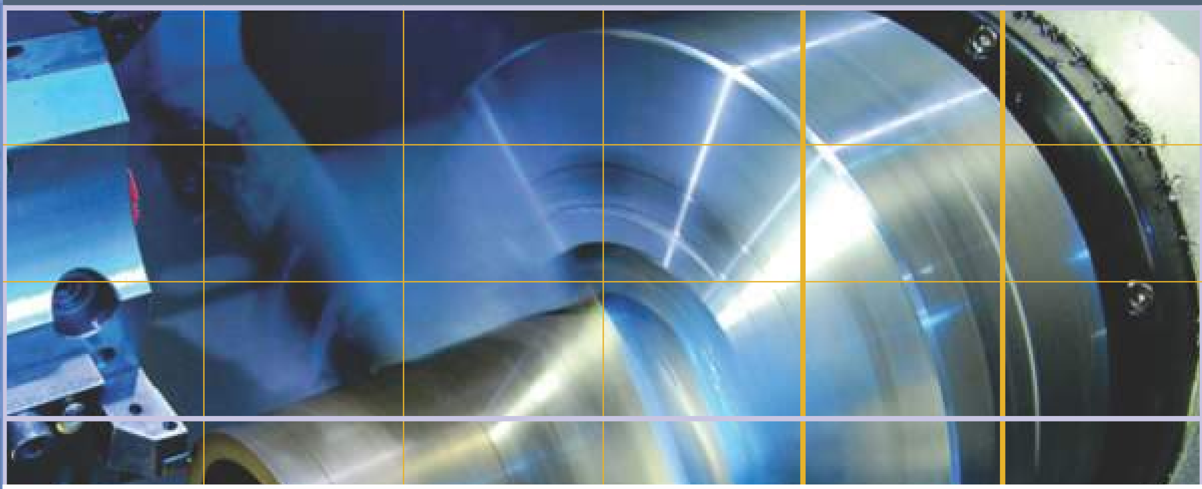
Nr. 1/2017

HIDRAULICA

HYDRAULICS-PNEUMATICS-TRIBOLOGY-ECOLOGY-SENSORICS-MECHATRONICS



http://hidraulica.fluidas.ro

ISSN 1453 - 7303
ISSN-L 1453 - 7303

CONTENTS

Published by:
Hydraulics and Pneumatics Research Institute, Bucharest-Romania
Address: 14 Cuţitul de Argint, district 4, Bucharest, 040558, Romania
Phone: +40 21 336 39 91; Fax:+40 21 337 30 40; e-Mail: ihp@fluidas.ro; Web: www.ihp.ro
with support from:
National Professional Association of Hydraulics and Pneumatics in Romania - FLUIDAS
e-Mail: fluidas@fluidas.ro; Web: www.fluidas.ro

HIDRAULICA Magazine is indexed by international databases:

EBSCO PUBLISHING ProQuest ULRICHSWEB GLOBAL SERIALS DIRECTORY INDEX COPERNICUS INTERNATIONAL AcademicKeys Google scholar Scholarsteer EZB JIFACTOR CiteFactor SIS Scientific Indexing Services INFOBASE INDEX TIB

ISSN 1453 – 7303; ISSN – L 1453 – 7303

EDITORIAL

Alternatives

Technical and especially technological progress recorded in recent years has led to a completely different layout and structure of complex equipment compared to what we had until now, although basic structure is about the same. The most important novelty as to structure consists in including mechanics (hydraulics or pneumatics), electronics, sensorics and computer science in a single block. Actually, I described, in general terms, mechatronics.

Ph.D.Eng. Petrin DRUMEA
DIRECTOR OF PUBLICATION

It is difficult to admit that there can be any complex equipment without a drive system, be it mechanical, electrical, hydraulic or pneumatic, without embedded electronics or without modern computerized control.

Interwining of these sub-parts has been so strong that, for instance, in pneumatics it is difficult to disassociate mechanics from electronics.

Integrating electronics and computer science into hydraulic systems and equipment has started long ago, with great effort, and it represents a quite clear development trend, although many months and years are still necessary until reaching a form accepted by everyone. Little by little the Fluid Power specialist has been forced to get knowledge also in related areas, such as electronics and computer science. As a matter of fact, many electronics engineers have become good mechanics specialists, so that there will be nothing strange for mechanical engineers to become good electronics specialists. This need to expand knowledge has been a must not only for researchers and designers but also for equipment users and maintenance specialists.

