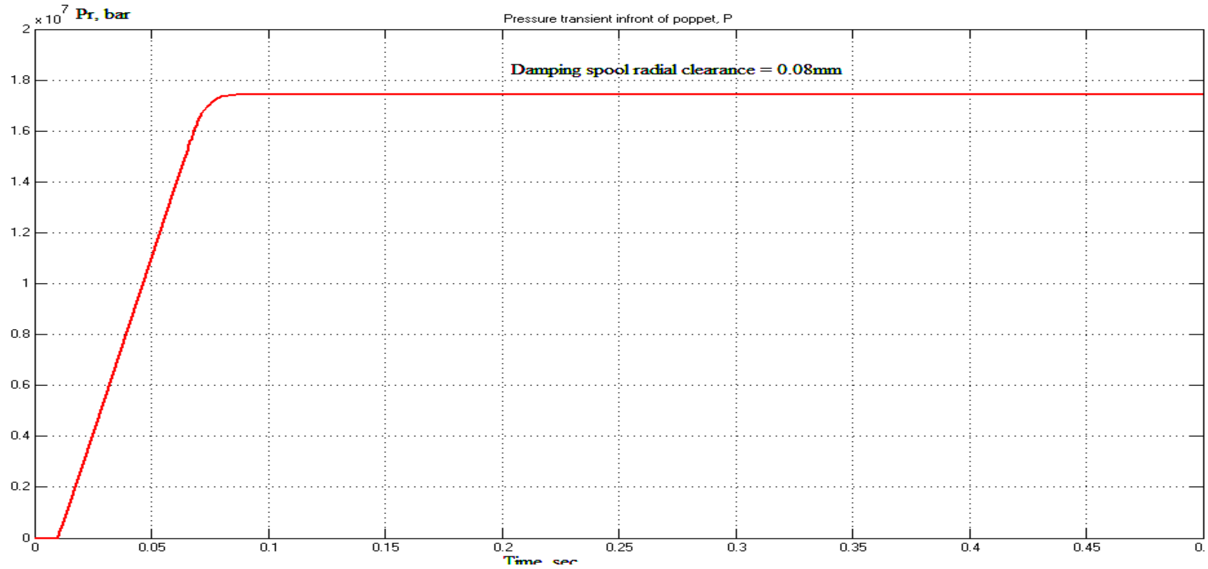


**Fig. 5.** Transient response of relief valve pressure for damping spool radial clearance of 0.08 mm

**TABLE 1.** Response

| Radial clearance, mm | Overshoot, bar | Peak time $T_o$, ms | Rise time $T_r$, ms | Settling time $T_s$, ms |
|---|---|---|---|---|
| 0.02 | 55 | 100 | 65 | 200 |
| 0.05 | 8 | 84 | 50 | 100 |
| 0.08 | zero | zero | 50 | 90 |

The simulation results show that the radial clearance has a significant effect on the valve response. For smaller radial clearances, the flow rate into the damping spool chamber is throttled and the pressure building in this chamber is delayed. The poppet takes a longer time to open, which results in greater pressure overshoot. For higher radial clearance of 0.08 mm, the overshoot is not observed and also rise time is less.

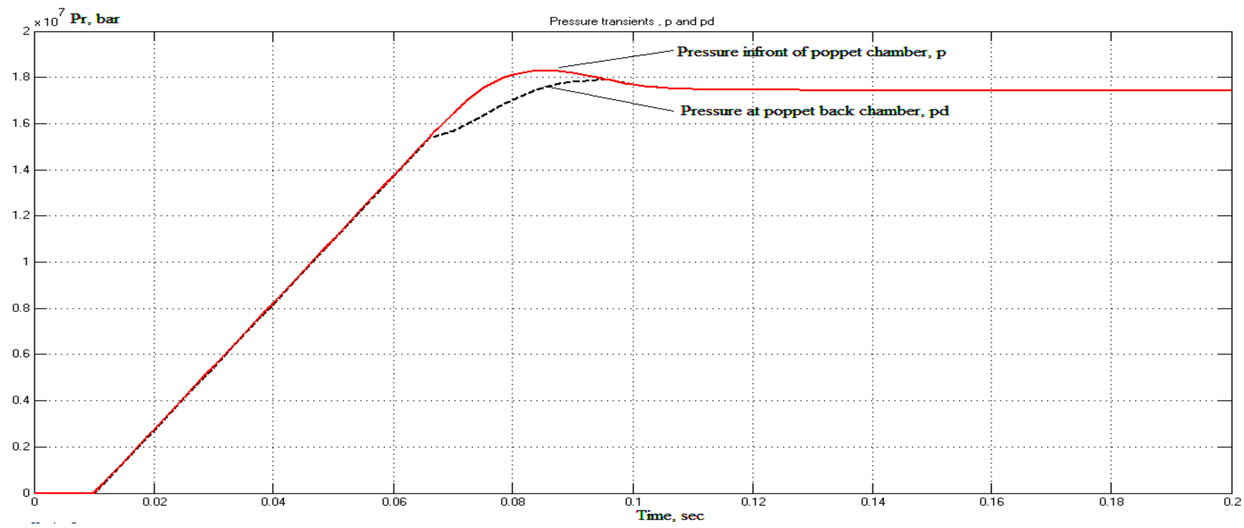


**Fig. 6.** Pressure transient curves of poppet front chamber and back chamber for damping radial clearance of 0.05 mm.

It is observed that pressure in front of poppet chamber and poppet back chamber remain same. It indicates that, for further analysis chambers can be considered as a single poppet chamber.

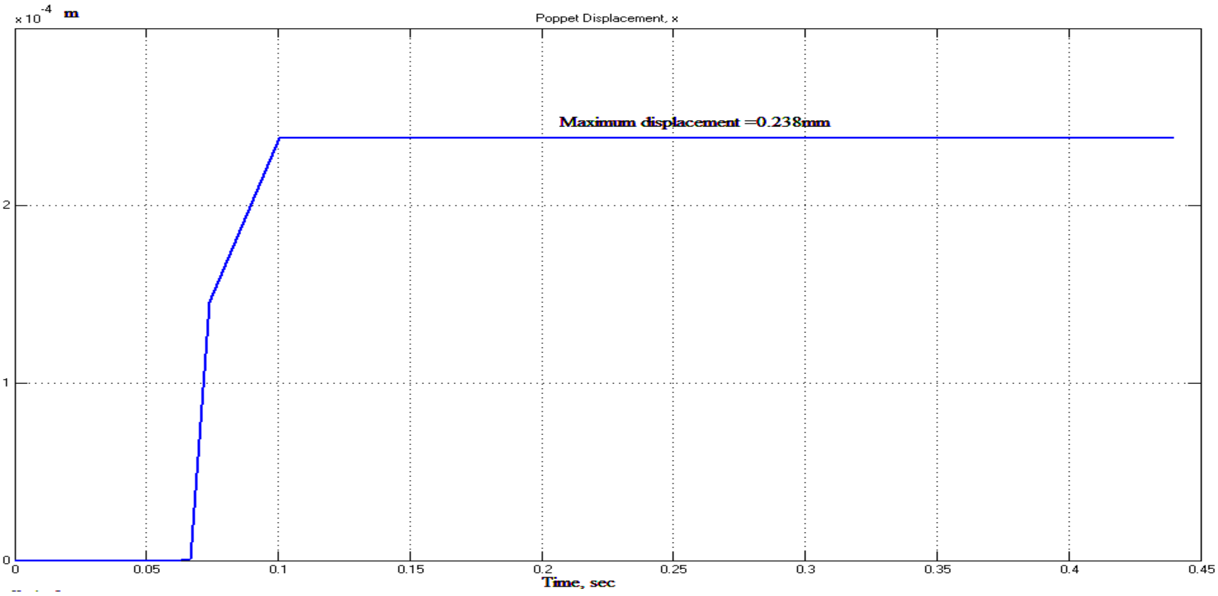


**Fig. 7.** Poppet displacement response

Referring to the above displacement curve, it is observed that valve gets opened after 55 ms and takes 35 ms to open completely to settled down to the final displacement of 0.238 mm.

## 6. Conclusion

Considering the transient response curves, it indicates that the poppet relief valve with damping spool has an excellent ability to reduce the pressure oscillations in the steady-state part of the response. In this study pressure oscillations not appear in the steady state part.
An important advantage of this simulation model is that its simplicity allows it to be used for a wider variety of system parameters. This model can be used in predicting application trends that may occur under various operating conditions, some of which may be difficult to create experimentally.

### Acknowledgment

I am very much thankful to the Directors of Polyhydron Pvt Ltd Belagavi, Karnataka India, for providing the technical assistance and resources to carry out research work on pressure relief valves.

NOMENCLATURE

$A_d$=Damping spool area, $m^2$
$A_{P=}$Poppet area subjected to pressure, $m^2$
$A_{po}$=Throttle area, $m^2$
β =Bulk modulus of oil, Pa
c =Radial clearance of the damping spool, m
$C_d$=Discharge coefficient
$D_d$=Damping spool diameter, m
f =Equivalent spring structural damping and piston friction coefficient, Ns/m
$F_s$=Seat reaction force, N
$K_{rv}$=Spring stiffness, N/m
$k_s$=Equivalent seat material stiffness, N/m
L =Damping spool length, m
m =Mass of the moving parts, kg
P =Valve inlet pressure, Pa
$P_d$=Pressure in the damping chamber, Pa
$P_T$=Return pressure, Pa

**References**

[1] Kenji Suzuki and Eizo Urata, "*Dynamic characteristics of a direct-pressure sensing water hydraulic relief valve",* Proceedings of the 6th JFPS International Symposium on Fluid Power, November 7-10, 2005, Tsukuba;

[2] Dr. Ing T. Hong, P.E , Dr. E. C. Fitch, P.E , "*Hydraulic System Design using the Static Performance Graph"* , Hydraulic System Modeling and Simulation by E. C. Fitch and I. T. Hong. Copyright © 2000 by BarDyne, Inc., Stillwater, Oklahoma USA;

[3] Eizo Urata, "*Thrust of Poppet Valve*", The Japan Society of Mechanical Engineers, vol.12. No.53, 1969;

[4] K. Dasgupta and R. Karmakar, *"Dynamic Analysis of Pilot Operated Pressure Relief valve"* Simulation Modelling Practice and Theory**,** Elsevier, December 2001;

[5] Sasko S. Dimitrov, *"Transient response of a pilot operated pressure relief valve with compensating control piston",* Technical university of Sofia, Bulgaria, Annals of Faculty Engineering Hunedoara-International Journal of Engineering, February 2014;

[6] Sung-Hwan Park, "*Design and performance characteristic analysis of servo valve-type water hydraulic poppet valve*", Journal of Mechanical Science and Technology 23 (2009) 2468~2478;

[7] John Mahrenholz, John Lumkes Jr., "*Analytical Coupled Modeling and Model Validation of Hydraulic On/Off Valves*", Department of Agricultural and Biological Engineering, Purdue University, West Lafayette, IN 47907;

[8] Gabor Licsko, Alan Champneys, Csaba Hos, "*Dynamical Analysis of a Hydraulic Pressure Relief Valve* ", Proceedings of the World Congress on Engineering 2009, Vol II WCE 2009, July 1 - 3, 2009, London, U.K.;

[9] Jonny Carlos da Silva and Nelson Back, "*Shaping the Process of Fluid Power System Design Applying an Expert System*", Research in Engineering Design (2000)12:8–17 2000 Springer-Verlag London Limited;

[10] K. Dasgupta, R. Karmakar, *"Modelling and dynamics of single-stage pressure relief valve with directional damping*" , Simulation Modelling Practice and Theory 10 (2002) 51–67, 1569-190X/02/$ - see front matter 2002 Elsevier Science;

[11] System simulation, The Math Works Inc.: Sim Hydraulics User's Guide, Version 1.2, The Math Works Inc., and Natick, MA. www.mathworks.com;

[12] Sasko Dimitrov, Simeon Simeonov, Slavco Cvetkov, "*Static Characteristics of the Orifices in a Pilot Operated Pressure Relief Valve*", Hidraulica (No 2/2015), ISSN 1453-7303;

[13] B. V. Hubballi, V. B. Sondur, "*Experimental Investigation on Transient response of Two Stage Pressure Relief Valve*", Hidraulica (03/2015), ISSN 1453-7303;

[14] John Watton, "Fundamentals of Fluid Power Control", Cambridge University Press. 2009;

[15] www.polyhydron.com /Pressure Control Valves.

# Considerations on Flow Regeneration Circuits and Hydraulic Motors Speed Variation at Constant Flow

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***Abstract:*** *The material shows a brief summary of simple practical applications of hydraulic drive systems that work on three principles: technology of regeneration of hydraulic motors feed flow, variation in steps of hydraulic motors speed at constant flow and control of vertical displacement speed of inertia loads. These applications have good energy efficiency, low price, high reliability and they are used to drive mechanisms that operate in several working phases, characterized by constant speeds within each phase, but different from one phase to another.*

***Keywords:*** *flow regeneration, speed variation at constant flow, control of vertical displacement of inertia loads*

## 1. Introduction

An important method of increasing, at low cost, energy efficiency of hydraulic drive systems is to reduce the capacity of pumping systems, due to the use of flow "regeneration circuits" for working phases of mechanisms with high speeds and low loads. These circuits allow the transferring of flow, which in a "classical" way would circulate towards the tank, from the passive chambers of hydraulic motors to the active chambers of motors, which are powered by pumping systems [1]. In the case of a differential hydraulic cylinder, for instance, by using a regeneration circuit, established between rod chamber and piston chamber in order to obtain quick advance with no load, the pump feeds piston chamber with a lower flow, equal to the difference between the pump flow without regeneration circuit and the regeneration circuit flow.
Other methods of increasing, at low costs, energy efficiency of hydraulic drive systems [2] are: achieving three speed steps, at constant flow, of a mechanism driven by a differential hydraulic cylinder; achieving equal speed rates, in both directions, for a hydraulic cylinder fed at constant flow; achieving constant speed rates at variable load of hydraulic motors; control of vertical descent speed of inertia loads.
The described solutions represent a much simpler and cheaper alternative to the use of linear or rotary hydraulic drive servo-systems [3].

## 2. Circuits of flow regeneration

Advance high or low speed movements of some presses or mechanisms can be performed with small cylinders through the flow regeneration technique.
In regeneration or *differential* circuits, the fluid in the rod chamber of a differential cylinder is transferred to the piston chamber during the advance stroke, while in normal applications the fluid *regeneration* is addressed to the tank. In this way, according to the flow regeneration technique, Figure 1, the flow from the rod chamber is added to the flow delivered by the pump in the piston chamber. As a result, the piston speed increases significantly.

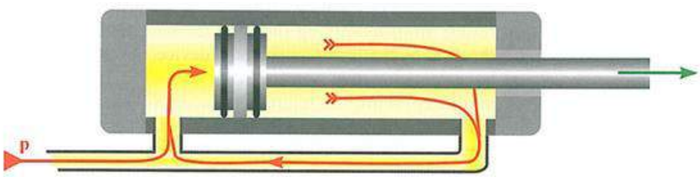



**Fig. 1.** Flow regeneration technique

At first glance, one might think that pump flow blocks the piston movement, because the fluid delivered by the pump is "pushed" over the two faces of the piston. In fact, the forces acting on the faces of a differential cylinder piston have different surfaces, because the rod area reduces effective surface of the piston face in rod chamber. *Regeneration technique* does not apply in the case of cylinders for presses or double rod cylinders.
Regeneration circuit effect can be demonstrated by the following simple sizing calculation:
There is given a piston with *2:1* ratio of areas. If the main area is equal to $S_1 = 150\ cm^2$, then the other area $S_2$ and rod area $S_s$ measure $75\ cm^2$.

$$S_1\text{-}S_2 = Ss \text{ or } Ss = S_1\text{-}S_2 \quad (1)$$

With a stroke *h* of *50 cm*, maximum capacity $V_1$ and minimum capacity $V_2$ of cylinder chambers is

$$V_1 = S_1 \cdot h = 150 \cdot 50 = 7500\ cm^3 = 7.5\ l \quad (2)$$

respectively

$$V_2 = S_2 \cdot h = 75 \cdot 50 = 3750\ cm^3 = 3.75\ l \quad (3)$$

If the pump delivers a flow *Q* of *10 l/min* and is connected to the main chamber of the cylinder, in standard version the rod needs *45* seconds to travel the entire stroke.

$$t_{normal} = V_1 / Q = 7.5 / 10 = 0.75\ min \cdot 60 = 45\ s \quad (4)$$

For the case in which the cylinder is connected in the version with regeneration circuit, the fluid from the chamber $V_2$ enters the chamber $V_1$, and the rod reaches stroke end in *22.5* seconds (as long as there are no pressure drops that exceed the adjusted pressure of discharge valve).

$$T_{regenerative} = V_2 / Q = 3.75 / 10 = 0.375\ min \cdot 60 = 22.5\ s \quad (5)$$
$$t_{total} = t_{normal} - t_{regenerative} = 45 - 22.5 = 22.5\ s \quad (6)$$

A higher discharge speed can be obtained by reducing the diameter of the rod (if the peak loads allow): for example, with a ratio of the piston surfaces *4:1*, the rod has a flat surface $S_2$ of $37.5\ cm^2$ and needs *11.4* seconds to travel the entire stroke:

$$S_2 = S_1\text{-}S_2 = 150 - 37.5 = 112.5\ cm^2 \quad (7)$$
$$V_2 = S_2 \cdot h = 112.5 \cdot 50 = 5625\ cm^3 = 5{,}625 l \quad (8)$$
$$t_{normal} = V_2 / Q = 5.625 / 10 = 0.56\ min \cdot 60 = 33.6\ s \quad (9)$$
$$T_{total} = t_{normal} - t_{regenerative} = 45 - 33.6 = 11.4\ s \quad (10)$$

A speed increase implies a reduction of the exerted force. The fluid flowing from one chamber of the differential cylinder to the other, through a regeneration circuit, reduces force resultant on the cylinder piston. If the safety valve is adjusted to 100 bar and if kept the same data used for a piston with a ratio of *4:1*, the forces $F_1$ on $S_1$ and $F_2$ on $S_2$ will be:

$$F_1 = p \cdot S_1 = 100 \cdot 150 = 15000\ daN \quad (11)$$

$$F_2 = p \cdot S_2 = 100 \cdot 112.5 = 11250\ daN \quad (12)$$

Since $F_2$ is opposed to $F_1$, then total force $F_t$ is:

$$F_t = F_1 - F_2 = 15000 - 11250 = 3750\ daN \quad (13)$$

Thrust on the annular surface of the piston, inside rod chamber, is compensated by thrust on the other frontal surface of the piston, minus rod surface; as a result, the force exerted in a regeneration circuit is equivalent to the force on the rod surface $S_s$.

$$F_t = p \cdot S_s = 100 \cdot 37.5 = 3750\ daN \quad (14)$$

**2.1. Check valves for flow regeneration circuits**

Hydraulic cylinders fitted with a regeneration circuit usually work in three main phases: a rapid initial stroke, during which regeneration circuit functions, a terminal extraction phase, subjected to a maximum force and the return phase. In normal conditions, they obviously require quarter phase for resting. One 4/3 directional control valve, Figure 2, does not allow these conditions, unless give up one of these four phases.

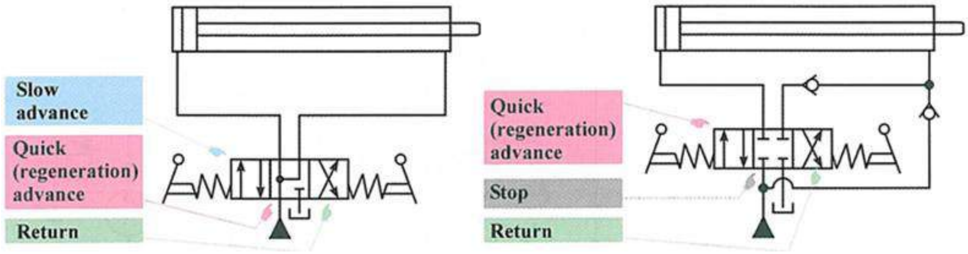



**Fig. 2.** Regeneration circuits with 4/3 directional control valves:
(a)=slow advance + quick advance (regeneration) + return; (b)=quick advance (regeneration) + stop + return

The most rational solution is given by a 6/4 directional control valve, Figure 3; it is possible to adjust the quick or slow advance movement, through a directional control valve manually driven. The size of the check valve does not depend only on the pump flow; it should be sized to pump flow plus regeneration flow.

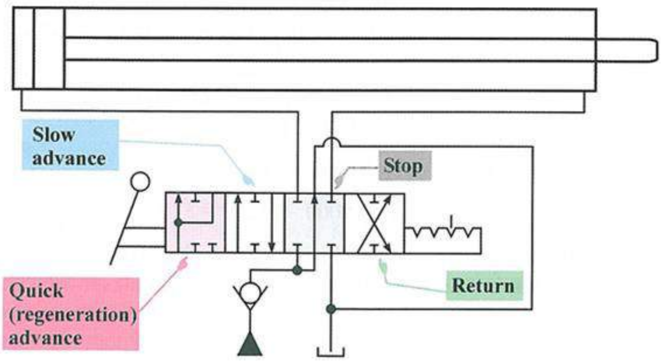



**Fig. 3.** Regeneration circuits with 6/4 directional control valves:
slow advance + quick advance (regeneration) and stop + return

## 2.2. Regeneration circuits with automatic slow or quick advance

Machines' operators cannot determine exactly the moment when they should turn from the quick regeneration advance to the completely slow regeneration advance. For this reason, the system is automated through a sequence valve, located between the 4/3 directional control valve and the hydraulic cylinder, Figure 4. The solution represents a much simpler and cheaper alternative to use of a linear hydraulic drive servo-system [3].

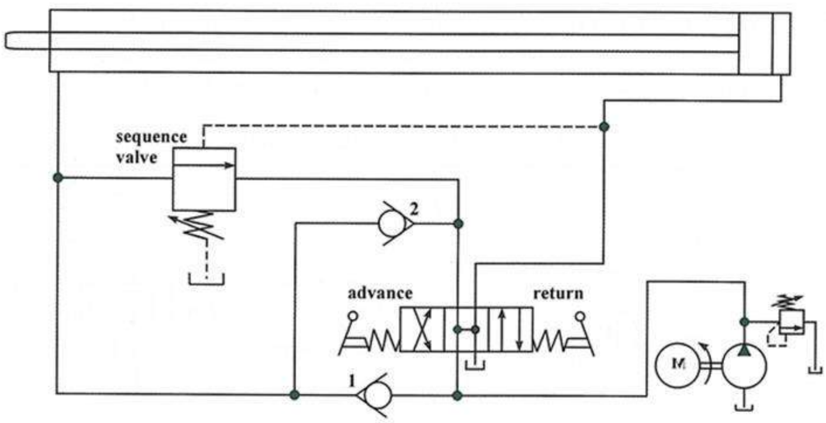


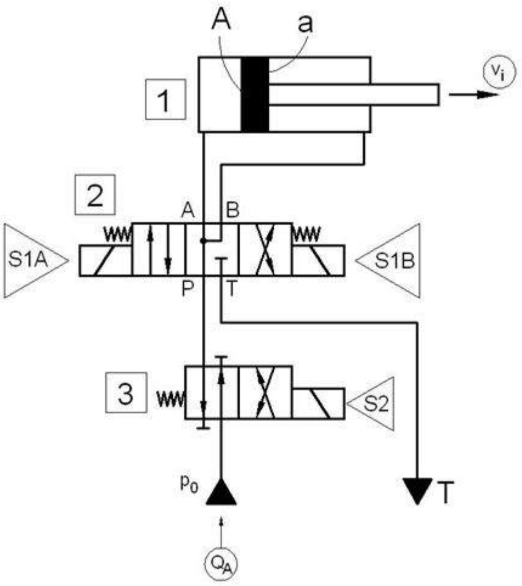

**Fig. 4.** Regeneration circuit with sequence valve

When the directional control valve is in the "advance" position, the quick advance stroke is supported by regeneration system; the fluid in the rod chamber is pushed through the check valve *1* and directional control valve to the piston chamber. Once there is developed a higher load on the rod, sequence valve normally closed, adjusted to that load, receives the pilot signal, opens and diverts the flow into the tank. In the regeneration phase, check valve *2* prevents the fluid from being conveyed to the tank and opens when the directional control valve is in the return position.