Most probably university curricula also need to line up to this trend which manifests internationally.

It is necessary that part of hydraulics to be reconsidered with the idea of combining it with electronics, so that we will not be forced to adjust, again and again, to each other components which are totally different and difficult to integrate. Starting early since the design phase from the need for structural and functional integration of all subsystems, the resulted unit will be easier to supervise, easier to maintain, it will be more ergonomic and more pleasant-looking. The direction of this type of integration will make it possible, in time, for a hydraulic unit to be brought up-to-date only by changes in the electronic or computerized parts of it.

IN MEMORIAM Dr. HENRYK CHROSTOWSKI

Dear Colleagues...
With deep regret and sadness we would like to inform that Dr. Henryk Chrostowski died suddenly at his home... *on 9 January 2017*. He will remain forever in our memory.
Dr. Krzysztof Kędzia*, Institute of Machine Building and Operation, Wroclaw University of Technology*

***Dr Henryk Chrostowski** during opening of periodic conference Hydraulic and Pneumatic Drives and Controls in October 2009*

Posthumous tribute to our colleague
Henryk Chrostowski, PhD, eng. (1945-2017)

It was with deepest sadness that we learned of the passing, on 9 January 2017, of our colleague with whom we had been bound with ties of friendship, partnership and collaboration in the field of research and education relating to broadly understood fluid systems and controls. We are paying our last tribute not only to the outstanding educator and researcher, but also to the brilliant organizer and technical knowledge propagator, the President of the Hydraulic Control and Drive Section of the Association of Polish Mechanical Engineers and Technicians (SIMP), the President of the Chamber of Commerce for Components and Technologies and the Vice-president of its Hydraulic Drives and Controls Corporation.

Henryk Chrostowski, PhD, eng. was born on 11 February 1945 in the Makowskie Village in the Podlasie Province. Since he was four he lived in Wrocław, which became his beloved city. In 1964 he finished the Mechanical Engineering Secondary School at the Aviation Research Centre in Wrocław, obtaining the title of mechanical engineering technician. In 1970 he graduated from the Mechanical and Power Engineering Faculty of Wrocław University of Science and Technology (WUS&T), obtaining the master's degree in mechanical engineering. In 1970 he began his professional career as junior research & teaching assistant at the Mechanical and Power Faculty of WUS&T, in the Water-Powered Machines Unit headed by Ryszard Rohatyński, PhD, eng. After restructuring, under the latter's supervision, now in the Mechanical Engineering Faculty Hydraulic Devices Modelling Unit, he defended his doctoral dissertation entitled: "Fundamental calculation problems relating to the self-acting lift valves in piston pumps" and was awarded the Doctor of Technical Sciences degree.

Henryk was and remains an important personage in my life, to whom I owe very much. My great friend, a person who wanted and was able to commit himself, but also a loving person capable of showing others friendship and love, has passed away. Farewell Henryk
Prof. Jan Koch

Why? Why such a hurry? Who or what has made you feel sad? Henryk has been one of my dearest friends, although we met in adulthood. I have always esteemed his seriousness, education and patriotism. He loved his country and made myself love Poland, too, and better understand my own country. In these moments I can only promise that for me Henryk Chrostowski will remain a great friend.
***Dr. Petrin Drumea**, President of National Professional Association of Hydraulics and Pneumatics in Romania–FLUIDAS; Hydraulics and Pneumatics Research Institute INOE 2000-IHP, Bucharest, **Romania**;*

In his work Dr Henryk Chrostowski was always an optimist, persistent in pursuing goals. Demanding of himself and others. He was a friendly person, always willing to give advice, especially to young teachers, also genuinely respected by his co-workers. He was very intelligent, but at the same time modest and with a great sense of humour. He loved books, travels and classical music…Henryk we shall miss You!
Renata Supranowicz

With sorrow in our hearts and sadness in our souls, we have received the announcement about passing away of our great Polish friend, Dr. Henryk Chrostowski, an European level expert, whom us in Romania have admired and valued very much for his expertise and deep knowledge on the evolution of Fluid Power field, both on the European and global level. Dr. Henryk Chrostowski will remain in our hearts and memories for as long as we live and we will never forget him!
Dr. Corneliu Cristescu**, Hydraulics and Pneumatics Research Institute INOE 2000-IHP, Bucharest, **Romania