## 3. The speed variation in steps of hydraulic motors at constant flow

### 3.1. Drive of a differential hydraulic cylinder with three speed steps

Hydraulic drive diagram of a differential cylinder with three speed steps and ***stop*** phase is shown in Figure 5, and functioning cyclogram is shown in table 1.




**Components and functional role:**
*1* = differential hydraulic cylinder; receives the flow provided by pump and runs three displacement speeds;
*2* = 4/3 hydraulic directional control valve with electric drive; by switching determines the three speeds;
*3* = 4/2 hydraulic directional control valve, 2 blocked ports, electric control; starts or stops oil supply to the hydraulic cylinder;
$S_{1A}$, $S_{1B}$ = drive electromagnets acting by pushing the slide valve of the directional control valve *2*;
$S_2$= drive electromagnet acting by pushing the slide valve of directional control valve *3*;
*T* = tank.
**Functional parameters:**
A, a = large and small section of hydraulic cylinder piston;
$V_{i,\ i=1,2,3}$ = hydraulic cylinder speeds;
$Q_A$ = flow; $Q_A$ = const.

**Fig. 5.** Hydraulic drive diagram of a differential cylinder with three speed steps

**TABLE 1:** Functioning cyclogram of cylinder in Figure 5

| Drive / Phase | $S_{1A}$ | $S_{1B}$ | $S_2$ | Speed |
|---|---|---|---|---|
| **Quick advance** (with regeneration circuit) | - | - | + | $v_2 = \dfrac{Q_A}{A-a}$ |
| **Slow advance** | + | - | + | $v_1 = \dfrac{Q_A}{A}$ |
| **Retraction** | - | + | + | $v_3 = \dfrac{Q_A}{a}$ |
| **Stop** | - | - | - | 0 |

**Caption:** "-" = unpowered electromagnet; "+" = powered electromagnet.

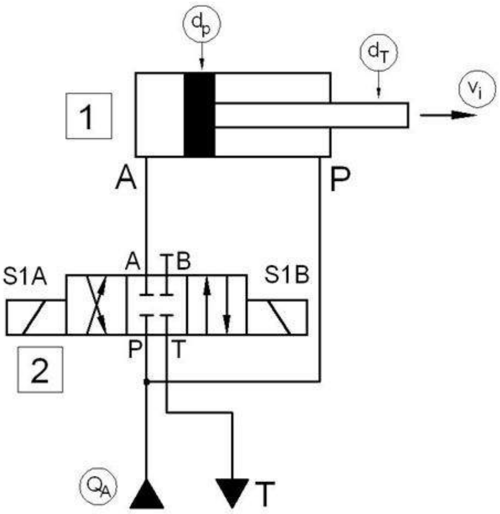
Between the three speeds of the cylinder, flow and piston sections the following relations are established:

$$v_2 = \frac{Q_A}{A-a}; \qquad v_1 = \frac{Q_A}{A}; \qquad v_3 = \frac{Q_A}{a}; \qquad a < A\ ; \quad v_1 < v_2 < v_3 \tag{15}$$

These hydraulic diagrams are used to drive mechanisms component of small and medium power machines, which do not require speed variations during the same phase, only between phases.

### 3.2. Drive of a differential hydraulic cylinder with equal speeds in both directions

Hydraulic drive diagram of a differential cylinder with equal speeds in both travel directions and ***stop*** phase is shown in Figure 6, and functioning cyclogram is shown in table 2.




**Components and functional role:**
*1* = differential hydraulic cylinder; receives the flow provided by pump and moves with equal speeds in both directions;
*2* = 4/3 hydraulic directional control valve with ***B*** blocked port, closed center, electric control; changes cylinder travel direction; starts and stops cylinder movement;
*$S_{1A}$, $S_{1B}$*= electromagnets driving the slide valve of the directional control valve;
*T* = tank;
*P, T, A, B* = the 4 ports (connections) of the directional control valve, namely: pressure, tank, consumer A, consumer B.
**Functional parameters**:
$d_p$ = piston diameter;
$d_T$ = rod diameter;
$Q_A$ = flow; $Q_A$ = const.

**Fig. 6.** Hydraulic drive diagram of a differential cylinder with equal speeds in both directions

**TABLE 2**: Functioning cyclogram of cylinder in Figure 6

| Drive / Phase | $S_{1A}$ | $S_{1B}$ | Speed |
|---|---|---|---|
| **Left movement** | + | - | $v_{stg} = \dfrac{Q_A}{A-(A-a)}$ |
| **Right movement** (with regeneration circuit) | - | + | $v_{dr} = \dfrac{Q_A}{a}$ |
| **Stop** | - | - | 0 |

**Caption:** "-" unpowered electromagnet; "+" powered electromagnet

If between piston diameter and rod diameter we have the relation:

$$d_p = d_T\sqrt{2} \tag{16}$$

then the total and annular areas of the piston will be given by the relations:

$$A = \frac{\pi d_p^2}{4};\; a = \frac{\pi d_T^2}{4}; \tag{17}$$

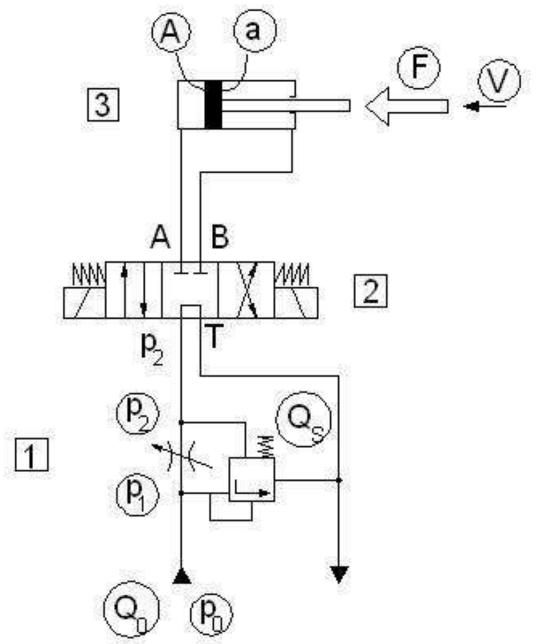
annular area of the piston being practically equal to area of the section rod, and the movement speeds to left/ right will be equal, respectively:

$$v_{stg} = \frac{Q_A}{A-(A-a)}; \qquad v_{dr} = \frac{Q_A}{a}; \qquad v_{dr} = v_{stg} \tag{18}$$

This type of driving can be found in mechanisms with oscillating motion in the structure of small power machinery.

### 3.3. Drive of a constant speed hydraulic motor at variable load

A linear hydraulic motor (cylinder) or rotary hydraulic motor can be actuated with constant speed or rotational constant speed, regardless of the force variation (at cylinder) or torque variation (at rotary motor) using a three-port flow regulator, placed between the constant flow hydraulic source and hydraulic directional control valve. Hydraulic drive diagram is shown in Figure 7.




**Components and functional role:**

1. Three-port flow regulator; maintains constant flow and speed regardless of load variation (force, at cylinder, respectively torque , at rotary hydraulic motor);
2. 4/3 hydraulic directional control valve with electric control; sets the travel direction, starts and stops hydraulic motor;
3. Differential hydraulic cylinder (or rotary motor); receives flow and performs linear or rotary motion.

**Functional parameters**:
*A, a (q)* = cylinder sections (or geometric volume);
*F (M)* = driving force (or torque);
$p_1$, $p_2$ = pressure drop across regulator;
$p_0$ = nominal pressure;
$Q_0$ = full flow;
$Q_S$ = flow discharged through regulator valve;
*V (n)* = velocity (or rotational speed)

**Fig. 7.** Hydraulic drive diagram with three-port flow regulator of a differential cylinder

Between functional parameters there are established the following relations:
The flow on A and B consumer circuits of directional control valve are given by the relation:

$$Q_{A,B} = K_{AB}\sqrt{p_1 - p_2} \quad \begin{cases} \mathrm{p}_1 > p_2 \\ K_{AB} = \alpha_D \cdot A_0\sqrt{2/\rho} \end{cases} \tag{19}$$

where $K_{A,B}$ is a constant which takes into account a coefficient of losses by friction $\alpha_D$, flowing area $A_0$, and working fluid (hydraulic oil) density $\rho$ , and $Q_{AB} = Q_0 - Q_S$. Stationary characteristic $Q = f(\Delta p)$ is represented in Figure 8.

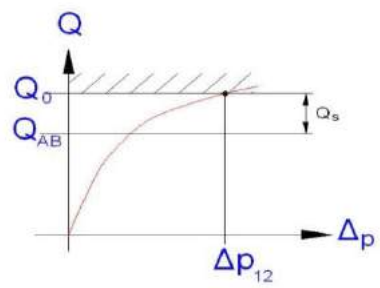



**Fig. 8.** Characteristic Q=f(Δp)

Relations between force/ torque and pressure, respectively speed/ rotational speed and flow rate are given by the expressions:

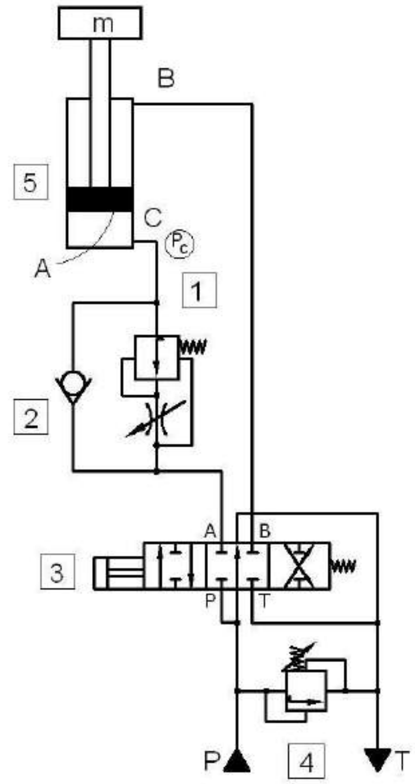
$$p_2 = \frac{F}{A}; \qquad \left( sau\, \mathrm{p}_2 \cong \frac{M}{q} \right); \quad v = \frac{Q_{AB}}{A} \left( \mathrm{sau\ n} = \frac{\mathrm{Q_{AB}}}{\mathrm{q}} \right) \tag{20}$$

This application is used to drive mechanisms component of small power machinery so that efficiency does not decrease below 65%.

**4. Linearity of hydraulic motor speed under the influence of load**

**4.1. Control of descendent speed of a vertical load using the two-port flow regulator**

Hydraulic drive diagram for lowering a vertical load at constant speed is shown in Figure 9.




**Components and functional role:**
1 = two-port flow regulator; maintains constant flow and speed regardless of load variation;
2 = bypass check valve; short -circuits the flow regulator in the load lifting phase;
3 = 6/3 distribution section of a battery directional control valve; reverses the travel direction of the hydraulic motor; starts / stops the hydraulic motor;
4 = input section of 6/3 battery directional control valve; contains safety valve of drive system;
5 = differential hydraulic cylinder (can be replaced with rotary hydraulic motor).
**Functional parameters:**
$P_c$ = pressure upstream of regulator;
$Q_c$ = flow upstream of regulator;
$\Delta p_{CA}$ = load loss in regulator;
$\Delta p_{AT}$ = load loss through directional control valve ($\Delta p_{AT} \leq 8$ bar);
$\Delta p_{lin}$ = load loss on paths;
*m* = lowered / lifted mass corresponding to the load;
*A* = cylinder piston section;
*P* = connection (path) P (pump) of the directional control valve;
*T* = connection T (tank or reservoir) of the directional control valve;
*A, B* = directional control valve consumer connections;
*C* = connection of the cylinder in piston chamber.

**Fig. 9.** Hydraulic drive diagram with two-port flow regulator of a vertical differential cylinder

To lower the load, directional control valve is switched so as there are created *P→B* and *A→T* connections. The flow on C connection circuit of the cylinder is given by the relation: $Q_C = K_C\sqrt{P_C}$

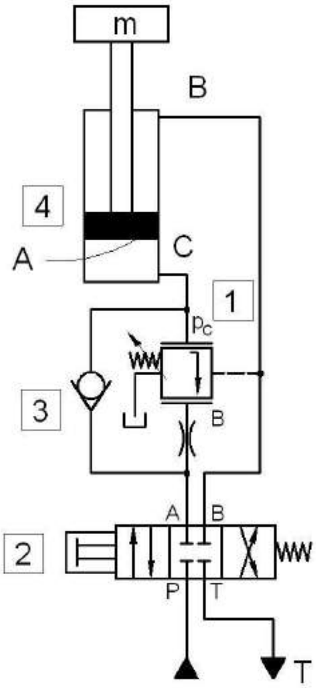
where $P_c = \frac{mg}{A} - \Delta p_{CA} - \left(\Delta p_{AT} + \Delta p_{lin}\right)$ (21)

This diagram is used to drive the mechanisms for arm and hook lifting at motor cranes for small and medium loads.

**4.2. Control of descent of a vertical load through the speed limiting valve with external control**

Hydraulic drive diagram for controlling descendent speed of a vertical load through a speed limiting valve with external control is shown in Figure 10.
The role of the two-port flow regulator is taken by a pressure limiting valve, with external control, acting as a valve that limits maximum flow which can cross the hydraulic resistance located upstream of the valve. In load lifting phase, the limiting valve and hydraulic resistance are short-circuited by a bypass check valve.

The directional control valve is so connected that there are created *A →T* and *p → B* connections.




**Components and functional role:**

1 = pressure limiting valve with external control; limits maximum pressure on C connection route of the hydraulic cylinder, respectively Δp by fixed hydraulic resistance, respectively flow in the outlet circuit of the piston chamber, respectively velocity of load lowering;

2 = 4/3 hydraulic directional control valve; changes the travel direction of the cylinder; starts / stops the hydraulic cylinder;

3 = bypass valve; short-circuits the hydraulic resistance and valve in the load lifting phase;

4 = differential hydraulic cylinder; receives flow and lifts / lowers the load.

**Functional parameters:**

$p_C$ = pressure upstream of the valve;

$Q_c$ = flow through the valve;

$p_A$ = pressure downstream of the valve;

$p_B$ = external control pressure;

$m$ = mass lowered / lifted corresponding to the load;

$A$ = cylinder piston section;

$\Delta p$ = load loss on circuits.

**Fig. 10.** Hydraulic drive diagram of a vertical differential cylinder with speed limiting valve

Flow through the valve $Q_c = k\sqrt{p_1 - p_2}$ , where $\begin{cases} p_1 = \dfrac{mg}{A} \\ p_2 = \Delta p_{AT} + \Delta p_{lin} \end{cases}$ (22)

This diagram is used as well to drive the mechanisms for arm and hook lifting at motor cranes for small and medium loads.

## 5. Canceling of the displacement speed under the influence of load

In the case of vertical displacement of a variable inertia load, upon accidental exceed of its maximum allowed value, during lifting or lowering phase, the load movement is canceled due to the opening of a safety valve, functioning as a shock valve, mounted in bypass on supply/discharge circuit of the hydraulic cylinder piston chamber.

Maintaining on an intermediate position of inertia load, located between the stroke ends of the hydraulic cylinder, is done in the center position of the 4/3 directional control valve, by means of an unlockable check valve fitted between the cylinder and connection A of 4/3 directional control valve.

The same shock valve does not allow lifting or lowering of inertia load, from an intermediate stationary position on the cylinder stroke (corresponding to the central position of the 4/3 hydraulic directional control valve) during exceeding its maximum allowed value.

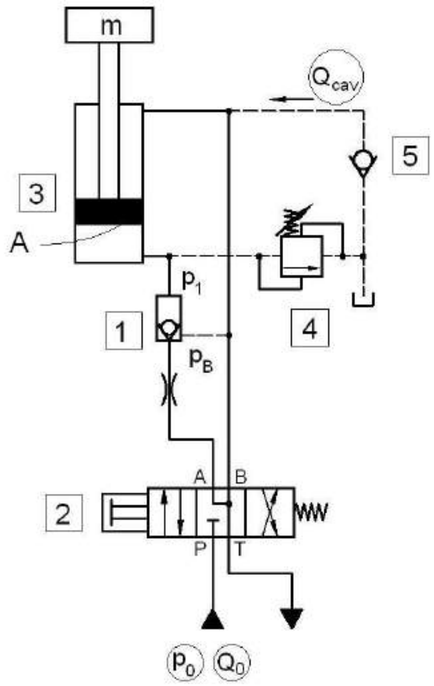
Hydraulic drive diagram of vertical displacement of inertia load, with movement canceling at a maximum allowed value of load, is shown in Figure 11.

Relations between pressures are:

$p_1(t) = \begin{cases} \cong p_0 \\ (1.25...1.35)p_0 \end{cases}$ ; $p_B$=(0.5...0.9)$p_0$ ; $p_{1\max} = \dfrac{mg}{A} < K \cdot p_0$; $K = 1.25...1.35$ (23)

The flow of displaced working fluid is given by the relation:

$$Q_{cav} = \frac{V}{\beta} \cdot \frac{dp}{dt} \begin{cases} \dfrac{dp}{dt} = \dfrac{p_{1\max} - p_0}{\Delta t} = \dfrac{(K-1)p_{nom}}{\Delta t} \\ V = \text{fluid volume in the circuit} \end{cases} \quad (24)$$




**Components and functional role:**
1 = unlockable check valve; ensures maintenance on position of inertia load in the center position of the slide valve of the 4/3 directional control valve, and by its unlocking in the right position of the slide valve, the load lowering;
2 = 4/3 hydraulic directional control valve with slide valve, diagram 02; ensures change of travel direction and also start / stop of hydraulic cylinder;
3 = differential hydraulic cylinder (can also be used a rotary hydraulic motor ); ensures displacement of inertia load;
4 = shock valve; stops displacement of inertia load;
5 = anti-cavitation check valve; ensures filling of the cylinder rod chamber, during shock valve opening, in the center position of the slide valve of the directional control valve.
**Functional parameters:**
$p_0$, $Q_0$ = nominal pressure and flow;
$A$ = cylinder piston section;
$Q_{cav}$ = displaced flow; $m$ = inertia load;
$p_B$ = pilot pressure; $V$ = fluid volume in the circuit;
$\beta$ = compressibility of the working fluid.

**Fig. 11.** Hydraulic diagram of inertia load displacement with speed canceling under the influence of load

This application is used to drive the mechanisms for vertical lifting of inertia loads in technological equipment.

## 6. Conclusions

Flow regeneration circuits develop important applications for the efficient execution of fast or slow advance movements of differential hydraulic cylinders. With the help of sequence valves there can be developed a simple automation for passing from quick advance movement, with flow regeneration, to slow advance movement.
By use of a differential hydraulic cylinder and a flow regeneration circuit there can achieved three speed steps for cylinder rod, namely a fast advance, a slow advance and a return.
A linear or rotary hydraulic motor can be actuated with constant speed or constant rotational speed, regardless of load (force or torque) variation, if a three-port flow regulator is introduced in the drive diagram.
Control of vertical descent speed of inertia load can be achieved by use of a two-port flow regulator or an external control pressure regulating valve, located on the outlet circuit of a linear (differential cylinder) or rotary hydraulic motor.
Vertical displacement of an inertia load, by use of a linear or rotary hydraulic motor, is stopped at accidental increase in load beyond the allowed limit, by means of a shock valve.

**Acknowledgement**

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**References**

[1] ASSOFLUID handbook "Hydraulics in Industrial and Mobile applications", Printed in Italy by Grafice Parole s.r.l.-Brugherio (Milano);
[2] N. Vasiliu, D. Vasiliu, "Fluid Power Systems", Vol. I, Technical Publishing House, Bucharest, 2005;
[3] C. Cristescu, P. Drumea, D.I. Guta, C. Dumitrescu, P. Krevey, "Experimental research regarding the dynamic behavior of linear hydraulic motors in frequency domain", Hidraulica, no. 3-4/2011, ISSN 1453-7303, pp. 35-46.

# Optimization of Axial Wind Turbines Operation

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***Abstract:*** *The operation of horizontal axial wind turbines was diversified through their applications in aero-energetic of variable speed of rotation. These situations are investigated and optimized in function of wind velocity, as intensity. Results are useful for wind power plants Marga and Ciugud designed and realized by a group of professors from the Hydraulic Machinery Chair, Faculty of Mechanical Engineering, University "Politehnica" Timişoara.*

***Keywords:*** *Axial wind turbines, wind aggregates with variable speed of rotation, optimization of the operation in respect of wind velocity, aerodynamic profiles*

## List of symbols

P – aerodynamic power.
ω – runner angular velocity.
N – runner blade number.
ρ – density.
$C_P$ – lift coefficient
$C_R$ – drag coefficient
r – runner current radius.
β – angle between relative velocity and the opposite of rotation velocity.
l - blade profile chord.
v – wind velocity.
n - runner speed of rotation.
i – profile angle of attack.
R – runner radius.

## 1. Introduction

Axial wind and hydraulic machines controls its operation through the rotation of their blades around the blades axis. Today through the progress of the electro-energetic systems it is possible to vary the speed of rotation of these machines. The classic axial hydraulic machines are able to rotate in the same time the wicket gates and runner blades in such a mode to obtain the maximum efficiency [9]. The classic axial wind turbines are controlled through the runner blades rotation in function of the wind velocity for assuring a maximum power [10]. In this paper it is investigated a different control modality of an axial wind turbine with fixed blades which is optimized through accord the speed of rotation of the aggregate to the wind velocity.

## 2. Wind aggregate calculus at nominal regime of operation

5 kW power plant wind aggregates in accord with the Marga and Ciugud projects [1] are designed for a nominal wind velocity of $v_0$ = 8.5 m/s and a speed of rotation of $n_0$ = 120 rot/min (for constant speed of rotation) and $n_0$ = 60 – 120 rot/min (for variable speed of rotation). The Ciugud blades geometry through aerodynamic profiles at different radiuses and their stagger angles of the grids are puts in evidence graphical for the tip profile of the blade in Fig. 1 and Fig. 3.

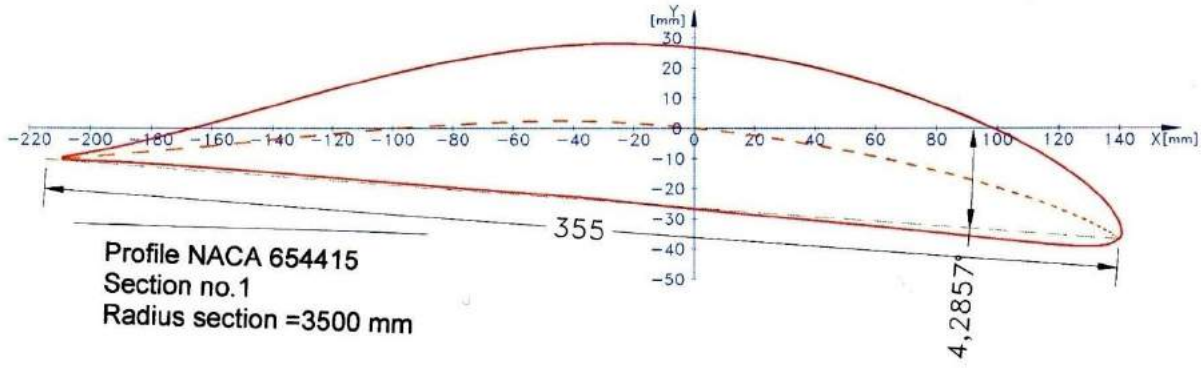


**Fig. 1.** Section through wind runner blade [1]

The investigation of the runner is with the two-dimensional blade theory based on fig. 2.

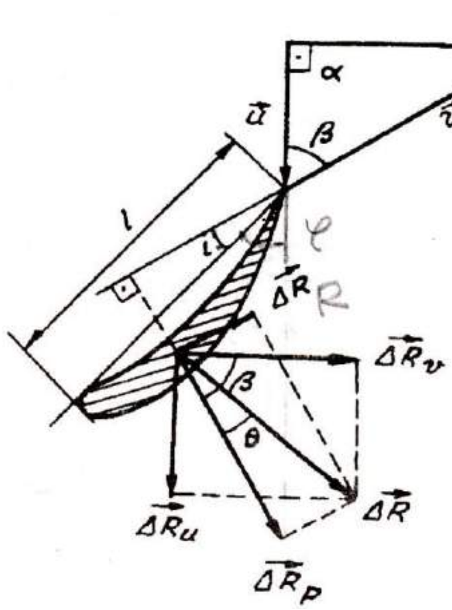

**Fig. 2.** Kinematic and dynamic of the elementary runner profile

There are calculated: rotation velocity, entrance angle of the wind on the runner and attack angle of the blade components profiles at different radiuses:

$$u = \omega \cdot r = \frac{\pi \cdot n}{30} \cdot r \tag{1}$$

$$\beta = arc\,tg\left(\frac{v_o}{u}\right) \tag{2}$$

$$i = \beta - \varphi \tag{3}$$

From the 28 elementary known sections of the wind runner blade it was chosen 3 representative sections for the hub, middle and tip of the blade. Data referring to blade chord "b" and the stagger angle of the profiles grid "φ" was extracted from [1]. Results are given in Table 1.

**TABLE 1**

| Nominal values $v_o$ = 8.5 m / s ; $n_o$ = 120 rot / min. | | | | | | |
|---|---|---|---|---|---|---|
| r | b | φ | Profile | u | β | i |
| m | m | grade | NACA | m /s | grade | grade |
| 3.5 | 0.355 | 4.2857 | 654415 | 43.982 | 10.938 | 6.6523 |
| 2.5 | 0.525 | 7.600 | 654418 | 31.416 | 15.140 | 7.54 |
| 1.5 | 0.695 | 15.3333 | 654421 | 18.850 | 24.272 | 8.9387 |

Analytic connection between the angle of attack of the profile in function of the blade radius is:

$$i = 11.99488 - 2.4207 \cdot r + 0.2555 \cdot r^2 \tag{4}$$

with the mean error : $\varepsilon_m = 4.6259 \cdot 10^{-18}$ . (5)

The rated regime rapidity is:

$$\lambda = \frac{u(at \quad r = 3.5\, m)}{v_o} = 5.174 \tag{6}$$

The value is smaller than the optimum value from literature $\lambda_{optim} = 6...8$ after [4].

### 3. Wind turbine operation at different values of the wind

For wind velocities between 5 and 9 m/s the results are in Table 2.

**TABLE 2**

| **Crt. no.** | **$v_o$** | **λ** | **r** | **b** | **φ** | **Profile** | **u** | **β** | **i** | **$C_P$** | **$C_R \cdot$ ·100** |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | m / s | - | m | m | ° | NACA | m / s | ° | ° | - | - |
| 1 | 5 | 8.796 | 3.5 | 0.355 | 4.2857 | 654415 | 43.982 | 6.4857 | 2.2 | 0.161 | 1.105 |
| 2 | | | 2.5 | 0.525 | 7.600 | 654418 | 31.416 | 9.0430 | 1.443 | 0.106 | 1.022 |
| 3 | | | 1.5 | 0.695 | 15.333 | 654421 | 18.850 | 14.8557 | -0.4776 | -0.035 | 0.099 |
| 4 | 6 | 7.330 | 3.5 | 0.355 | 4.2857 | 654415 | 43.982 | 7.7683 | 3.4826 | 0.255 | 1.337 |
| 5 | | | 2.5 | 0.525 | 7,600 | 654418 | 31.416 | 10.8125 | 3.712 | 0.271 | 1.390 |
| 6 | | | 1.5 | 0.695 | 15.333 | 654421 | 18.850 | 17.6564 | 2.3231 | 0.170 | 1.122 |
| 7 | 7 | 6.283 | 3.5 | 0.355 | 4.2857 | 654415 | 43.982 | 9.04312 | 4.7555 | 0.348 | 1.680 |
| 8 | | | 2.5 | 0.525 | 7.600 | 654418 | 31.416 | 12.5612 | 4.9612 | 0.363 | 1.746 |
| 9 | | | 1.5 | 0.695 | 15.333 | 654421 | 18.850 | 20.3726 | 5.0393 | 0.368 | 1.772 |
| 10 | 8 | 5.497 | 3.5 | 0.355 | 4.2857 | 654415 | 43.982 | 10.309 | 6.0233 | 0.440 | 2.134 |
| 11 | | | 2.5 | 0.525 | 7.600 | 654418 | 31.416 | 14.2866 | 6.866 | 0.502 | 2.498 |
| 12 | | | 1.5 | 0.695 | 15.333 | 654421 | 18.850 | 22.997 | 7.6637 | 0.560 | 2.888 |
| 13 | 9 | 4.887 | 3.5 | 0.355 | 4.2857 | 654415 | 43.982 | 11.565 | 7.2793 | 0.532 | 2.694 |
| 14 | | | 2.5 | 0.525 | 7.600 | 654418 | 31.416 | 15.986 | 8.386 | 0.613 | 3.279 |
| 15 | | | 1.5 | 0.695 | 15.333 | 654421 | 18.850 | 25.522 | 10.1887 | 0.745 | 4.414 |

From the Table 2 it is observed the large range of the attack angles and lift and drag coefficients. For the wind turbines with variable speed of rotation it is considered the minimum speed of rotation in Table 3.

**TABLE 3**

| Nominal values $v_o$ = 8.5 m / s ; $n_o$ = 60 rot / min. | | | | | | |
|---|---|---|---|---|---|---|
| r | b | φ | Profile | u | β | i |
| m | m | grade | NACA | m /s | grade | grade |
| 3.5 | 0.355 | 4.2857 | 654415 | 21.99 | 21.1335 | 16.8478 |
| 2.5 | 0.525 | 7.600 | 654418 | 15.7079 | 28.4197 | 20.8197 |
| 1.5 | 0.695 | 15.3333 | 654421 | 9.4248 | 42.0459 | 26.7126 |

Analytic connection between the angle of attack of the profile in function of the blade radius is:

$$i = 39.15383 - 9.7349 \cdot r + 0.9605 \cdot r^2 \tag{7}$$

with the mean error: $\varepsilon_m = 8.0954 \cdot 10^{-18}$ (8)

Comparative analysis, of the results about attack angles from Table 1 and 3 shows that speed of rotation decrease produce greater attack angles and the separation of the flow by blade's hub. Data from NACA catalog [6], offer characteristic curves of the used profiles, from which here there are presented only the curves for the most uploaded, in Fig. 3.

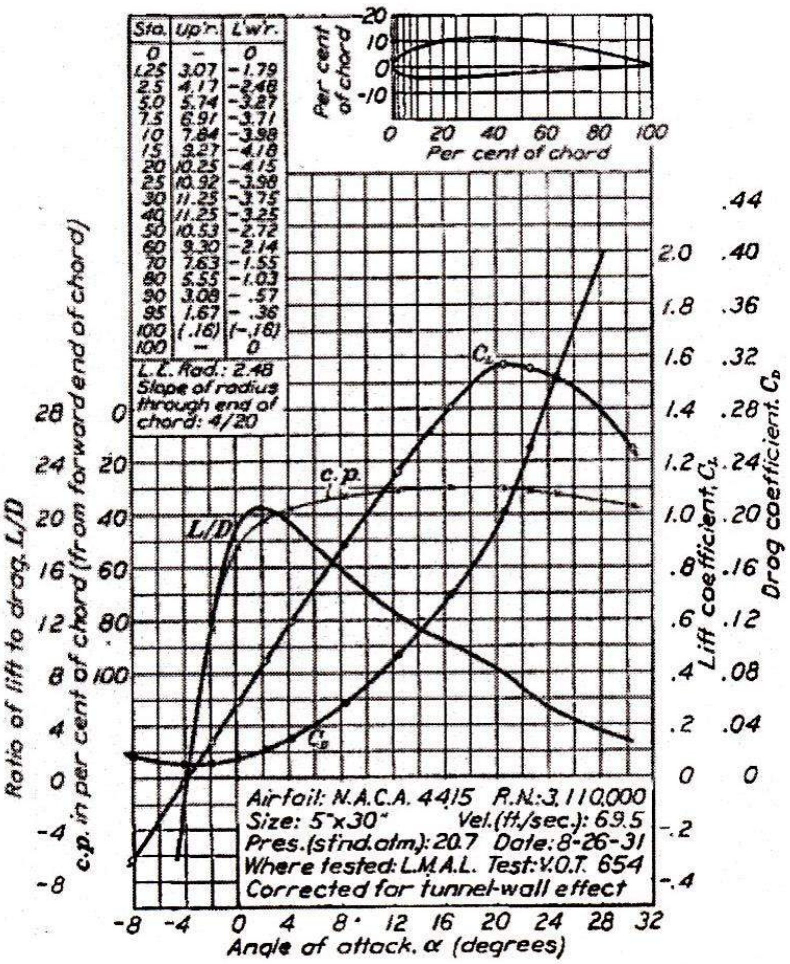


**Fig. 3.** Characteristic curves for the profile NACA 654415

## 4. Optimization of the operation regime

Considering the developed power in a wind turbine:

$$P = \frac{N \cdot \omega \cdot \rho}{2} \int_{r_0}^{R} \left\{ C_P(r) \cdot \sin\left[\beta(r)\right] - C_R(r) \cdot \cos(r) \right\} \cdot \left(\omega^2 \cdot r^2 + v^2\right) \cdot l(r) \cdot r\, dr \tag{9}$$

or

$$P = \frac{N \cdot \omega \cdot \rho}{2} \int_{r_0}^{R} \left[ C_P(r) \cdot v - C_R(r) \cdot \omega \cdot r \right] \cdot \sqrt{\omega^2 \cdot r^2 + v^2} \cdot l(r) \cdot r\, dr \tag{10}$$

Knowing the usual values of the parameters along the blade and that the maximum power is extracted by the tip of the blade and also that:

$$\frac{C_P}{C_R} \cong 20 \quad ; \quad \frac{\omega \cdot R}{v} \cong 5 \quad ; \tag{11}$$

With these values it is deduced from relation (10) that the higher power is obtained for the greater lift coefficient $C_P$. The power depends on a lot of factors and the lift coefficient is accepted for different safety degrees in two-dimensional models, against the non-stable operation. It is considered the tip profile of the blade because the outside zone of the blade transfers large part of the power. To the axial wind turbine with fixed blades and variable speed of rotation it was calculated through a program "trail and error" the speed of rotation values of the aggregate in function of the wind velocity for an angle of attack imposed, therefore an adequate power and a safety degree in respect of flow separation at normal operation or gusts. So it was established for the outside profile NACA 654415 in function of the wind velocity "v" ( in m/s) at the attack angle of i = $6^0$, the optimum speed of rotation of the aggregate (wind turbine) "n" (in rot/min):

$$n = 14.25 \cdot v + 0.625 \tag{12}$$

At i = $10^0$

$$n = 10.75 \cdot v - 0.125 \tag{13}$$

At i = $14^0$

$$n = 8.933 \cdot v - 0.666 \tag{14}$$

The speed of rotation values calculated with these relations, depending from the wind velocity are introduced in the automation device of the power plant and acts upon the exciting device of the electric variable speed generating system [2], [3].

## 5. Conclusions

a) Modern wind aggregate often operate with variable speed of rotation acting on the electric generator. The automation device has an anemometer as the entrance transducer.
b) Exploitation strategy of the wind power plant imposes the aggregate speed of rotation modification law in function of wind velocity with different degrees of confidence with one of the formulas (12), (13) or/and (14).
c) In function of the wished power and the assumed in respect of the separation of the flow from the runner's blades it is chosen one of the above mentioned relations.
d) All the conclusions obtained in this article are valuable in the hypothesis of zero azimuth (yaw) angle or after the stabilization of the aggregate in the wind direction.
e) The validity of the results obtained through the relations (12), (13) and (14) can be verified on the base of measurements made on the actual automated system of the wind power plant from Ciugud which works with a program of maximum power searching of the axial wind turbine.

**References**

[1] T. Miloş ,..., Project RO 0018 "Improvement of the structures and efficiency of small horizontal axis wind generators", EEA Grants 2009 – 2012;
[2] E. Hau, "Windkraftanlagen". Springer Verlag, Berlin, 2014;
[3] T. Burton ,.., "Wind Energy Handbook", John Wiley ed., Chichester, 1997;
[4] M. Bărglăzan, "Turbine hidraulice şi transmisii hidrodinamice", Ed. Politehnica, Timişoara, 2000;
[5] P. K. Chang, "Control of Flow Separation", Hemisphere Publ. Washington, 1978;
[6] Report 460, "The Characteristics of 78 Related Profiles Sections from Tests in the Variable Wind-tunnel", 1993;
[7] G. Yu, …, "An insight into the separate flow and stall delay for HAWT", Renewable Energy , vol. 36, Issue 1, Jan. 2011;
[8] ed. Sathyajith M.,…, "Advances in Wind Energy Conversion Technology", Springer, Berlin, 2011;
[9] T. Milos, M. O. Popoviciu, I. Bordeasu, R. Badarau, A. Bej, D. Bordeasu…"The 3D blade surface generation for Kaplan turbines using analytical methods and CAD techniques", Hidraulica no. 2/2013, pp. 65-74;
[10] M. Barglazan, I. Bordeasu…"Contribution to hydraulic turbines draft tube design", Hidraulica no. 2/2014, pp. 32-38.

# Considerations on Energy Losses in Hydraulic Drive Systems

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***Abstract:*** *The article presents some considerations on energy losses in hydrostatic drive systems, identifying the elements and specific places where flow and pressure losses occur. Some considerations are made on flow regimes and calculation relations are presented for assessing the pressure and flow losses; also on an example of hydrostatic drive system there is assessed energy dissipation and system efficiency is calculated. In the end, there are presented some modern methods and concepts for design of hydraulic diagrams used for the purpose of increasing energy efficiency of hydrostatic drive systems.*

***Keywords:*** *hydrostatic drive systems, energy efficiency, energy losses, pressure losses, flow losses*

## 1. Introduction

In recent years, hydrostatic drives have developed a lot and have penetrated into many areas. Technological facilities have become increasingly complicated and complex, and requirements on automation of their operation, on the one hand, and increased requirements regarding energy efficiency and productivity to increasingly higher levels, on the other hand, have imposed on the drive systems, and implicitly on the automation hydraulic equipment that are part of the former, new requirements and new performances. Among them the requirements with regard to increased operational reliability and accuracy, improved static and dynamic performance and, in particular, the **decrease in the consumption of energy** and materials are very important [1].
To achieve the required performance, there is a need not only for modern components with suitable static and dynamic parameters, but also a **modern concept** of the working diagram of a hydrostatic transmission. There also needs to be done a very careful analysis on mass and energy flow, identifying all critical points that may cause flow and pressure losses, **in order to increase energy efficiency** of the hydrostatic drive system.
Since a hydrostatic drive system is composed of various parts through which the hydraulic agent flows, it follows that the process of flowing is always accompanied by **load losses**. It is important, however, that these load (pressure) losses to be as small as possible, because the sum of all load losses in the system implicitly results in **unnecessary consumption** that the power supply of equipment has to cover in order to overcome those losses. Load loss manifests, eventually, in **dissipation of some of the energy of the conveying fluid**, because this energy is transformed into heat energy, which is taken up by the fluid, heating it. This process is more pronounced the higher the pressure drop on a system component is. Those presented before are phenomena that exist and they **always accompany the flowing phenomena of the working agent**, but the losses in the system should be as small as possible for the system to be a high efficiency one [2].
**Worldwide** there are noted intensive concerns with regard to developing hydrostatic drive systems **with low energy consumption.** Since in hydrostatic drive systems very high power is circulated, the energy consumption issue is particularly significant [1].

## 2. Assessment of local and linear pressure losses

It is known that in real (viscous) fluid flow via pipelines and equipment parts part of the total energy of the mass of fluid is used to overcome **viscous friction resistance forces**, turning irreversibly into heat. Another part of this energy is consumed when **turbulence phenomena occur**, when the flow velocity changes its direction and / or size (due to changes in the flowing section). These energy losses can be defined as **resistance mechanical work** due to the viscous and turbulent resistances of real fluids [1].

**Bernoulli's principle**, customized to the case of incompressible viscous fluids, leads to:

$$\Delta P_{tot} = \rho \cdot \Delta e_{tot} = (P_1 - P_2) + \frac{\rho}{2} \cdot (w_1^2 - w_2^2), \tag{1}$$

where the term $\Delta P_{tot}$ is total load loss on the section considered, and the term $\Delta e_{tot}$ is precisely **specific energy lost** by the fluid when it moves between two sections, 1 and 2, characterized by pressures p1 and p2 and flow velocities w1 and w2, ρ being fluid density. Relation (1) highlights the two **components of load loss**:

- **Linear** or **distributed component** $\Delta P_d$ , due to the fluid viscosity;
- **Local component**, $\Delta P_l$, due to turbulence, so:

$$\Delta P_{tot} = \Delta P_d + \Delta P_l \tag{2}$$

This division into linear and local losses is a conventional one, because, in reality, pressure losses are indivisible physically. Based on numerous studies and research, **it was agreed** that load loss to be related to the kinetic energy of moving fluid [1].

For this purpose, the strength or loss coefficient ξ has been defined.

$$\xi = \frac{\Delta e_{tot}}{e_c} = \frac{Specific\ energy\ dissipated}{Kinetic\ energy\ of\ mass\ flow\ unit}\ or\ \ \xi = \frac{m \cdot \Delta e_{tot}}{m \cdot \frac{w^2}{2}}, \tag{3}$$

And if density is constant:

$$\xi = \frac{\Delta P_{tot}}{p \cdot \frac{w^2}{2}} \tag{4}$$

According to the relation (4), **the coefficient ξ** can be defined as **the ratio of total load loss and dynamic pressure in the benchmark section.**

The above relation can be rewritten also as:

$$\Delta P_{tot} = \xi \cdot \rho \cdot \frac{w^2}{2} \tag{5}$$

In reality, the flow velocity through a narrowed section is not constant, therefore in the relation (5) it is considered the average speed of the fluid through the section considered, speed given by the relation: $w = q/S$.

In this case, **local load loss** can be written:

$$\Delta P_l = \xi \cdot \frac{1}{2} \cdot \rho \cdot (q/S)^2 \tag{6}$$

where ξ is a dimensionless numerical coefficient, dependent on the geometry of the hole, but sometimes also on the fluid, on its temperature and flow.

If one aims to express **linear / distributed load loss**, in a tube / pipe, it is obvious that its length must interfere, and the relation (6) will no longer be convenient.

In this case the following expression will be chosen:

$$\Delta P_d = \lambda \cdot (L/D) \cdot \frac{1}{2} \cdot \rho \cdot (q/S)^2 \tag{7}$$

where *L* is pipe length, *D* – its diameter (if the pipe has not a circular section, *D* is hydraulic diameter, defined as $D_h = 4 \cdot A/P_u$, where A is pipe section, and $P_u$ - perimeter wetted by the fluid), λ a dimensionless numerical coefficient dependent on the pipe, fluid and flow velocity.

If one takes into account **load losses in the transitional regime** as well, they are expressed through the practical relation [5]:

$$\Delta p_{din} = L_H \frac{dQ}{dt} \tag{8}$$

where:

$L_H = \frac{m}{A^2}\left[\frac{daNs^2}{cm^5}\right]$ is **hydraulic inertance** coefficient, and $m\left[\frac{daNs^2}{cm}\right]$ is mass volume of fluid that changes the flow velocity through the flowing section with the area A, expressed by Q.

If there is also considered the **concentrated capacitance** of a volume of fluid flowing through the section, they are calculated by the relation

$$\frac{1}{C_H}\int Q\,dt \tag{9}$$

In the above relation, $C_H\,[cm^5/daN]$ is the **coefficient of capacitance,** according to the following relation:

$$C_H = \frac{V}{E} \tag{10}$$

Based on the similarity hydraulic-electric, it can be considered that the hydraulic drive circuit *CHA*, between the pump *P* and the linear hydraulic motor *MHL*, shown in Figure 1, is similar to an equivalent electric circuit *CEE*, shown in Figure 2.

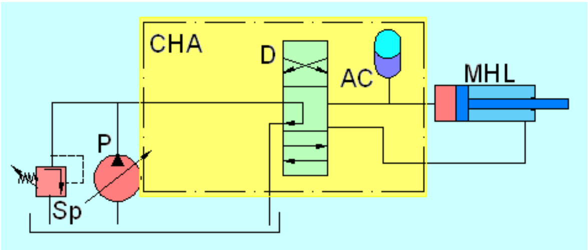


**Fig. 1.** Hydraulic drive circuit

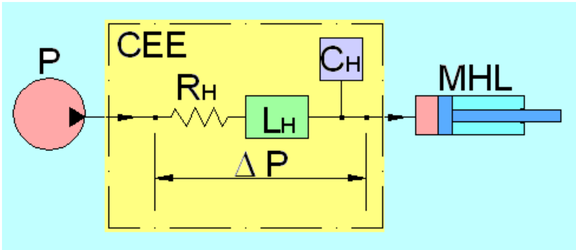


**Fig. 2.** Similar equivalent electric circuit

In this way, one can calculate a **generalized resistance equivalent to a hydraulic circuit [5].**
If fluid flow along the circuit in question takes place regularly, at a **frequency f,** the circuit impedes the flow by **total impedance** $Z_H\left[\frac{daN/cm^2}{cm^3/s}\right]$ which synthetically expresses flow-dependant pressure variation:

$$Z_H = \sqrt{R_H^2 + \left[2\pi f L_H - \frac{1}{2\pi f C_H}\right]^2} \tag{11}$$

Based on the above relations the characteristics of hydraulic circuit can be calculated using terms specific to electrical circuits, adjusted to the fluid power specific language.
**In conclusion,** assessment of load losses in a fluid drive system only requires **knowing the value of coefficients ξ or λ** for each system component. In a wider range, these coefficients are **characteristic constants** of the elements taken into account. In some cases, they depend on fluid flow, as a matter of convenience and in these cases one shall use the basic relations (6) and (7), but this time taking into account **the coefficients ξ and λ as variable**. Before giving numerical values to the coefficients ξ and λ, there will be defined two flow ranges corresponding to ξ and λ as constant and respectively as variable coefficients and the limits of these ranges shall be specified. [1].
There are known two types of flow in fluid dynamics: **laminar flow** and **turbulent flow**.
**In laminar flow**, adjacent fluid molecules advance with parallel speed vectors: one can imaginarily separate adjacent fluid layers between which there is no exchange of fluid. Due to the viscosity of the fluid, each layer tends to drag the adjacent one. The forces of friction being proportional to the viscosity coefficient and velocity gradient, **load losses**, in this case, will be proportional to the coefficient of friction and the fluid flow.

**In turbulent flow,** adjacent fluid molecules have no parallel trajectories. Speed vectors of molecules tend to depart from the direction of the average flow speed [1].
**Laminar flow** mode of the working agent in a finite circular pipe is shown in Figure 3, while **turbulent flow** is shown in Figure 4 [2].

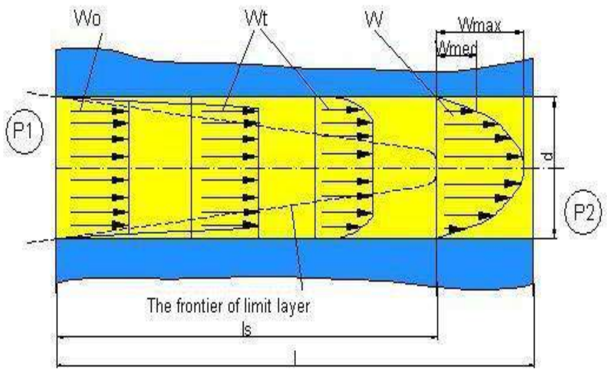


**Fig. 3.** Laminar flow in tubes

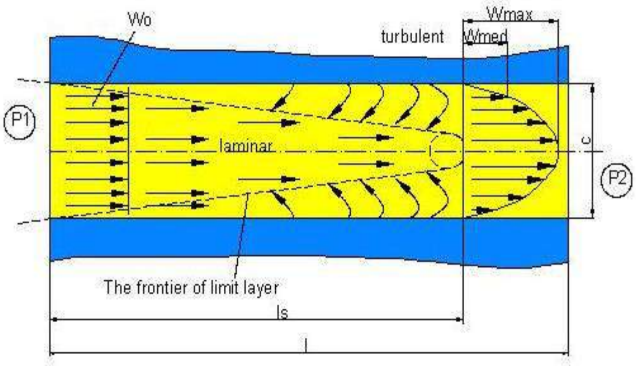


**Fig. 4.** Turbulent flow in tubes

**Fluid flow mode** depends on **the ratio** of **inertia forces** and **viscosity forces**, which is expressed by a dimensionless number, named **Reynolds number** [1]:

$$Re = \frac{\rho \cdot D_H \cdot w}{\eta} = \frac{D_{H} \cdot w}{\nu} \tag{12}$$

where: η – dynamic viscosity, ν – kinematic viscosity; $D_H$ – hydraulic diameter, defined in a previous paragraph; $w$ – fluid flow rate in a given point.
To calculate **Reynolds number**, suitable formulas are given in the literature for different situations. There are also charts and tables based on which there can be set the **coefficients:** λ, which characterizes linear load losses and ξ, which characterizes local load losses [1], [2], [5]. On these grounds, one can assess local and linear load losses in hydrostatic drive circuits.

### 3. Assessment of flow losses in hydrostatic systems

As regards **fluid losses in** complex **hydrostatic** drive **systems**, literature presents a series of specific elements through which there occurs leakage of working fluid and it also gives a series of specific mathematical relations and graphs for assessment of losses, [1], [2], [3], [4], [5]. From the literature investigated it can be noticed that **local flow loss,** through functional clearances of drive components, can be assessed by a relation of type [3]:

$$\Delta Q_i = K_{Qi} \cdot \Delta p; \ [m^3/sec]; \tag{13}$$

where: $K_{Qi}$ is **flow loss coefficient**, which is dependent on the structural sizes (linear and angular dimensions) of the element in which leakage occurs and also on the **dynamic viscosity** of hydraulic fluid used; $\Delta p$ – pressure difference between the high pressure enclosure and low pressure one, which causes the flow loss.
In a situation where in the construction of a component there are comprised several structural types of interstices, slots, clearances, subjected to the same differential pressure $\Delta p$, as in Figure 5, the relation for assessment of flow losses will be:

$$\Delta Q_i = \left(\sum_{i=1}^{s} K_{Qi}\right) \cdot \Delta p = K_Q \cdot \Delta p; \ [m^3/sec]; \tag{14}$$

If in the construction of a hydraulic component there are comprised several structural types of interstices, slots, clearances, which are connected in series, as in Figure 6, the global relation for assessment of flow losses will be:

$$\Delta Q_i = \sum_{i=1}^{s} K_{Qi} \cdot \Delta p_i;\ [m^3/sec]; \tag{15}$$

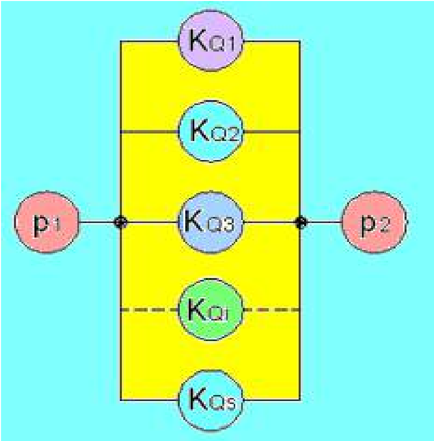


**Fig. 5.** Resistive elements in parallel

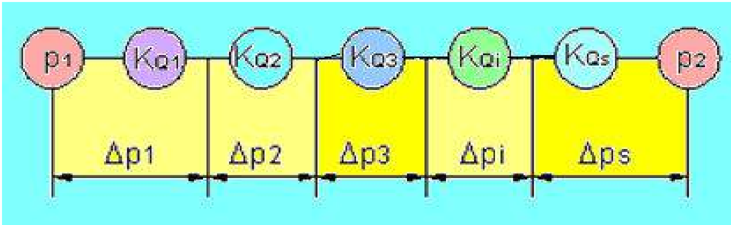


**Fig. 6.** Resistive elements in series

Both situations are commonly found in the structure of hydraulic equipment components [3].

## 4. Energy dissipation in hydrostatic drive systems

All the components of a hydrostatic drive system are impacted, more or less, by pressure and flow losses, resulting in significant energy dissipation in the system.
Energy dissipation in hydrostatic systems manifests itself by default in all drive components that, in order to convey energy in the system, use transporting fluids, in the form of liquids or gases [2].
Hydraulic power used in order to overcome load drop on a component of the system, at a circulated volumetric flow rate Q = constant, is given by the relation:

$$N_h = \Delta p \cdot Q = \xi\,(x)\,\frac{\rho}{2} \cdot \frac{Q^3}{A\,(x)};\ [Watt] \tag{16}$$

And for (i) components crossed by the flow Q, it results:

$$N_h = \left(\sum\nolimits_i \Delta p_i\right) \cdot Q;\ [Watt] \tag{17}$$

The above presented are phenomena that exist and always accompany the phenomena of working fluid flow through hydrostatic drive systems, but the amount of losses in the system should be as small as possible for the **system to be a high efficiency one** [2].

There are situations in which, **in order to adjust** some hydraulic parameters of the drive system, the phenomenon of **energy dissipation** described **is used** in spite of increased energy consumption in the system.
Such an example is a **hydrostatic drive system with resistive speed control.** This is done, for instance, in the case that the hydrostatic drive systems has **fixed displacement pumps,** by using pump flow control devices, which work by dissipating a part of the available energy. In other words, **part of the pump flow is discharged** to the system tank at a specified pressure drop. Such a flow control system is shown in Figure 7 and Figure 8. Since the pump P is a fixed displacement one, the only way to decrease the flow feeding the motor is discharging a part of it, at the pressure required by the motor, to the system tank (T). It results that the flow that must be discharged to the tank gets:

$$Q_T = Q_p - Q_p^{**} = \frac{\tau-1}{\tau}\,Q_p \tag{18}$$

To be possible to adjust the speed of the receiver it is considered the diagram in Figure 7, in which the flow limiter (fixed throttle valve) or the hydraulic resistance LD has been inserted, **on the active** circuit **line**, which serves as a agent **flow regulator**, while the valve S serves as a **pressure limiting device** [2].

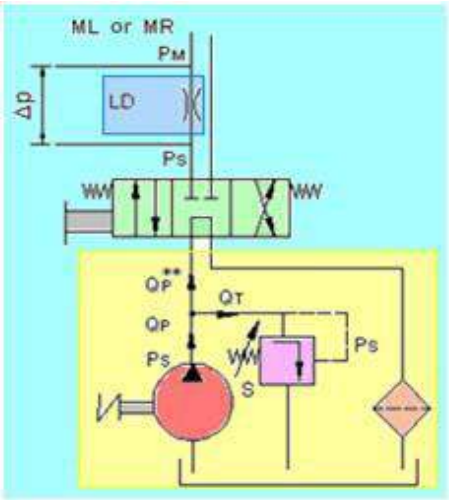


**Fig. 7.** Flow limiting device montage

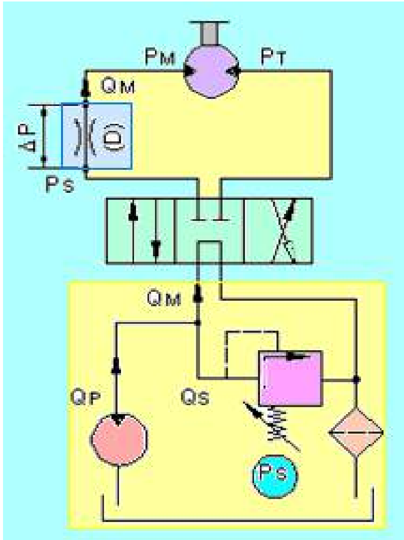


**Fig. 8.** Diagram of a hydrostatic system with fixed displacement pump and rotary hydraulic motor

If it is considered that the hydraulic circuit losses are negligible, then pressure drop on the throttle valve LD will be:

$$\Delta_p = p_s - p_M = \xi \cdot \frac{\rho}{2} \cdot \left(\frac{Q_p^{**}}{a^2}\right)^2; \tag{19}$$

where (a) is flow section of the throttle; $p_s$ is control pressure of the pressure limiting valve; $p_M$ is pressure required by the (linear or rotary) motor to develop the load at its shaft or rod. In these circumstances, if one assesses power consumption in the system, there results the following distribution of power on the system components:

- **Power supplied** by pump and taken over by the working agent:

$$N_p = Q_p \cdot p_s, [\mathrm{W}] \tag{20}$$

- **Power** effectively **taken over** by the hydraulic motor for overcoming load at the rod or shaft:

$$N_{MOT} = Q_p^{**} \cdot p_M;\ [\mathrm{W}] \tag{21}$$

where: $Q_p^{**} = Q_p/\tau$ and $p_M \cong \frac{p_s}{\gamma}$; τ and γ being supra-unitary numbers that define functions of pump flow, respectively, functions of adjusted pressure.

- **Power dissipated** by the pressure limiting device, namely the valve S, will be:

$$N_s = Q_T \cdot p_s = \frac{\tau-1}{\tau} \cdot Q_p \cdot p_s; \tag{22}$$

- **Power dissipated** by the flow limiting device LD will be:

$$N_{LD} = Q_p^{**} \cdot (p_s - p_M) = \frac{Q_p}{\tau} \cdot p_s \left(1 - \frac{1}{\gamma}\right) = Q_p \cdot p_s \cdot \frac{\gamma-1}{\gamma\tau}; \tag{23}$$

In the case shown in **Figure 8,** there is presented an application for a hydrostatic system with fixed displacement pump and rotary hydraulic motor, with throttle valve on the supply line [2].
In this case, we can calculate:
- Pressure drop on the hydraulic resistance (throttle valve):

$$\Delta p = p_s - p_M = \xi\ \frac{\rho}{2} \frac{Q_M^2}{Q^2} \tag{24}$$

- Flow circulated through the motor:

$$Q_M = Q_p - Q_s \tag{25}$$

- Theoretical angular speed at motor shaft:

$$\omega_M = \frac{Q_M}{V_{OM}} = \frac{Q_p - Q_s}{V_{OM}} \tag{26}$$

- Power taken over by motor, in (kW):

$$N_M = Q_M p_M \tag{27}$$

- (Theoretical) pump power:

$$N_P = Q_P p_s \tag{28}$$

- Power dissipated via throttle:

$$N_D = Q_M (p_s - p_M) \tag{29}$$

- Power dissipated via valve:

$$N_s = Q_s p_s = (Q_p - Q_M)\, p_s \tag{30}$$

- **Efficiency** of hydrostatic drive system with fixed displacement pump:

$$\eta = \frac{N_M}{N_p} = \frac{Q_M p_M}{Q_p p_s} = \frac{(Q_p - Q_s) p_M}{Q_p p_s} = \left(1 - \frac{Q_s}{Q_p}\right) \frac{p_M}{p_s} \tag{31}$$

From the above analysis it can be noticed that **efficiency of this hydrostatic system** is lower when the flow of hydraulic agent conveyed through the pressure limiting valve is greater and the ratio of motor pressure ($p_M$) and valve pressure ($p_s$) is lower, which is also shown in Figure 9.

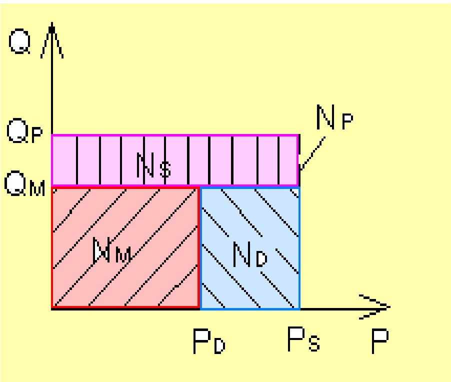

**Fig. 9.** The chart of power types in the system

Such a drive and control system is **totally ineffective** the more the power values required by the driven machine are higher, namely over 5 – 8 kW. Such system for drive and control of parameters **is used in control** or auxiliary **circuits** or as the main drive system in small power equipment or machines [2]. It should be mentioned that the flow limiter can be also installed on a bypass of the active circuit, as well as on the motor return line. All these variants are not efficient in terms of energy for high power hydrostatic systems.

In this chapter we presented **fixed** (constant) **displacement pump applications**, used in **open circuit,** in which **control** of energy and kinetic parameters of the circuit is based **on resistive principles,** with **large energy dissipation**, these systems being **ineffective in terms of energy**. These control principles are used in small power drives, fixed laboratory actuators, actuators for control circuits, and less in medium and high power drive equipment.
We will present below a series of modern methods for increasing energy efficiency of hydrostatic drive systems.

## 5. Methods for increasing energy efficiency of hydrostatic systems

Considering the above, to reduce load and flow losses, both local and linear/distributed, there have been proposed a range of techniques and methods to reduce these losses, which have been taken into account both during the design of components and when designing the functional diagrams of

hydrostatic drive systems, in order to **improve energy efficiency** of these systems, as well as to **increase** their **functional performance**.
In the following we will present some of these techniques / methods and **modern concepts** for **increasing energy efficiency of hydrostatic systems**, by developing functional diagrams based on:
- **primary adjustment**, where variation of kinematic parameters is achieved by variation of the pump displacement / geometric volume, respectively by variation of flow;
- **secondary adjustment**, where variation of kinematic parameters is achieved by variation of the rotary hydraulic motor displacement / geometric volume;
- **mixed adjustment**, both pump displacement adjustment and rotary hydraulic motor geometric volume adjustment;
- **energy recovery**, where some of the energy is stored and reused in the next phase;
- **recirculation of** available **energy** between component equipment.

### 5.1. Hydraulic system with primary adjustment of displacement

Figure 10 presents a **primary adjustment diagram** of pump displacement, working in **open circuit,** for a system with open-circuit rotary motor, system that can be identical also for linear motor drives. The diagram presents a drive system for a pulling winch in which the pump *P* is set to zero displacement, on normal position (no control), displacement being adjusted by control of the operator between zero and maximum displacement [4]. Adjustment of winch speed is done from pump displacement value, and direction of motion - from directional valve *DP* control. The directional valve *DP* is an electrohydraulic pilot valve with the possibility of braking in the neutral position.
Figure 11 presents a diagram of a hydrostatic drive system with **primary adjustment**, but simple **closed circuit** and fixed displacement rotary motor, **with no motor brake.**

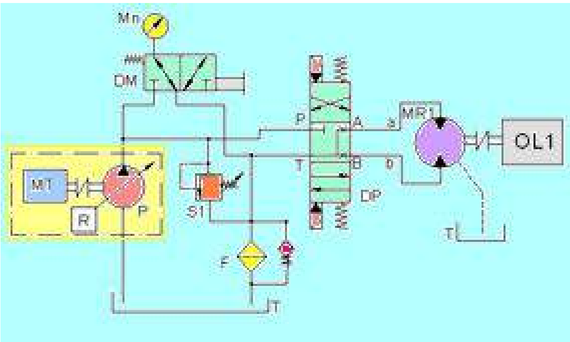


**Fig. 10.** Diagram of a hydrostatic system with primary adjustment and open circuit

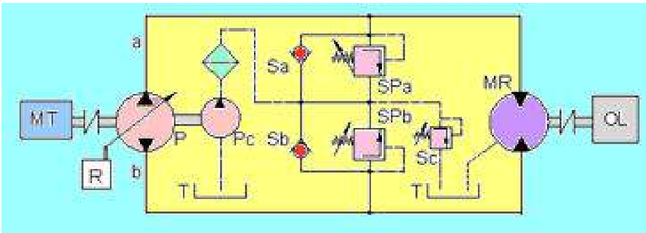


**Fig. 11.** Diagram of a hydrostatic system with primary adjustment and closed circuit

The diagram includes the **positive displacement pump** *P*, which is main pump, **with displacement primary adjustment by directional control valve** ***R***, which also performs the function of changing the distribution direction in the system. The pump is driven by the heat motor *MT*. The diagram includes a pump *Pc*, positive displacement, installed in tandem with the main pump, serving the system control and compensation circuit, which comprises the filter *F* and the drive pressure control valve *Sc*. To protect the power circuits *a* and *b* there is used the group of valves *SP* and *S*, *a* and *b*, which performs secondary and cavitation protection of system circuits. The final component of the system is the motor *MR* which does the actuation of the working body *OL,* or the equipment of the machine. **The directional control valve R** of the pump performs adjustment of pump *P* displacement, both according to the machine operator's order, and according to the adjustment requirements imposed by the technological process which the working body *OL* is involved in.
If the hydraulic units used in the diagram are **reversible units,** which can operate both as pumps and as hydraulic motors, the hydrostatic system **enables the use of braking energy** of a vehicle

as motor brake, which it can capture and reuse at starting up. These energy recovery systems fall into the category of **hydrostatic regenerative systems**.

### 5.2. Hydraulic system with secondary adjustment of displacement

**Secondary adjustment** is an adjustment version in hydrostatic systems in which, for instance, an adjustable axial piston unit, with a predetermined operating pressure, seeks its adequate capacity to maintain a given rotational speed. In order words, there takes place capacity adjustment at constant pressure, so as to result in constant (or almost constant) motor speed [6], [7].
Figure 12 presents a hydrostatic drive with secondary adjustment at constant pressure. The capacity of the volumetric machine in the secondary unit of the winch drive system can be changed, in size and direction, by means of an adjustment device, type positioning cylinder, dependant on flow, controlled by a proportional solenoid valve, ensuring proportionality between geometric volume variation, consequently torque variation, and control signal [6].
In Figure 13 one can see the structure of a **hydrostatic servo system with secondary adjustment** under a **quasi-constant pressure** regime, maintained by the hydro-pneumatic accumulators (see Figure 12).
The structure is as follows: *1*-solenoid valve; *2*-positioning cylinder; *3*-secondary unit; *4*-electric speed voltage generator; *5*-speed regulator; *6*-amplifier; *7*-load; *8*-regulator of tilt angle; *9*-transducer of tilt angle; *10*-hydraulic coupling; *11*-pressure filter.

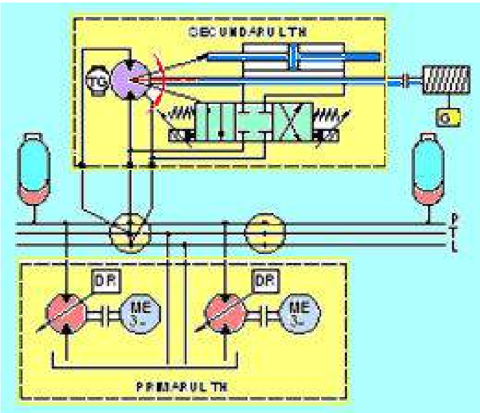


**Fig. 12.** Diagram of secondary adjustment

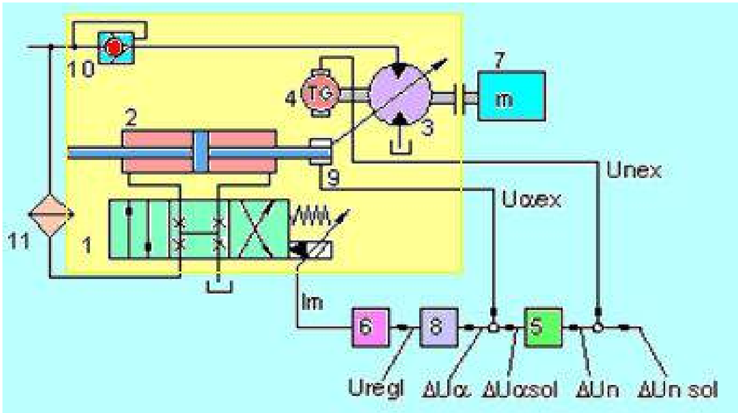


**Fig. 13.** Electrohydraulic control at secondary adjustment

**Secondary adjustment** is able to convert hydraulic energy into mechanical energy (operation as a motor) and mechanical energy into hydraulic energy (operation as a pump), **with almost zero losses**.
Secondary adjustment is possible both in **open circuit** (open-loop) and **closed** drive **circuit** (closed-loop).

### 5.3. Hydraulic system with mixed adjustment of displacement

For hydrostatic systems with special requirements concerning functional performance there is applied **the concept of mixed adjustment,** where one can find both **primary adjustment in pumps** and **secondary adjustment in hydraulic** drive **motors**. Figure 14 presents a hydraulic diagram with mixed adjustment, used in hydrostatic drive of a forging manipulator [4].
The forging manipulators require **precise movements** of the incandescent steel block, during processing by plastic deformation, on hydraulic forging presses. In addition to the **rotational movement** of the piece during forging, which is performed by the hydraulic motors M1 and M2, there must also be provided the manipulator **travel movement**, necessary to process and transport the ingot / piece between the oven and press, performed by the hydraulic motors M3 and M4. Figure 14, does not present the performing of translational movements that use hydraulic cylinders hydrostatically driven as well.

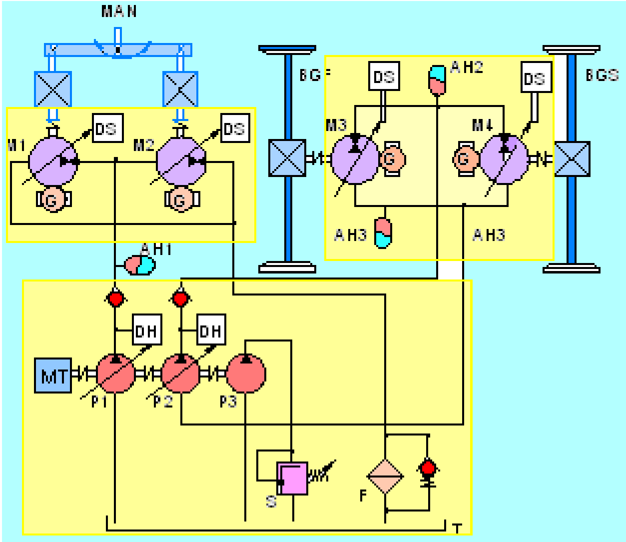


**Fig. 14.** Diagram of mixed adjustment used in forging manipulators

Diagram structure is as follows: *MT* – heat motor (or electric motor in some applications) *P1*, *P2*, *P3* – machine pumps; *P1* – pump serving the ingot rotation system, **in open circuit**, *P2* – pump serving the machine moving system, **in closed circuit**; both are **variable displacement pumps** with **primary adjustment** (*DH*), and *P3* is a **fixed displacement pump,** used for compensation of losses in the closed circuit of movement and for driving auxiliary equipment. *M1* and *M2* are **secondary adjustment** (*DS*) hydraulic motors, used for driving the ingot rotation system, while *M3* and *M4* are also variable displacement hydraulic motors with **secondary adjustment** (*DS*), used in the manipulator moving system.
The hydraulic motors, both for rotation (*M1* and *M2*) and for moving (*M3* and *M4*), work in **closed loops for tracking** the rotation angle, based on electrical signals given by speed voltage generators mounted on the rotation axes, processed by electronic control and adjustment systems, by which there is established correlation between the desired rotation angle and actual achieved angle.

### 5.4. Energy recirculation hydraulic system

Figure 15 presents a hydraulic diagram designed on the principle of energy recirculation. This refers both to **storing the energy** from pieces of equipment in an energy overplus (braking phases, load lowering, idle running etc.) and to **giving the stored energy** to other pieces of equipment that are in an energy shortage (acceleration phases, starting-up from rest, lifting etc.) [4].

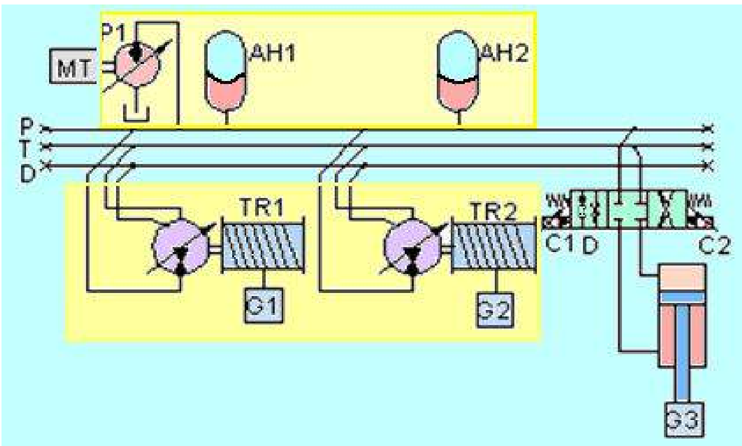


**Fig. 15.** Drive diagram with energy recirculation

The diagram consists of two winches *TR1* and *TR2*, which lift the loads *G1* and *G2*; these winches are driven by the hydraulic units *MR1* and *MR2*. The diagram also comprises the lever *G3*, driven by a linear motor *ML*, which is switched on or off the hydraulic buses *P* and *T* by use of directional control valve *D*. The machine is equipped with a heat motor *MT* and pump *P1*, which can work both as pump and motor. Both the pump *P1* driven by the heat motor *MT* and the rotary motors are **variable displacement,** and displacement adjustments are selected in such way that they facilitate energy delivery both from load to accumulators and from accumulator to load, as close as possible to technical and economical optimum, with minimum energy loss, that is **with higher** drive **efficiency**.

## 6. CONCLUSIONS

Based on the specialized literature, the paper presents some general considerations on pressure and flow losses in hydrostatic drive systems, relations for calculating them, as well as relations for assessing dissipated power and total efficiency of hydrostatic drive systems [8].
Following the presentation of energy losses and places where they occur, there is presented a set of **principles and modern concepts** for the design of hydrostatic drive diagrams, in order to minimize losses and **improve energy efficiency**.
By the above mentioned, the paper proves useful, especially to young professionals, in choosing how to design hydrostatic drive systems depending on actual requirements, with the aim of **developing high energy efficiency hydrostatic systems.**

**Acknowledgement**

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**References**

[1] M. Avram, “Acţionări hidraulice şi pneumatice” (Fluid Power), University Publishing House, Bucharest, 2005;
[2] G. Axinti, “Acţionări hidraulice şi pneumatice. Vol. I. Componente şi sisteme, funcţii şi caracteristici” (Fluid Power. Vol. I. Components and systems, functions and features), Tehnica-Info Publishing House, Chişinău, 2008;
[3] G. Axinti, “Acţionări hidraulice şi pneumatice. Vol. II. Dinamica echipamentelor şi sistemelor” (Fluid Power. Vol. II. Dynamics of equipment and systems), Tehnica-Info Publishing House, Chişinău, 2008;
[4] G. Axinti, “Acţionări hidraulice şi pneumatice. Vol. III. Baze de calcul, proiectare, exploatare, fiabilitate şi scheme de acţionare” (Fluid Power. Vol. III. Calculation bases, design, operation, reliability and drive diagrams), Tehnica-Info Publishing House, Chişinău, 2008;
[5] V. Marin et al., “Sisteme hidraulice de acţionare şi reglare automată” (Automatic drive and control hydraulic systems), Technical Publishing House, Bucharest, 1981;
[6] T.C. Popescu , D.D. Ion Guta, Al. Marin, “Adjustment of Hydrostatic Transmissions through Virtual Instrumentation Technique”, Proceedings of The 8-th International Industrial Simulation Conference (ISC 2010), 7-9 June 2010, Budapest, Hungary, ISBN: 978-90-77381-5-57, pp. 248...253;
[7] D. Călăraşu, “Reglarea secundară a sistemelor de acţionare hidrostatică în regim de presiune cvasiconstantă*”* (Secondary adjustment of hydrostatic drive systems under a quasi-constant pressure regime), Media-Tech Publishing House*,* Iaşi, 1999;
[8] *** INOE 2000-IHP. “Studiu teoretic si experimental de analiza a consumurilor energetice in sistemele hidraulice de actionare , comanda si reglare” (Theoretical and experimental study for the analysis of energy consumption in hydraulic drive, control and adjustment systems), Core Programme, April 15, 2016 (*internal document*).

# Functional Description of a Hydraulic Throttle Valve Operating inside a Hydraulic Circuit

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***Abstract:*** *Nowadays the global manufacturers for industrial and mobile equipments offer products containing advanced hydraulic systems by means of which are accomplished the multiple tasks imposed by the designers in a convenient and easy manner. The hydraulic systems are comprised of components of last generation that are able to circulate the working fluid (mineral oil) at high values of pressure and considerable speed through their proper circuit. Besides the hydraulic pump and engine as the primary components of the hydraulic circuit there are also the pressure and flow control components necessary to change the pressure and flow rate parameters of circulated fluid at a specific moment of time. A variable throttle valve operation that can modify the fluid flow rate within the circuit is analyzed in this paper. A 3D model for a variable throttle valve unit was built and introduced into numerical analysis using ANSYS CFX. Based on the defined incoming data the model operation was analyzed and the results are presented in order to highlight how the parameters changes when using the throttle valve within the hydraulic circuit.*

***Keywords****: fluid, hydraulic unit, fluidic actuation, hydraulic throttle valve*

## 1. Introduction

We are witnessing today an unprecedented development of multifunctional machines designed to achieve different heavy works in the fields of construction or agriculture. These works may consist of land digging using the provided bucket, ground leveling using a proper blade, loading various construction materials using the wheel loader with bucket, or different work tasks for agricultural land preparation. The simultaneous fulfilling of these tasks using a single machine is possible by means of multiple working devices that may be coupled to the basic machine. All of these special auxiliary equipments have mounted advanced hydraulic systems which enable them to carry out the tasks for which they are designed. A hydraulic system that works within a construction or agricultural machinery is a complex combination consisting of multiple components fitted together in a working circuit having a fluid as working agent. The primary components of the hydraulic circuit are represented by the pump and motor, but also components for limiting the pressure or flow control, filtering elements for hydraulic agent filtration, coupled by means of flexible or fixed ducts to allow a continuous circulation of the working fluid to accomplish the working process of respective equipments. The working fluid, represented by the mineral oil is the element by means of which the hydrostatic energy is transmitted from the hydraulic pump to the hydraulic motor which finally provides rotational or translational motion to the machine working body. The operation of a hydraulic circuit involves the use of hydraulic throttle valves that are capable of modifying the pressure or flow rate values for the circulated fluid at a specific moment of time, depending on momentary needs at the machine working body on which the hydraulic system is mounted.

## 2. Hydraulic valve models commonly used within hydraulic circuits

There are many models of pressure or fluid flow rate control devices used in hydraulic circuits. For each model are available a range of nominal sizes function of mounting parameters. The nominal size determines the maximum value of fluid flow rate circulated through the valve and the maximum pressure values inside the hydraulic circuit. The valves can be attached to the circuit using various mounting solutions. The most frequently encountered mounting solutions are the direct installation on pipeline, using the threaded connection, as cartridge, or in-plate mounting.
As hydraulic components the devices used for pressure and flow rate control inside a hydraulic circuit can be classified as in table 1. [5]

**TABLE 1**

| Directional Control | Pressure Control | Flow Rate Control | Electro Hydraulic |
|---|---|---|---|
| Check valves<br>Directional valves | Fixed throttle valve<br>Variable throttle valve | Throttle valve<br>(flow dependent)<br>Control valve<br>(flow independent) | Servo hydraulic valve<br>Proportional hydraulic valve |

A check valve mounted inside a hydraulic circuit is making possible fluid circulation in a certain direction and fluid flow blocking in the other direction.
The directional control valves are used as switching valves which can control the start, stop and change in direction of fluid flow.
The pressure control valves are generally used to realize the adjustment or pressure control at a certain value.
The flow rate control of the hydraulic agent (working fluid) during system operation can be achieved when using special valves capable to modify the momentary fluid flow by adjusting the orifices area.
Modern solutions in the field of pressure and flow rate control are represented by electro-hydraulic valves also known as proportional hydraulic valves and servo control valves, which may be used to perform the directional control, pressure control or flow control function.
Directional valves may be direct operated or pilot operated and depends on the actuating force needed to move the control element. It can be said that a directional valve is flow dependent. A direct-operated directional valve can be mechanically, electrically, hydraulically or pneumatically controlled.
When the pressure values of the hydraulic system is high, should be taken into account some losses at the valve level and this is happening when the pressure is over 250 bars. Besides the high values of pressure within the hydraulic system, the losses are determined by the gap size between the control element and the valve body, but also by the viscosity of the working fluid. [5]

### 3. Modeling aspects for hydraulic throttle valve

A three-dimensional model for hydraulic throttle valve was built using Solid Edge software as presented in Fig. 1. On the model will be declared fluid inlet and outlet and the fluid volume placed inside as fluid region. Inside it is placed the control element having the possibility of translational motion inside on the vertical direction, providing in this manner a variation of fluid flow area at a given time.

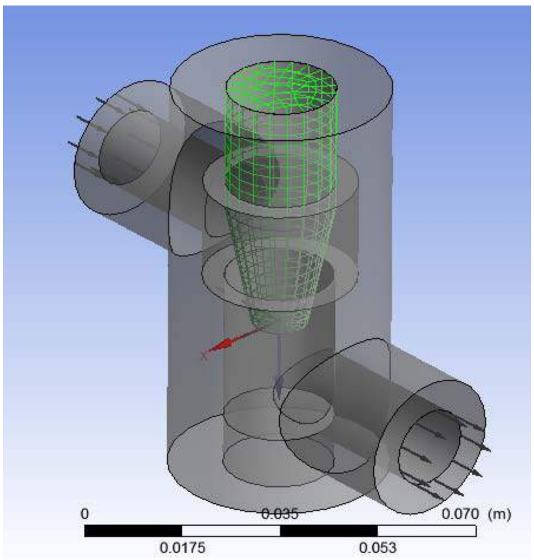



**Fig. 1.** Hydraulic throttle valve assembly model

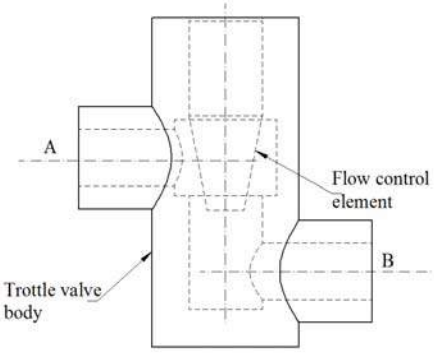
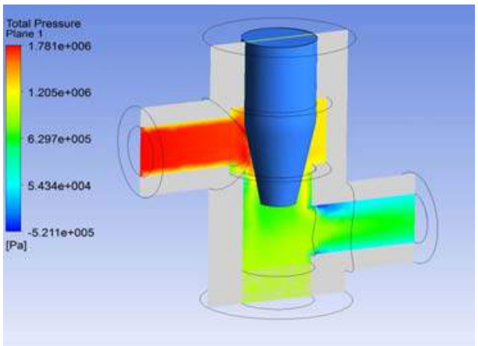


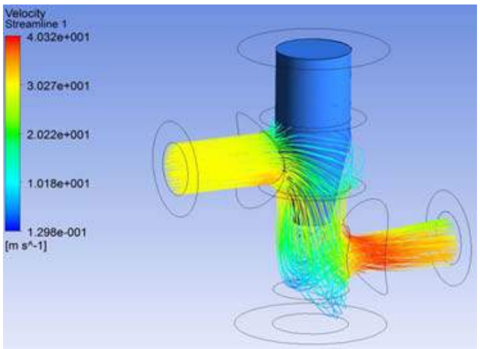

**Fig. 2.** Schematic representation for hydraulic throttle valve (A – fluid inlet; B – fluid outlet)

## 4. Computation fluid dynamics analysis for anchoring model

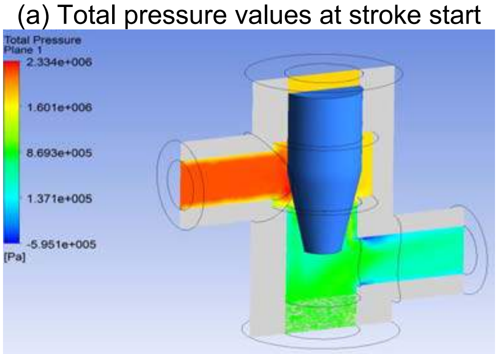
The 3D model was analyzed using ANSYS CFX software to highlight the hydraulic throttle valve operation while the control element is performing a translational motion on the vertical direction. It was declared a fluid velocity at the inlet of 30 m/s, the reference pressure value inside the fluid region was set at 100 bars, valve material body is represented by steel, the flow actuator is stated as immersed solid within the fluid region and it has a speed of 0.01 m/s on vertical (z) direction.



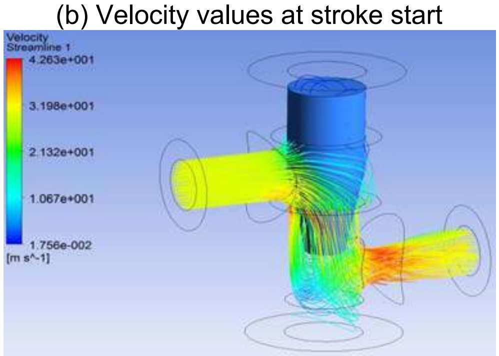

(a) Total pressure values at stroke start



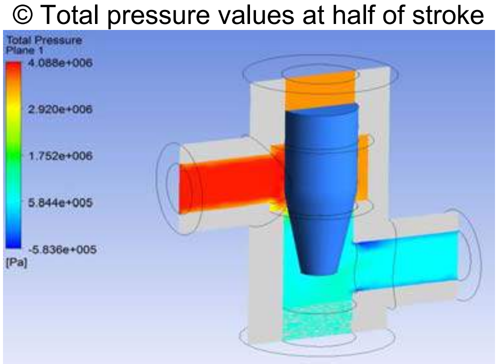

(b) Velocity values at stroke start



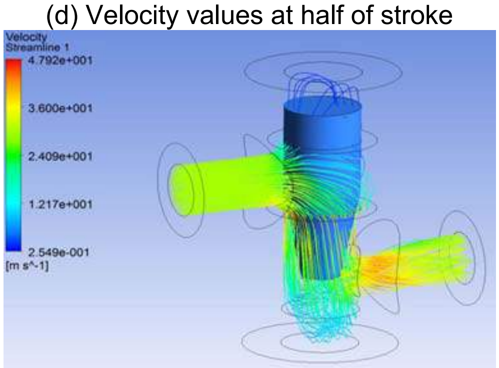

© Total pressure values at half of stroke




(d) Velocity values at half of stroke




(e) Total pressure values at stroke end




(f) Fluid velocity values at stroke end

**Fig. 3.** The obtained results for hydraulic throttle valve operation

A final report was generated by the program where all information is available regarding the numerical analysis performed on the hydraulic throttle valve model. They were defined three domains (fluid domain, throttle valve body and the flow control element). For each of the domains a mesh was made having the number of nodes and elements as shown in table 2.

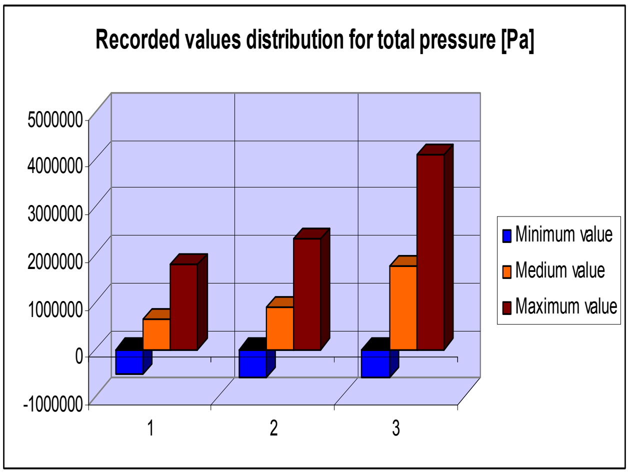
**TABLE 2**: Mesh information

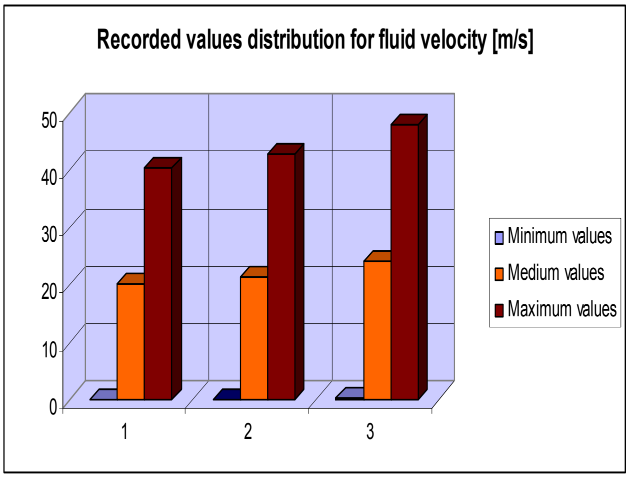
| CFX Mesh | Fluid domain | Throttle valve body | Control element |
|---|---|---|---|
| Nodes | 3238 | 4884 | 1265 |
| Elements | 14719 | 21879 | 968 |

The movement of the valve control element is achieved on the vertical direction and the results presents the values obtained for total pressure and velocity within working fluid region. The results are presented with recording values from the stroke start, middle stroke and at stroke end as shown in table 3.

**TABLE 3**: Values recorded for total pressure and velocity

| Position number | Control element position | Total pressure Values [Pa] | | | Velocity values [m/s] | | |
|---|---|---|---|---|---|---|---|
| | | Minimum | Medium | Maximum | Minimum | Medium | Maximum |
| 1 | Stroke start | -521100 | 629700 | 1781000 | 0.1298 | 20.2200 | 40.3200 |
| 2 | Half of stroke | -595100 | 869300 | 2334000 | 0.0175 | 21.3200 | 42.6300 |
| 3 | Stroke end | -583600 | 1752000 | 4088000 | 0.2549 | 24.0900 | 47.9200 |




## 5. Conclusions

A hydraulic throttle valve model was realized and analysed using ANSYS CFX software in order to highlight the fluid flow inside the device body when the valve control element performs a translational motion on the vertical direction. They were presented the values for total pressure and fluid velocity registered inside the fluid region. The role of the control element is highlighted inside the hydraulic throttle valve as the component able to modify the working fluid flow rate inside and downstream from the hydraulic device.

### References

[1] G. Axinti, A. S. Axinti, "Acționări hidraulice și pneumatice", Vol III, Editura Tehnica-Info, Chișinău, 2009;
[2] F. D. Scheaua, A. S. Axinti, G. Axinti, "Mathematical model analysis on hydraulic energy dissipation devices", Hidraulica Magazine, no. 1-2 / 2012, ISSN 1453 - 7303, pp. 38-42;
[3] F. D. Scheaua, "Seismic protection of structures using hydraulic damper devices", Annals of the University Dunarea de Jos of Galati, Fascicle XIV, Mechanical Engineering, 14, 2010;
[4] F. D. Scheaua, F. Nedelcut., "Energy dissipation device using fluid dampers", The Annals of Dunarea de Jos, University of Galati Fascicle XIV Mechanical Engineering, Galaţi, Romania, 2012;
[5] http://dta.eu/hydraulics/hydraulic-valves/

# Fertilizer Injection Device

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***Abstract:*** *The fertilizer injection device is a component of the equipment for fertigation of horticultural crops (the main vegetable species cultivated in protected areas, and respectively tree species and shrubs) used at works in aggregate with drip irrigation and micro sprinkling systems.*
*The injection device (type double diaphragm pump with hydraulic control) developed under the PN-II-PT-PCCA-2013-4-0114 –Contract no. 158/2014 in Experimental Model phase, offers a number of advantages compared to products of prestigious companies in the field such as VERDER AIR, DEBEM, TUV, TAPFLO, namely [4, 5]:*
*- It uses as working (driving) fluid the irrigation water, taken from the same pipe in which the primary solution is injected, which combined with the irrigation water forms the fertilizing solution;*
*- It does not require electricity or compressed air, which ensures operating autonomy in any spot of the arrangement for irrigation;*
*- The injection pressure is achieved on the principle of difference between the active surfaces of drive chambers and injection chambers, and it can be determined very precisely according to the hydraulic parameters of the irrigation facility which it works in an aggregate with, early since the product design stage ; the primary substance flow can be adjusted within wide limits, by changing the flow supplying the drive chambers, thus modifying the frequency of the pump central shaft (joint with the membranes that delimit the drive from injection chambers);*
*- It shall be installed in parallel with the main circuit of the irrigation facility (bypass system) through two quick couplings for taking over the water used as the driving fluid, and respectively for injecting the primary solution; this mounting system does not introduce pressure losses in the irrigation facility pipe.*

***Keywords:*** *Injection device, fertigation, primary solution*

## 1. Introduction

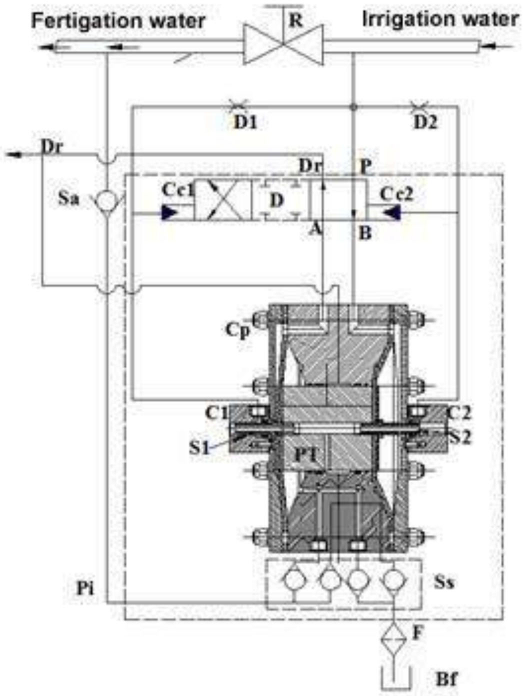


**Fig. 1.** Schematic diagram of the fertigation equipment

The fertigation equipment is structured as shown in Fig.1 above.
It includes the device for injecting the fertilizing substance into the irrigation water, consisting of the dosing pump *Pd*, the hydraulically actuated directional control valve *D* and the block of primary solution inlet/discharge valves *Ss*, the container for preparing the fertilizing solution *Bf*, the equipment for measurement and adjustment of working parameters (*D1* and *D2*- throttles, *Sa*-return valve, *R*- tap valve), the hydraulic connection elements between pieces of equipment [2].

## 2. Description of the injection device

**The injection device** is of type double positive displacement pump with diaphragms [1], [4], [5], with hydraulic control (commutation of the directional control valve is hydraulically performed).
It is mounted in a hydraulic circuit which is parallel with the supply (main) circuit of the irrigation facility (by-pass). Between the points of connecting the injection device to the irrigation facility there is mounted the tap valve *R*, acting as an adjustable diaphragm, with which there is created pressure drop downstream of the location point, thus facilitating the injection process (this increases, in a wide adjustment range, the pressure difference Δp between connection points).
The device uses water taken from the supply pipe of irrigation facility as driving fluid, the overpressure necessary to inject the primary solution in the same pipe being created on the principle of surface difference between the drive chambers and the injection chambers, respectively through tap valve *R*.
The functioning of this injection device does not imply the existence of electricity or compressed air, which ensures operating autonomy in any spot of the arrangement for irrigation. The design, technical and functional parameters can be set very accurately, from the product design phase, depending on the hydraulic parameters (pressure, flow rate, configuration and sizes of the supply and distribution network) of the irrigation facility which it works in aggregate with and the technical elements of fertigation (the watering quota administered by use of the irrigation facility, the features of injected primary solution: chemical composition, content in chemical elements, solubility and compatibility of chemical fertilizers used in the preparation of primary solution, density, concentration, electrical conductivity, acidity, water quality, injection dosage and dilution dosage).
Primary substance flow can be adjusted within very wide limits, by changing the feed rate of the drive chambers, thus modifying the frequency of the pump central shaft (joint with the membranes that delimit drive chambers from injection chambers).
The double positive displacement pump with diaphragms [6], Figure 2, can be assimilated to a hydraulic amplifier with two identical sections, separated by a central disc (pump body) *1*. The rubber membranes with cloth insertion (180) x20 -3x2.5 are in shape of discs with center hole. They are fixed between external covers of pump *2* and front sides of the central disc on outer contour, respectively front sides of the piston *4* and flanges *8* on the inner contour. The membranes separate drive chambers of hydraulic amplifier (located on the outside) from injection chambers (located on the inside).

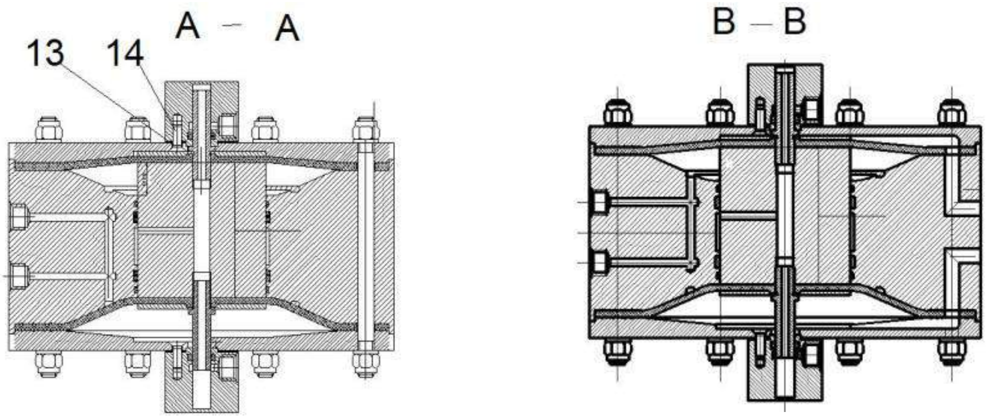


**Fig. 2**. Double positive displacement pump with diaphragm

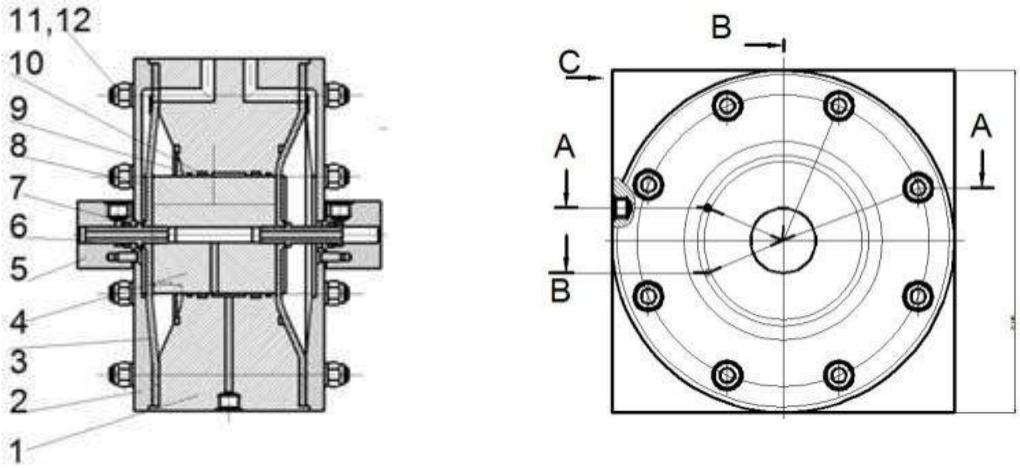


**Fig. 2**. Double positive displacement pump with diaphragm *(continued)*

The connection between drive chambers and exterior hydraulic circuit of the working fluid is accomplished through the passage holes in the covers and central disc, and the connection between injection chambers and exterior circuit of primary solution inlet-discharge is accomplished through the holes in the central disc.
The novelty of the technical solution taken up when developing the injection device consists in the directional control valve actuation, which is done by taking over hydraulic signals at the stroke ends of the pump mobile assembly, and also in the constructive manner adopted for the slide valve of the directional control valve, so that to meet the fertigation process requirements: quick switching; in the switching phase it should not generate pressure drop in the system, leading to blockage of the pump mobile assembly; once switched, there should exist forces to maintain it in position.

**The hydraulic directional control valve**
To control the travel direction of the mobile assembly of the injection device there is used a directional control valve supplied with water under pressure from the irrigation pipe upstream of the tap valve *R*, which it distributes alternately in the two drive chamber. Thus injection chambers increase or decrease in volume, sucking the primary solution from container *Bf* and discharging it through the branch with return valve *Sa* in the same irrigation pipe, downstream of the tap valve *R*, at a higher pressure than the one inside the pipe [6].
Actuation of the directional control valve occurs by sensing the stroke ends of the mobile assembly with membranes, by means of some slide valves, which open the way of directional control valve commands to an external drain *Dr*. Thus there is ensured the pressure drop on commands, so safe operation. Contact with the outside is done through the hole in the slide valve and the holes in the central piston, which regardless of the position it occupies, connects with the outside.
Control pressure is brought on the pistons ends of the distributor slide valve through two nozzles (throttles) *D1* and *D2* from pressure circuit *P*. These nozzles allow separation of the two control circuits, balancing of the distributor slide valve after its actuation and maintaining of the slide valve in position until receiving a new command.
The directional control valve, fig. 3, is a structure consisting of a body *1*, in which there is mounted an assembly comprising the slide valve *5* and two threaded bushings *2*, in which the slide valve pistons *8* delimit the two drive chambers. In the central area of the slide valve there are mounted symmetrically two valves, which by their displacement close or open paths from pressure port P to the consumer ports A or B, respectively to the drive chambers of the slide valve. At the same time connections are created between ports A or B and the output port T (*Dr*).
The valves are mounted on the slide valve so that in the central position they completely close the port P (positive coverage).

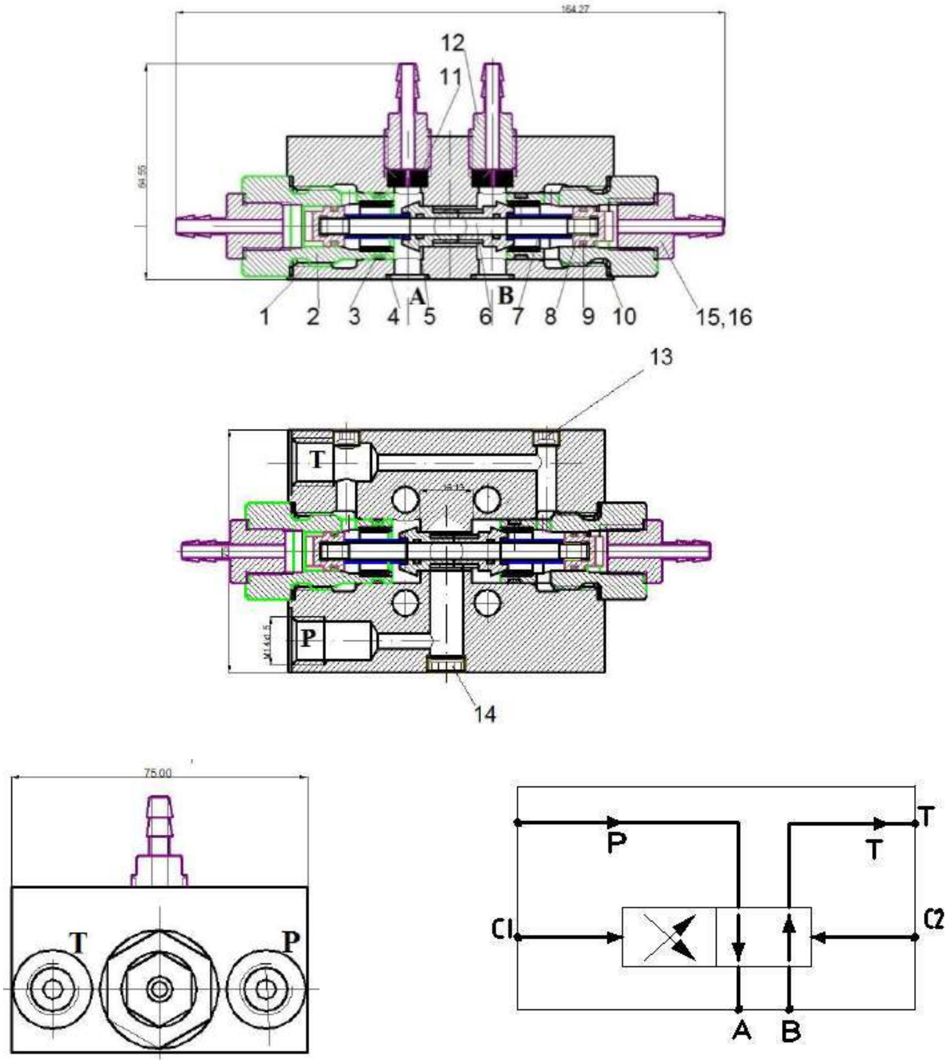


**Fig. 3.** Directional control valve

Under the action of pressure, the mobile assembly of the directional control valve is unbalanced, because the closing surface on the seat *ϕD* is larger than the closing surface type slide valve *ϕd* – see schematic diagram of the directional control valve, Figure 4. This creates a force that maintains in position the slide valve until it gets unbalanced as a result of pressure drop in the control chamber, by opening the paths at the stroke ends of the pump mobile assembly.

By moving to the stroke end, the slide valve of the pump sets the connection between the drive chambers of the directional control valve and the outside, through the port in the piston and pump body to *Dr*.

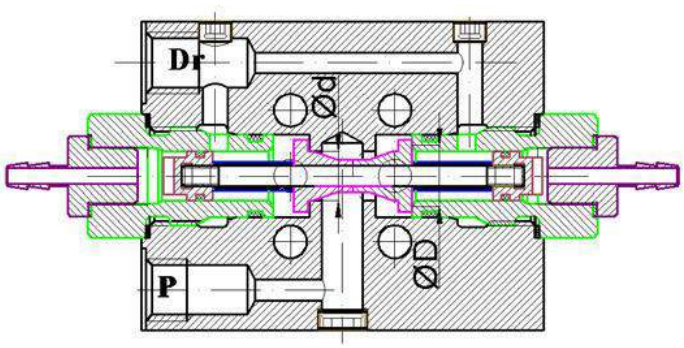


**Fig. 4.** Schematic diagram of the directional control valve

At the end of the slide valve stroke (e.g. the extreme right position) the right valve seals frontally on a seat $\phi D$, and the left valve on the cylindrical surface, at contact with the directional control valve body $\phi d$. Due to the difference between the surfaces $\phi d$ and $\phi D$, pressurized water in the pressure circuit *P* generates a force that opposes switching of directional control valve:

$$\phi D > \phi d;\ AD > Ad;\ \text{so } FD > FD$$

where: D, d are diameters of the sections which water pressure is acting on; $A_D$, $A_d$ - areas of the two sections; $F_D$ and $F_d$ - forces acting on the valves' assembly; $F = A \cdot p$, p- water pressure, A- area.
In this position of the slide valve, there are established the next paths: from *P* to the pump drive chamber *B*, respectively from the drive chamber *A* to the drainage *Dr*.
At switching, by pressure drop, there occurs unbalancing of forces:

$$F_D < F_d + F_{ccd},$$

where $F_{ccd}$ is the force produced by pressure on the extremity of the control piston which is no longer balanced.
The control chambers of hydraulic directional control valve are connected by nozzles (throttles) *D1* and *D2* to the pressure circuit *P* and the control slide valves of the pump, Figure 5. When the pump mobile assembly reaches the stroke end, the slide valve in question opens the path to the outside, causing pressure drop in the control circuit, which unbalances pressure forces exerted on the distributor slide valve, and as a result causes switching of the later.
In the switched position, through nozzles *D1* and *D2*, pressure in the control chambers is balanced, and due to the differences in surface between $\phi D$ and $\phi d$, there occurs a force that maintains the slide valve assembly on actuated position.

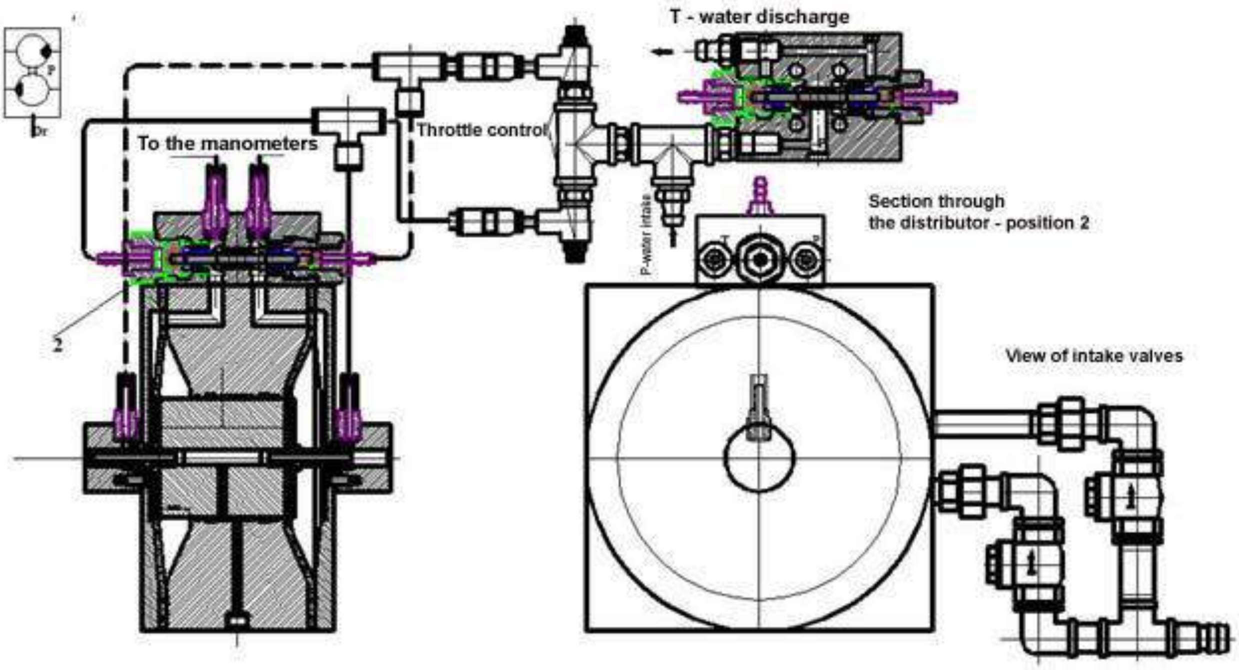


**Fig. 5.** Schematic diagram - Injection Device

**The block of primary solution inlet /discharge valves**
The block of primary solution inlet /discharge valves, Figure 6, consists of four needle type check valves and the related connection items.

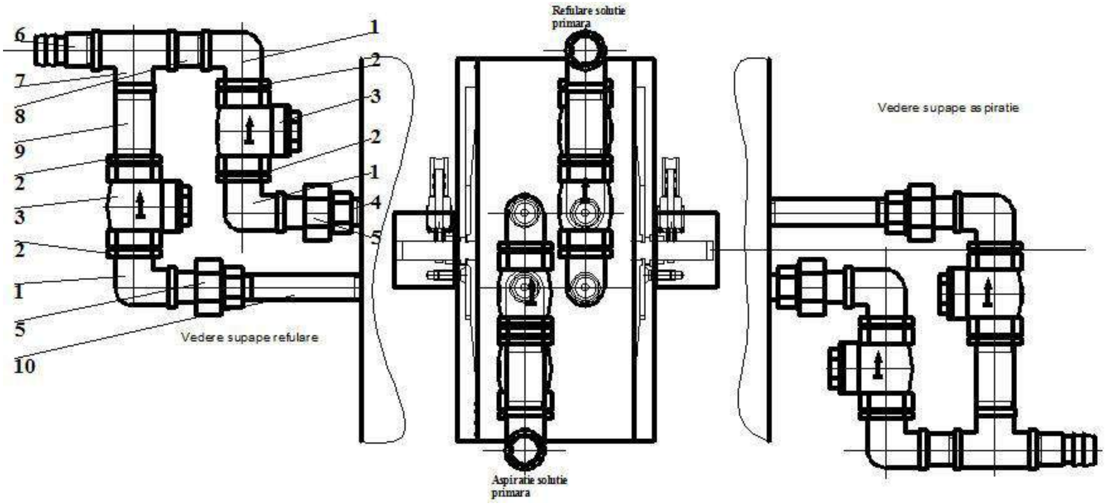


**Fig. 6.** The block of the primary solution inlet /discharge valves

The valves are arranged in two parallel planes, to each chamber being attached an inlet valve and a discharge valve, Figure 7. The inlet valves are connected to the branch *Ti*, by which the primary solution sucked from container *Bf* (Fig. 1) reaches alternately the two chambers, while the discharge valves are connected to the branch *Pi*, by which the primary solution is injected into the main pipe of the irrigation facility [6].

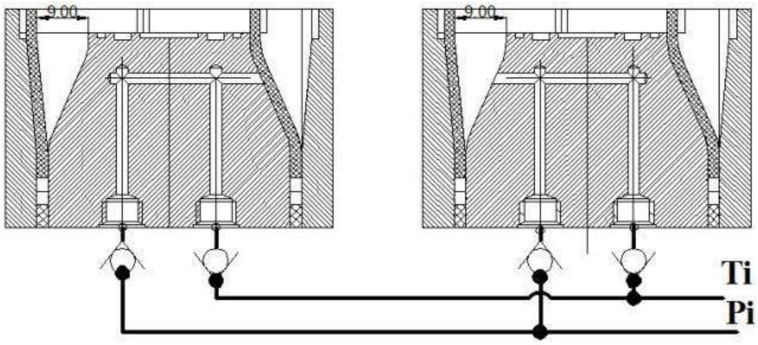


**Fig. 7.** Mounting scheme of the primary solution inlet /discharge valves

*Ti*-branch of primary solution intake valves;
*Pi*-branch of primary solution discharge valves.

The primary solution intake/discharge process takes place continuously in both directions of pump mobile assembly displacement, by varying continuously the volume of drive chambers and the volume of primary solution inlet/discharge chambers. Water inlet in one of the drive chamber determines its volume to increase, and respectively causes decrease in volume of the primary solution inlet/discharge chamber from which it is separated by the membrane, generating injection; at the same time the opposite drive chamber decreases its volume, water (after it played the role of driving fluid) being discharged through the drainage to the outside, and in the inlet / discharge chamber, by increasing volume, the primary solution intake process takes place. Depending on whether pressure or depression exists in the inlet/ discharge chambers, needles of valves component of the hydraulic axle are on seat or spun off. It should be mentioned that needle type valves have preferential installation position, so that, disengaged, the needle be in contact with the seat.
The injection device developed in EM phase is shown in Figure 8.

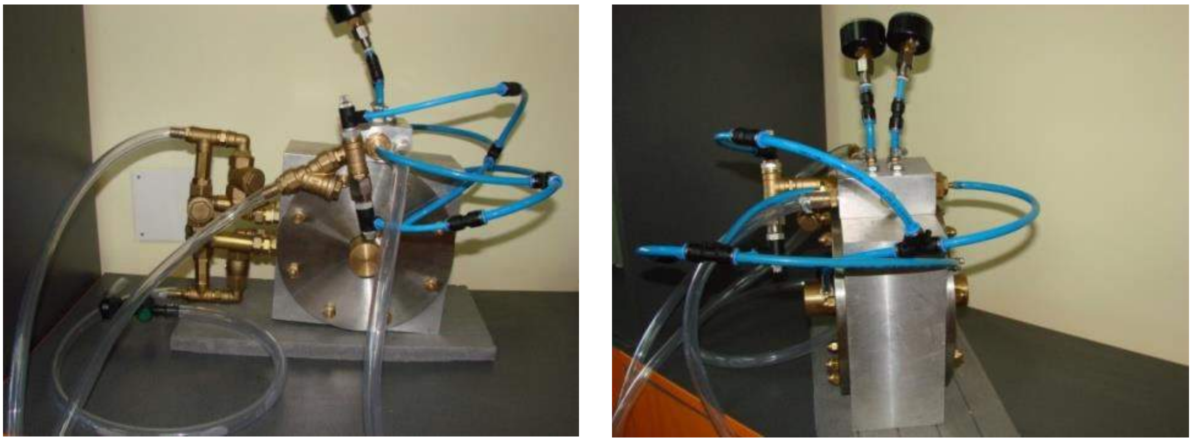

**Fig. 8.** Injection device - EM phase

## 3. Elements of calculus specific to fertigation [2]

- ***Calculation of water volume required by the irrigation facility***

1. Calculation of the area to be irrigated, S ($m^2$):

$$S = L \times l, \quad (1)$$

where: L- watering length of the installation, (m); l- watering width of the installation, given by the number of lines equipped with distribution devices (drip or micro sprinkler) and the spacing between the rows of the horticultural crop, (m).

2. Number of devices for distribution of the irrigation water, imposed by the spacing of planting for a certain horticultural crop along a row (in-row spacing):

$$N = n_l \cdot l_d, \quad (2)$$

where: $n_l$- the number of lines for distribution of irrigation water; $l_d$- optimal spacing between distribution devices, determined through tests performed under real operating conditions, (m).

3. Flow rate of the distribution device $q_d$, imposed by the horticultural crop irrigation technology (l/h); for the horticultural species of apple tree, the recommended flow rate of the dripper is 2 l/h.
4. Flow rate of the facility, $Q_i$ (l/h):

$$Q_i = N \times q_d, \quad (3)$$

where $q_d$- flow rate of the distribution device, (l/h);

5. Pluviometry of watering, p (mm/h):

$$p = Q_i : S \quad (4)$$

6. Irrigation quota, m ($m^3$/ha), imposed by the horticultural crop irrigation technology;
7. Duration of watering (irrigation time), T (h):

$$T = m / p \quad (5)$$

8. The volume of water to be administered, V ($m^3$):

$$V = m \times S \quad (6)$$

- ***Water consumption of the facility***:

$$MET = PET \times K_c, \text{ [mm]} \quad (7)$$

where PET is potential evapotranspiration, in (mm); $K_c$ - crop coefficient.

- **Preparation of the primary solution**

*The primary solution* to be introduced into the irrigation system in order to produce *the fertilizing solution* can be:
- in the state of a solution made by the manufacturer, which is diluted;
- made by mixing the fertilizers with other chemicals.

Some systems inject the solution directly from cans, others from one, two or three containers.
Choosing the chemical solution to be used is dependent on a number of factors:
- the nutrient content and the proportion of each item; correspondence in milliequivalents for the nutrient solution; the form of certain elements (nitrogen and ammoniacal nitrogen);
- other items which cause adverse effects (chlorine for certain plants);
- the desired effect in terms of pH value of the solution (the acid effect of various fertilizers).

- **Compatibility**

Certain chemicals must not be mixed in the tank, and others should not be injected simultaneously into the system, others depend on the quality of the water used (fertilizers containing phosphates should not be used in tanks containing calcium). This parameter is usually presented in tables provided by the manufacturer.

- **Solubility of chemical fertilizers commonly used in fertigation**

The solubility is the maximum amount of fertilizers that can be dissolved in a certain amount of water (100 liters). It depends on the fertilizer itself, the composition of each, the possibility of mixing chemical fertilizers, their temperature and pH.

- **Concentration of injected solution**

The injection equipment inserts the *primary solution* (of concentration $C_m$) in the irrigation water existing within the irrigation facility in order to produce the *fertilizing solution* (end use solution), of concentration $C_s$). The concentration of primary solution $C_m$ is calculated by using the formula:

$$C_m = \frac{M}{V} \quad [g/l] \tag{8}$$

where:
M is the amount of solid fertilizers which is dissolved in a given volume, in (g);
V- volume of water in which fertilizers have been dissolved, in (l). This volume must be smaller than the water volume in which chemical fertillizers reach saturation.
A dilution takes place at the injection point, depending on the flow (Q) of the irrigation facility and *the injection flow* (q) of the equipment injecting the primary solution.

- **Injection dosage (r)** and **dilution dosage ($C_s$)**

$$r = \frac{q[l/h]}{Q[l/h]} \tag{9}$$

where r is expressed as per-cent (%) or per-mille (‰).
Concentration of the end use solution ($C_s$) is determined by using the multi-line equation:

$$C_s[g/l] = C_m[g/l] \times r \tag{10}$$

$$C_s\,[\%\ volum] = C_m\,[\%\ volum] \times r$$

## 4. Monitoring the injection process performed by injection devices which are component of the fertigation equipment [2]

If the injection is performed by means of a positive displacement pump, to calculate the pump flow **q**, there must be known the volume **Vs** of the primary solution injected per stroke and the pump frequency **f**. Because the volume $V_s$ of the primary solution injected per stroke is a value imposed by design, pump frequency is calculated as follows:

$$f = \frac{n[\text{strokes}]}{t\ [\text{min}]} \tag{11}$$

where n is the number of strokes performed; t – time for performing the strokes.
Flow of the injection equipment is calculated as follows:

$$q\ [\text{l/h}] = 60 \times f\ [\text{strokes / min}]\ x V_s \tag{12}$$

Flow of the irrigation facility is calculated as follows:

$$Q\ [\text{l/h}] = N \times q_i, \tag{13}$$

where: N is the number of distribution devices (drippers, sprinklers); $q_i$ – average flow of distribution devices, [l/h].
*In the case of using soluble solid fertilizers,* where the primary solution is produced by the person who performs the irrigation, time $T_f$ is calculated by using the formula:

$$T_f[\text{min}] = \frac{60 \text{ X } M[g]}{Q\left[\frac{1}{h}\right] \text{X } C_S\left[\frac{g}{l}\right]} \tag{14}$$

*In the case of using fluid fertilizers* (which represent the primary solution), time $T_f$ is calculated by using the formula:

$$T_f[\text{min}] = \frac{60 \text{ X } M[g]}{Q\left[\frac{1}{h}\right] \text{X } C_S[\%]} \tag{15}$$

The value of $T_f$ time parameter must be smaller or equal to duration of watering T, to ensure environmental protection.
If the irrigation facility performs fertigation on the go, than fertigation time is equal to the irrigation time (duration of watering):

$$T_f = T \tag{16}$$

where: T is irrigation time; $T_f$ - fertigation time, in minutes.

## 5. Results and discussion

When designing the injection device we started from the specific requirements of the mentioned horticultural crops (technical parameters of irrigation and fertigation) and also from the technical and functional features of the irrigation facilities that the fertigation equipment works together with as a single unit [3].
For instance, for the horticultural crop of apple tree- sensitive varieties, located on the experimental plot belonging to the Research Institute for Fruit Growing ICDP Piteşti-Mărăcineni, which is project partner, these parameters and features are:
- The pipeline conveying water to the land plot (main pipeline)- Ø 100 mm;
- The pipeline connecting the fertigation pump to the main pipeline - Ø 60 mm;
- The irrigation hoses arranged on the rows of trees, on which the distribution devices (drippers or mini sprinklers) are located - Ø 16 mm;
- Spacing between the distribution devices along the irrigation hoses – 0.5 m
- Flow rate of distribution devices - 2 l/h;
- Length of the rows of trees (length of the irrigation hoses) - 160 m;
- Number of hoses (no. of rows) - 45;
- Fertilizing quota (primary solution administered during a irrigation sequence) - 100 ...150 l;

- Time of administering - 60 ...120 min.

A simulation of fertilizing solution distribution in a drip irrigation facility with the above mentioned technical and functional features has been developed by using ANSYS Fluent software and presented in HIDRAULICA journal issue 4/2015 [7].

## 6. Conclusions

- Injection of fertilizers can be done in any point of the irrigation arrangement, the drive fluid being the irrigation water itself;
- To increase pump efficiency it is recommended to install several throttle devices (tap valves, diaphragms) between the connection points of the injection device to the irrigation facility;
- The irrigation facility may be of the type: dripper, perforated tubes, or micro sprinkler installation;
- From calculations it results that the flow *q* required by the injection device is 2.5 l /min;
- Working pressures can be in the range 0.5 – 3 bar (imposed by working pressure of distribution devices in the structure of the irrigation facility).

**Acknowledgement**

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**References**

[1] M. Avram, "Actionari hidraulice si pneumatice. Echipamente si sisteme clasice si mecatronice"/ Hydraulic and Pneumatic Drives. Classical and Mechatronic Systems and Equipment, Academic Publishing House, Bucharest, ISBN 973-7787-40-4, 2005;
[2] I Biolan., I. Serbu, Gh. Sovaiala, F. Mardare, "Tehnici si tehnologii de fertirigare a culturilor agricole"/ Techniques and technologies for fertigation of crops, AGIR Publishing House, Bucharest, ISBN: 978-973-720-344-1, 2010;
[3] Guiding principles for the fertilization of fruit plantations *agroromania.manager.ro/.../principii-de-baza-in-fertilizarea-plantatiilor-pomicole*
[4] Diaphragm pumps – Tapflo *tapflo.ro/products/diaphragm*
[5] Double diaphragm pumps - Verderair VA www.verder.ro/pompe/pompe-cu-membrane/pompe-cu-membrane ;
[6] Gh. Sovaiala, Innovative technologies and equipment for implementation of the modern concept of fertigation in irrigated agriculture - Scientific and Technical Report (STR) Phase II, Project: FERTIRIG, PN-II-PT-PCCA-2013-4-0114, Financial Agreement: No. 158/2014;
[7] P. M. Cârlescu, O.-R. Corduneanu, I.Țenu, Gh. Șovăială, G. Matache, S. Anghel, N. Tănăsescu, "CFD Study on the Distribution of Fertilizer in the Fertigation Plant", "HIDRAULICA" Magazine of Hydraulics, Pneumatics, Tribology, Ecology, Sensorics, Mechatronics, (No. 4/2015), ISSN 1453 – 7303, pp. 23-31.

# Alternating Flow Hydraulic Generator for Water Jet Cutting Systems

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***Abstract:*** *The paper describes new approaches regarding the water jet machining systems. There are presented some aspects of the design possibilities of the water jet cutting head in the context in which the main hydraulic power supply is not a conventional one, but one who can provide directly an alternating flow, considering also the involved disadvantages of this solution.*
*The second part present a some concepts regarding the design of an alternating flow hydraulic generator, and details regarding the functioning conditions which can increase the reliability of the alternating flow generator and also of the entire system.*

***Keywords***: *alternating flow, hydraulic generator, water jet cutting*

## 1. General aspects

The water jet cutting technology was patented in 1968 by Dr. Norman Franz under the name "jet cutting technology at high pressure" from the McCartney-Ingersoll Rand, U.S.A.
After three years, in 1971, the water jet cutting process was implemented on the first industrial facility, and in that way the Flow Systems Company from U.S.A. became its main promoter in the industrial sector and the world leader in this technology. [3], [4]
The first steps in industrial applications have been made in the aviation industry, where the jet cutting technology was used for processing materials such as plastic, multilayer materials, composite, and others.
In 1984, the Flow Systems Company produces the first water jet cutting-abrasive installation. This innovation constitutes a genuine revolution in jet cutting technology, giving it a whole new dimension. The technique enables cutting of most materials, metallic or non-metallic, special alloys, composites and ceramics. This versatility gives it a good position compared to other recent cutting technologies (ex. plasma, laser). [3], [4]
In general, the abrasive water jet cutting processes are used for cutting operations, but there are some other applications for operations like turning, milling, or deep drilling.
The main parts of any water jet cutting machine are: the hydraulic pump, which provides pressured oil, an amplifier, in which the pressure of water is increased and the cutting head, where the water is mixed with abrasive from the system, if the processing conditions require that.
Like [4] presents, after the experimental researches with a sonic wave generator, as primary source of hydraulic energy, the conclusions were: transmitting energy in the form of pressure waves from sonic generator to cutting head allows mounting the cutting head amplifier into the cutting head, which offers the advantages of water transport pipeline working pressure metal or Kevlar and also, that the amplifier has very low reliability. Improving the system performances require an increasing of the operating pressure and reliability. For this purpose, the proposed changes are: the increasing of the pressure wave amplitude provided by the sonic generator and the increasing of the amplification ratio in the cutting head.
The sonic wave generator, previously mentioned, is an alternating flow generator which can provide, during the functioning of the system, a pressure and a volumetric flow in each connection pipe, which varies in a harmonic way, around an average value.
Generally, an alternating flow driven hydraulic transmission consists in an alternating flows and pressures generator and a motor, the connection between them being realized with a number of pipes, equal with the number of phases. The pipes must be first being filled with hydraulic oil at a pre-established pressure, obtaining in this way the correct functionality of the entire alternating flow hydraulic system. This pressure is modifying itself, naturally, during the functioning. [5]

## 2. Approaches on the water jet cutting head design

Water jet machines can manufacture parts to very good tolerances. Today, the modern water jet cutting machines can create parts with a tolerance as small as 0.05 mm, although it is usually easier to obtain tolerances under 0.1 mm. [3], [4]
The productivity and costs of processing are other important factors which determine the parameters of the abrasive jet processing and sometimes they may be even critical in choosing this process.
In [2] is described a water jet machining system comprises a alternating flow generator with the role of transforming mechanical energy produced by the electrical motor in pressure waves, that are transmitted through a hydraulic fluid to the amplifier mounted in the cutting head. The amplifier consists of two pistons with different diameters.
The disadvantage of this solution is the difficulty of obtaining a reliable piston seals that act directly on the water at high pressure.
Also, another problem is the reliability of the alternating flow generator, due to high pressure and low displacement. To eliminate these disadvantages, was proposed a solution using an amplifier with membranes, which can increase the amplification ratio and avoid water penetration in sealing area.
In figure 1 and in figure 2 is presented the architecture of a cutting head with amplifier for alternating flow.

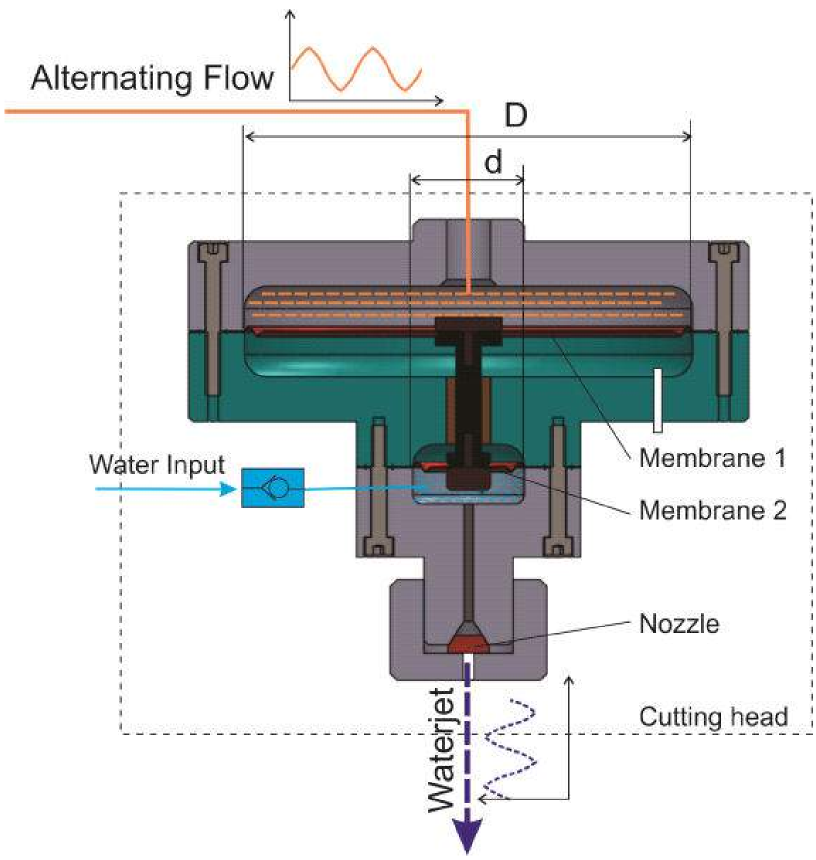


**Fig. 1.** Water jet cutting head without check valve and dumper

The pressure waves from alternating flow generator acts to the large membrane 1 that move up-down together the small membrane 2 and increase the water pressure due amplifier ratio. A continuous jet is obtained using a check valve and a dumper after amplifier, figure 2.
Also, to increase the performances and the lifetime of the alternating flow generator, this one can be designed with two functional opposite phases. But, in this way, between the cutting head amplifier and the generator we must include other specific components which provide a single exit

way for the main hydraulic power supply, using for this purpose some hydraulic pressure transformers and one way directional valves. These are not represented in the principle schema.

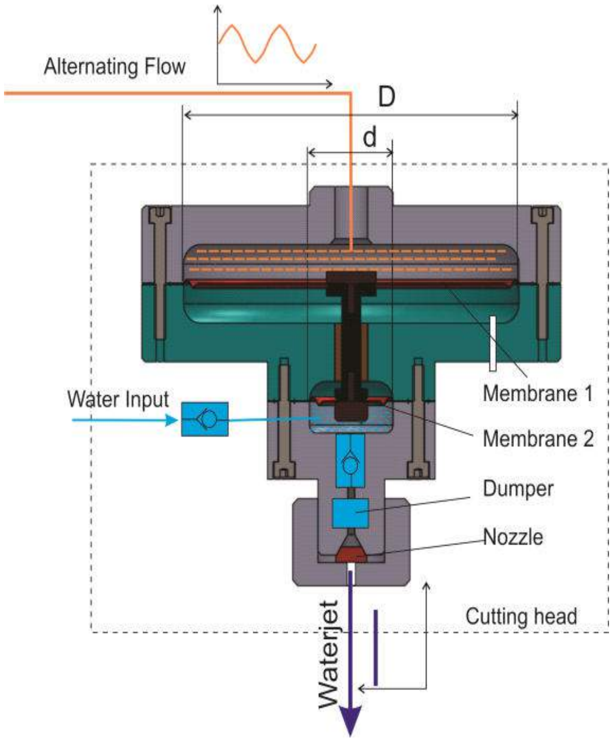


**Fig. 2.** Water jet cutting head with check valve and dumper

A cutting head with amplifier for pulsed jet that permit to obtain the amplifier ration over 20:1 is shown in figure 3.

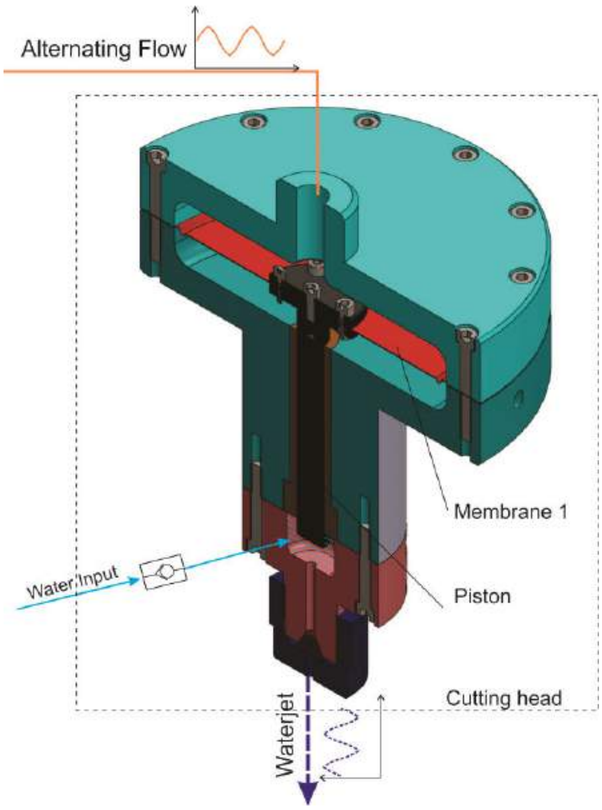


**Fig. 3.** Water jet cutting head for pulsed jet

### 3. Approaches on the alternating flow generator design

A hydraulic transmission using alternating flows involves a bidirectional displacement of a finite volume of hydraulic oil through the connection lines between a hydraulic generator and at least one small hydraulic linear motor. [5]
If the system is working in two phases, then the equations that governing the volumetric flow (instantaneously flow) for the each phase, according to [5] are:

$$Q_{i1} = Q_{a\max 1} \cdot \sin(\omega t + \varphi_0) \tag{1}$$

and

$$Q_{i2} = Q_{a\max 2} \cdot \sin(\omega t + \varphi_0 + \pi) \tag{2}$$

in which

$$Q_{a\max} = \omega \cdot r \cdot S \tag{3}$$

is the amplitude of alternating flow for the each phase.
Figure 4 presents the principle design of a biphasic alternating flow hydraulic generator with two opposite radial pistons.

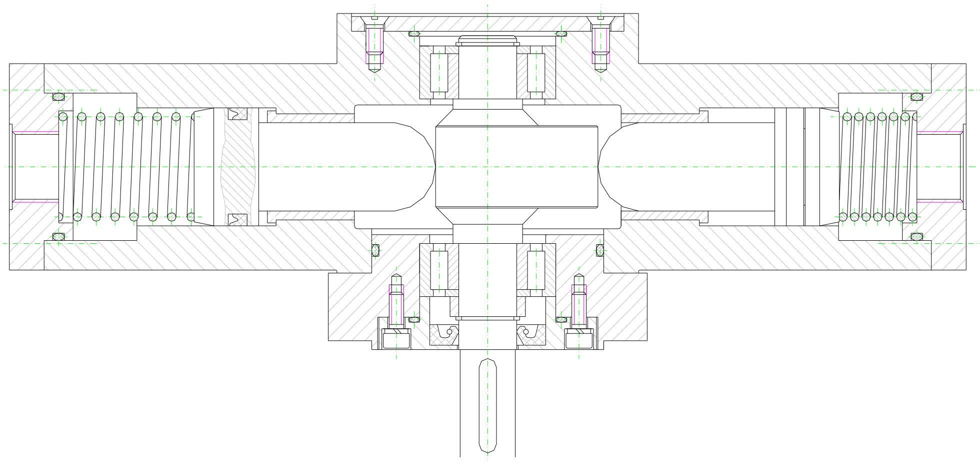

**Fig. 4.** The principle design of the biphasic alternating flow hydraulic generator

Figure 5 present the evolution of the piston volume and volumetric flow for the each phase of the alternating flow generator, based on equations (1) or (2) and (3), and also considering the established dimensions of the main components.

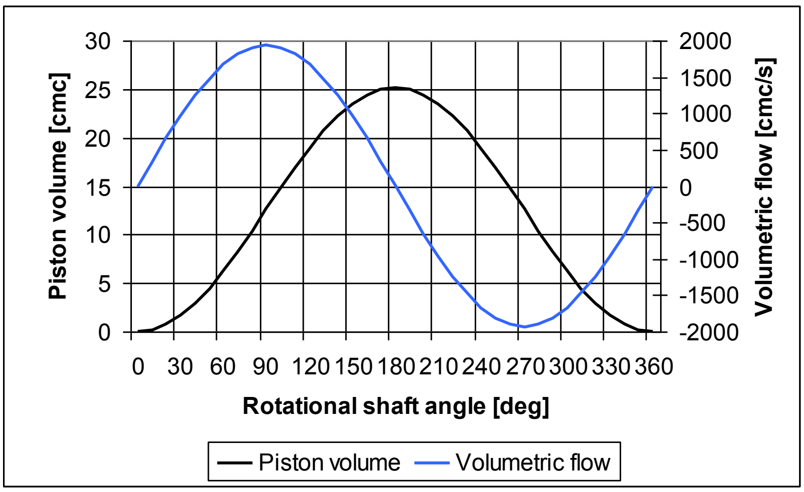


**Fig. 5.** Theoretical evolution of the piston volume and volumetric flow of the generator

The main shaft of the generator, presented in figure 4, can be realized using sinusoidal cam profile, preventing in this way the pressure shocks. According to the constructive parameters of the alternating flow generator and the sinusoidal cam law, the obtained stroke of the pistons is presented in figure 6.

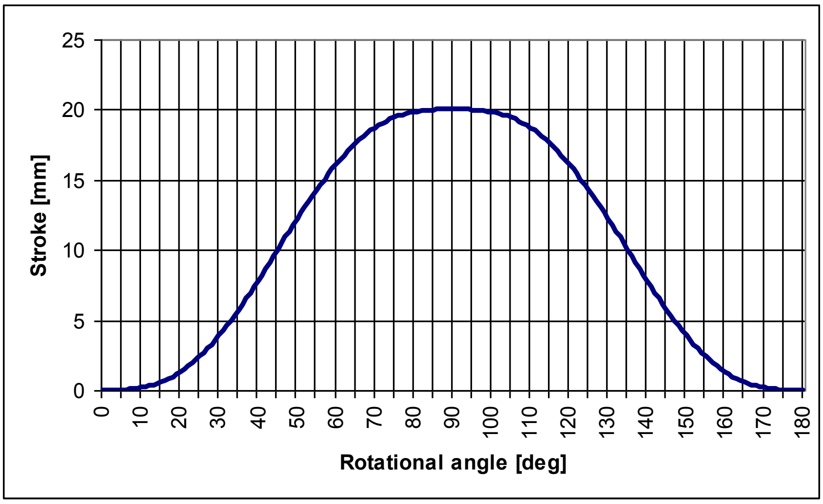


**Fig. 6.** Theoretical evolution of the alternating generator piston stroke

In figure 7 is present the principle schema of the biphasic alternating flow hydraulic system which provides the alternating flow for the membrane amplifier of the water jet cutting heat.

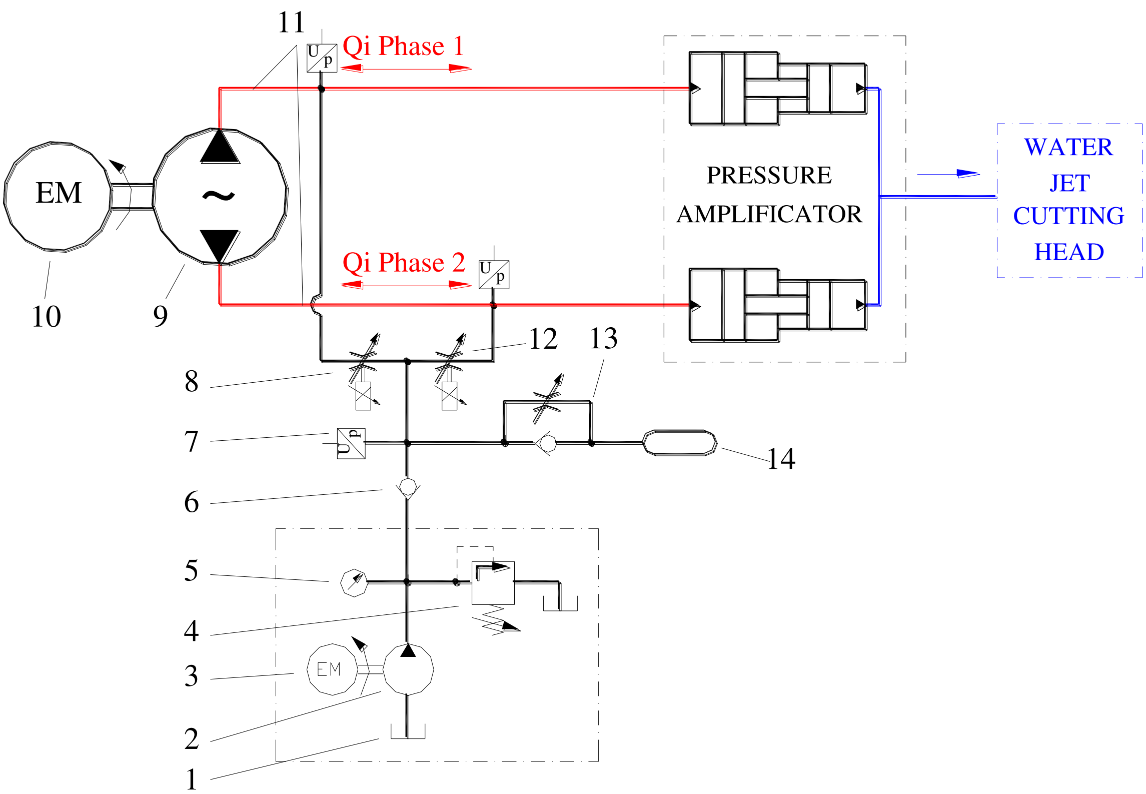


**Fig. 7.** The principle schema of the biphasic alternating flow hydraulic system

During the functioning of the alternating flow system, the pressure and the flow within each pipe varies in a sinusoidal way, around an average value, and not continuously like in the classical

hydraulic systems. These can be easily controlled with a monitoring system, deigned like in [1] and [6] is presented.
As figure 7 shows, the alternating flow generator is functioning in a closed circuit. Due to that fact the accumulator 14 presence is compulsory, and it makes the pressures in the connection lines to be all the time approximately constant. Also, the hydraulic accumulator is able to take over the oil surplus from the dilatations and in the same time to complete any external oil loses (in drains).
Between the phase lines must be placed some hydraulic resistances, proportional controlled in this case. These can eliminate the maximum average pressure rising value in one second when the diminution of the flow amplitude from a phase does not exceeds 1%, according with the researches presented in [7].
The medium oil pressure value in the accumulator is provided by using an auxiliary hydraulic system which includes a small volumetric hydraulic pump 2.

## 4. Conclusions

The paper describes the constructive principles of an innovative water jet cutting head. The membrane amplifier offers the possibility to obtain a very compact design, with low inertial forces.
The uses of the alternating flow generator, together with an adequate pressure transformer, provide for the cutting head amplifier the right form input signal.
All of these, combined with sensor and computer application especially developed, can to collect and control in an optimal manner functional parameters of the entire system.

**References**

[1] D.V. Banyai and I.L. Marcu, "Adaptive Robust Control Applicable on Variable Pumps", In: Hidraulica no. 2/ 2013, pp. 16-24, ISSN 1453-7303;
[2] C. Ciupan, M. Galis, L. Morar and A. Pop, "Water Jet Processing System", Patent RO121987/2008;
[3] C. Ciupan, E. Ciupan and S.A. Ferect-Pipas, "Neural Model for Abrasive Water Jet Cutting Machine", In: Nonconventional Technologies Review, no. 2/2013, pp. 25-29, ISSN: 1454-3087;
[4] R. Petrus, C. Ciupan and E. Ciupan, "A New Solution for Water Jet Cutting Machine Tool", Proceedings of 2012 International Salon of Hydraulics and Pneumatics–HERVEX 2012, 7- 9 November, Calimanesti-Caciulata, Romania, pp. 305-308, ISSN 1454-8003;
[5] I. Pop, I.L. Marcu, M.A. Khader et al., "Sonics Applications. Experimental Results", Performantica Publishing House, Iasi, 2007.
[6] R. Radoi, I. Dutu and G. Matache, "Data Acquisition System for Static and Dynamic Testing of Pneumatic Axes", In: Hidraulica no. 4/ 2013, pp. 18-22, ISSN 1453-7303;
[7] V. Saceanu, "In Improvements in the Functioning of the State-Of-The-Art Multiphase Sonic Transmissions, by Using Hydraulic Resistances Between the Phases as well as Hydraulic Accumulators", In: Scientific bulletin of the "Politehnica" University of Timisoara, Transaction of Mechanics, Tom 49 (63), Special Issue, October 2004, ISSN 1224-6077, pp. 531-536.

# A Review of Heat Engines

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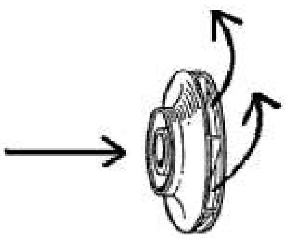
***Abstract:*** *Engines are common engineering devices which have become essential to the smooth running of modern society. Many of these are very sophisticated and require infrastructure and high levels of technological competence to ensure their correct operation, for example, some are computer controlled, others require stable three phase electrical supplies, or clean hydrocarbon fuels. This project focuses on the use of novel Stirling engine which can be used to pump water up to certain distance without use of expensive material. Some models working on this model have also been discussed.*

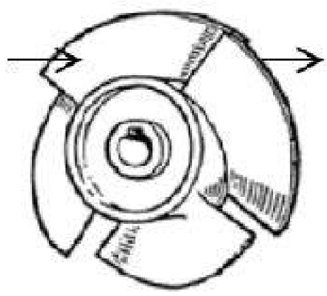
***Keywords***: *Engines, Pumps*

## 1. Introduction

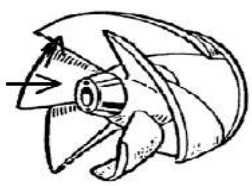
Pumps are the most important mechanical devices that play an important role in our daily lives. They have been used in the form of Persian wheels or water wheels since ancient times for irrigation purposes. They cause displacement of the working fluid by adding energy to it.
A common way to classify these pumps is on the basis of method of addition of energy to the working fluid. On the basis of this classification, pumps can be classified as following:
1) Dynamic pumps-in these pumps, energy is added to increase velocity of fluid. They may be further classified as axial flow, Radial flow or mixed flow pumps.
In Radial flow pumps, fluid enters axially and flows out radially. They are more suited for high volume flow rates and high heads, whereas in axial flow pumps, fluid enters and leaves axially. These pumps are suited for low heads and low flow rates.
In mixed flow pumps, fluid enters axially and leaves both radially and axially. They are more suited for medium heads and medium flow rates.



a) Radial pumps



b) Axial pumps



c) Mixed Flow Pumps

**Fig. 1.** Dynamic pumps

2) Displacement type-energy is added periodically by application of force to the fluid boundary. Piston pumps are common type of such devices.

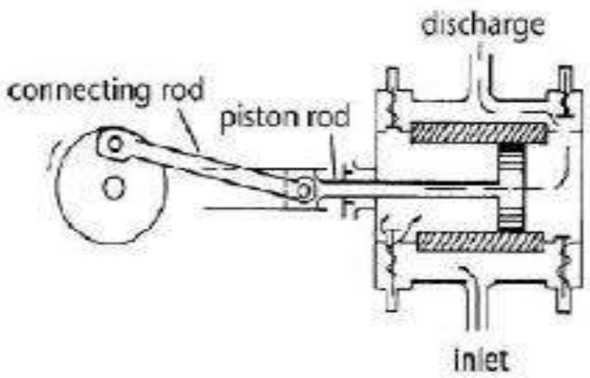



**Fig. 2.** Piston engine

## 2. Historic development of Stirling engines

*As you enter the past, you will find direction for the future*"- Ivo Kolin,[2]

During the industrial revolution of 18th century, steam engine became a primary source of power. But this device has its own drawbacks. Its maximum efficiency is at the most 2% and there were many accidents involving explosions. This prompted engineers to look for alternative sources of power like Stirling engines.
A Stirling engine is a hot air engine operating on the principle that air expands on being heated and contracts on being cooled. These devices have zero exhaust and are external combustion engines, hence wide variety of fuels can be used to run a Stirling engine which include alcohol, bio -products or waste gases etc These engines are suitable for operations which have following needs**:**
A) Constant power output.
B) Noise less operation.
C) Long startup period.
D) Low speeds.
Development of Stirling engine is widely attributed to the Scottish scientist Sir Robert Stirling. The first version of this engine developed in 1815 was heated by fire and air cooled. Figures of some of these early versions are presented in coming sections.

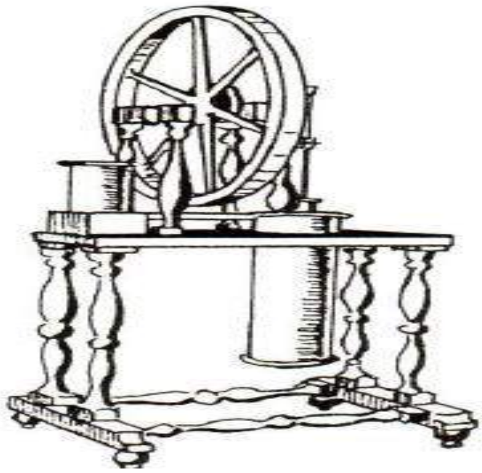


**Fig. 3.** Earliest version of a Stirling engine developed by Stirling brothers

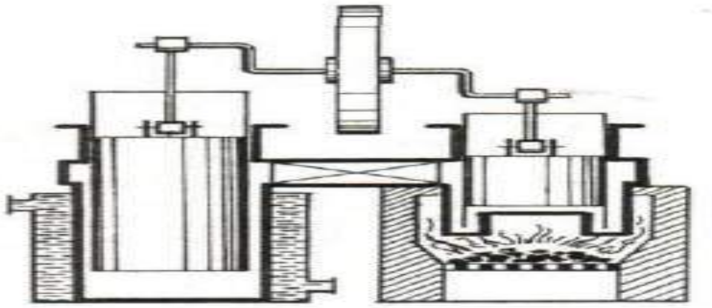


**Fig. 4.** Alpha type Stirling engine developed in 1875

Later Erickson in the year 1864 invented the solar powered engine to heat the displacer tube at hot side. The heat was obtained by use of solar reflectors. First alpha type engine was built in the year 1875 by Rider. Reader and Hooper proposed the first solar powered heat engine for irrigation purposes in the year 1908. Following this Jordan and Ibele designed a 100W solar powered engine for pumping of water. In year 1983 a low temperature difference Stirling engine was patented by the White having an efficiency of about 30%.Colin later presented a design with a low temperature difference of 15°C & Senft published specifications of a engine with very low temperature difference of 5°C between hot and cold ends [3].
Some of following events can be considered as important milestones in the design and development of a Stirling engine for use as a pump:

1688: Thomas Savery develops a drainage pump which was a liquid piston machine.
1909: Development of Humphrey pump.
1931: Malone designed and developed an engine with regenerative cycle similar to a Stirling engine.
1965: Philips Company patented a Stirling engine.
1977: The metal box company develops Stirling engine for irrigation purposes in Harwell lab.
1985: McDonnell designed an engine with parabolic reflectors to focus solar energy thus achieving a high temperature of 1400°C.

**Fig. 5**. McDonnell Engine

## 3. Heat engines [4]

A thermal engine is a device which converts heat energy into mechanical energy. The operation of a heat engine can be described by a simple thermodynamic cycle as follows:

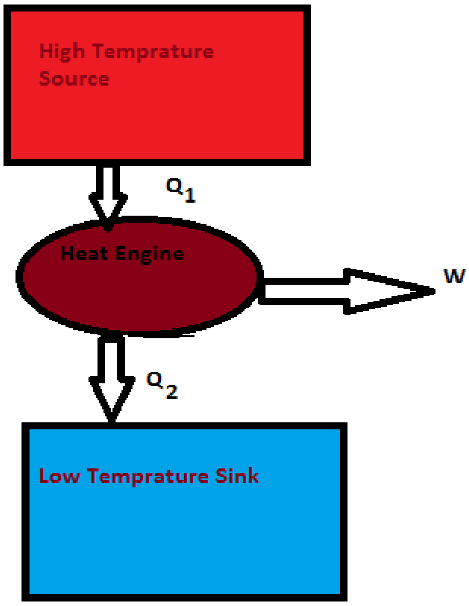



**Fig. 6.** Heat engine

Efficiency

$$\frac{W}{Q_h} = \frac{Q_h - Q_c}{Q_h}$$

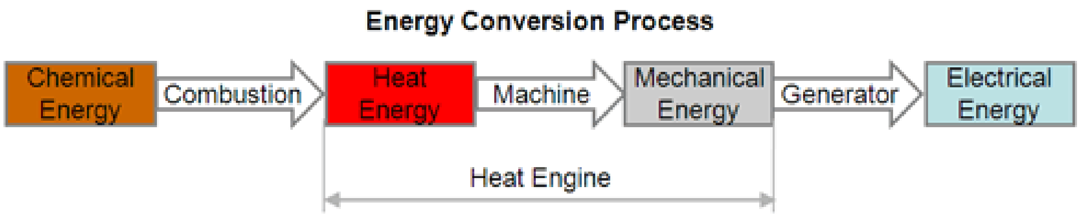



**Fig. 7**. Energy conversion in a Heat Engine

Heat engines can be further classified as external combustion engine or internal combustion engine. An engine where fuel is burnt outside the engine is an external combustion engine, whereas in the internal combustion engine, the fuel is burnt inside the engine. An engine operating on a Carnot or Stirling cycle is an example of an external combustion engine while one operating on an Otto or Diesel cycle is an internal combustion engine. Comparison of these cycles is presented below:

**TABLE 1:** Comparison of various engines

| **Type of combustion** | **Cycle** | **Compression** | **Heat addition** | **Expansion** | **Heat Removal** |
|---|---|---|---|---|---|
| External | Carnot | Adiabatic | Isothermal | Adiabatic | Isothermal |
| External | Stirling | Isothermal | Isometric | Isothermal | Isometric |
| Internal | Otto | Adiabatic | Isometric | Adiabatic | Isometric |

## 4. Conclusion [5, 6, 7]

Parabolic mirrors can be used to focus solar energy for operation of a liquid piston engine; such a device is shown below.

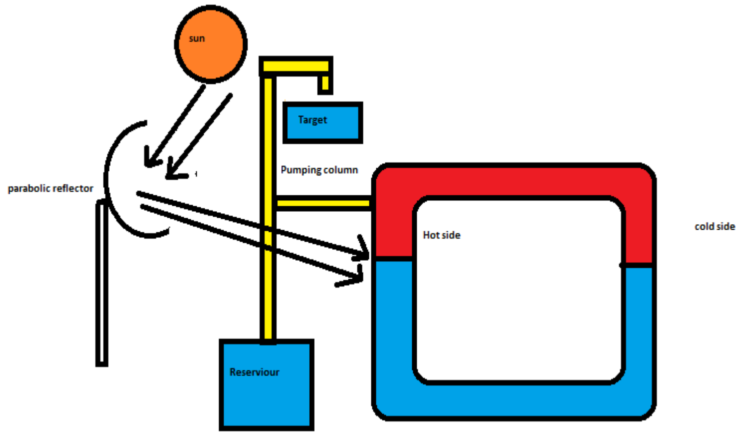



**Fig. 8.**Commercial set ups for solar liquid piston engine

Many commercial setups have been built, tested and operated by the team of Dr Tom Smith and Dr. Markides at the engineering department of the University of Cambridge. Typical data for cost, efficiency and $CO_2$ emissions is discussed here assuming a lift head of 10 m.

**TABLE 2:** Comparison of irrigation costs for various methods

| Mode of irrigation | Cost or irrigation per hectare per day |
|---|---|
| Electric pumps | £0.34-£0.55 |
| Diesel pumps | £0.29-£0.17 |
| Photovoltaic pumps | £1.27-£4.07 |
| Liquid piston pumps | £0.29-£1.07 |

**TABLE 3:** Comparison of pumping costs and efficiency of various methods

| Mode of irrigation | Efficiency | Pumping cost per unit power output |
|---|---|---|
| Photovoltaic pumps | 20-40% | £3.35-£10.7 |
| Liquid piston pumps | 2-4% | £1.50 -£3 |

**TABLE 4**: Comparison of emissions

| Mode of irrigation | CO2 emissions per hectare per day(kg) |
|---|---|
| Diesel pumps | 2.3-3.6 |
| Solar P-V pumps | 0.8-1.3 |

Going by the reliable data obtained, future of this technology seems to be bright and to tap the economic potential of there are several organizations currently involved in research in field of Stirling and liquid piston engines. Some of these are listed below:

1) STM co-operation-holders of various Stirling engine patents and developed a 40KW engine for use in GE hybrid vehicle.

2) Sun power-founded by Beale, pioneers in development of cryogenic coolers of capacity 35 W-7.5KW.

3) Infinia-developers of 1KW free piston engines and cryogenic coolers.

4) SES-makers of large parabolic dish operating solar power stations of 850 Mw capacities.

5) Thermo fluidics limited-formed in year 2006 by Dr Tom Smith of the University of Cambridge and supported by carbon trust is developing such pumping devices for use in Brazil, India and Ethiopia.

**References**

[1] http://www.unesco.org/water/wwap/wwdr/wwdr3/pdf/WWDR3_Water_in_a_Changing_World.pdf;
[2] http://www.engin.swarthmore.edu/academics/courses/e90/2005_6/E90Reports/FK_AO_Final.pdf;
[3] http://www.inference.phy.cam.ac.uk/sustainable/refs/solar/Stirling.PDF;
[4] http://www.mpoweruk.com/heat_engines.html;
[5] C. D. West, "Liquid Piston Stirling Engine", Van Nostrand Reinhold, New York, 1983;
[6] S. Narayan, 2015, "A Review of Design of Stirling Engines", Hidraulica no 3/2015, ISSN 1453 – 7303, pp. 18-26;
[7] S. Narayan, 2015, "Designing of Liquid Piston Fluidyne Engines", Hidraulica no 2/2015, ISSN 1453 – 7303, pp. 18-26.

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# HYDRAULICS AND PNEUMATICS RESEARCH INSTITUTE

# INOE 2000 – IHP

**HYDRAULICS AND PNEUMATICS RESEARCH INSTITUTE in Bucharest** is a research-development and engineering unit in the field of hydraulics and pneumatics – a subsidiary of the National Institute of Research&Development for Optoelectronics – INOE 2000.

The institute, holding over 55 years of experience in its activity field, is structured on research-development departments, research and test laboratories, all equipped with the latest systems and devices.

The Institute conducts three types of activities:

**A. RESEARCH**, with three main directions - areas of competence:
1. HYDRAULIC AND PNEUMATIC EQUIPMENT AND SYSTEMS
2. GREEN ENERGIES
3. HYDROTRONICS, MECHATRONICS AND TRIBOLOGY

**B. ENGINEERING**
> Upgrades of hydraulic and pneumatic systems
> Hydraulic equipment repair and adjustments
> Design, construction and commissioning of complex hydraulic systems and hydraulic automation systems, and also maintenance activities
> Design, construction and commissioning of stands with hydraulic, electro hydraulic or pneumatic drive

**C. PROFESSIONAL TRAINING COURSES**
- Professional training courses in the fields of competence of INOE 2000-IHP
- Consulting activities

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INOE 2000-IHP is a founder member
of the National Association of Hydraulics and Pneumatics
“FLUIDAS” www.fluidas.ro

INOE 2000-IHP is a co-organizer
of the International Conference on Hydraulic and Pneumatics
“HERVEX” http://hervex.fluidas.ro

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