Henryk passed away too quickly. I feel that I am indebted to him, similarly as most of the persons he ever met. We have not managed to redress the balance of kindness. Nevertheless, I believe that "the one who is loved will be saved".
***Leszek Jurdziak**, Lecturer at Wrocław University of Science and Technology and at Polish-American Schools of Business in Wrocław and Cracow*

It is hard for me, terrible hard, to write about a great person whom I personally knew exclusively on occasion of the scientific debates taking place in the framework of scientific conferences held in our mutual professional field. Even so, I could not help but notice the seriousness and devotion behind his lectures, which were structured flawless, I would say. Moreover, Dr. Chrostowski was a warm, open and friendly person, telling well-targeted jokes, to relax or focus the audience.
We will always miss this “giant”, both literally and figuratively, kind person. May God rest his soul in peace!
Dr. Gabriela Matache**, Hydraulics and Pneumatics Research Institute INOE 2000-IHP, Bucharest, **Romania

Henryk would never show impatience or become discouraged. There are fewer and fewer such Teachers in the whole ecumene of the taught and the teaching. And it is also for this reason that the passing away of Henryk is a loss to us all. Therefore we should remember him, long and with gratitude.
***Krzysztof Brocławik**, PhD, lecturer at the Polish-American School of Business in Cracow*

I have met Dr. Henryk Chrostowski on occasion of the scientific events held in Romania and Poland. Through all his deeds and acts, he was a leading example for those he was communicating or collaborating with. Through his remarkable professional and scientific achievements he will always be present in the memory of my colleagues at Politehnica University of Timişoara, Chair of Hydraulic Machinery.
Prof. Ilare Bordeasu,** Politehnica University of Timişoara, **Romania

I received a card from Him with a present in the form of a book and I keep asking myself the question: did he want to say goodbye to me in this way? and I neither managed to say goodbye nor thank him enough for collaboration and friendship. THANK YOU HENRYK!
“People live as long as others nurture memories of them, thinking and loving”.
***Piotr Kubica** - Chairman of the Board, Fair Centre FairExpo Ltd.*

Professor Henryk Chrostowski, PhD. eng. has been a great promoter of Fluid Power development on European level. Professor Henryk Chrostowski has been our close friend; with great generosity he has facilitated effective collaboration between Fluid Power companies, institutions, specialists in Poland and Romania, attending the Hervex conference, edition by edition, and getting involved in European level international collaboration close beside representatives of FLUIDAS, bringing his great support in promoting and developing the Fluid Power industry, education and research in the field.
We will keep in our hearts a vivid memory and gratitude in response to his generosity offered to us all.
Prof. Constantin Chiriță,** "Gheorghe Asachi" Technical University of Iași, **Romania

He was faithful to his views, could defend them in a cultured, but unequivocal way, sometimes with a note of sarcasm. He was characterized by courage and had his own opinion, regardless whether this was or was not to his benefit. This is how he is and will remain in our memory: a wise person with broad horizons, who could be relied on.
*Recollected by: **Irena Dzięgielewska, Katarzyna Gemsa** – directors of the SIMP Personnel Training Centre in Wrocław*

Since 2008 I have had the opportunity to meet Dr. Chrostowski both as a specialist in hydraulic drives and as a person with broad education in history and civilization. He knew and used to communicate great many things, still always eager to discover new things. We were looking forward to his presentations, each time with great pleasure, as each time we knew we were going to learn something new. Life goes on, naturally, but we will always miss him, both in Poland and here in Romania.
Dr. Cătălin Dumitrescu**, Hydraulics and Pneumatics Research Institute INOE 2000-IHP, Bucharest, **Romania

The European Oil Hydraulic & Pneumatic Committee CETOP has lost somebody very important and at the same time a close friend. CETOP will never forget Dr Henryk Chrostowski.
***Sylvia Grohmann-Mundschenk**, General Secretary of CETOP*

I met Dr. Henryk Chrostowski for the first time few years ago, in Kielce International Fair. Apart from his professional qualities, I have greatly esteemed his human being quality. I met him for the last time in Hervex 2016 International Conference - the same energetic, friendly and humorous person. He has been a great patriot, but also loved Romania! I am glad I had the privilege to know him and I terribly regret him pass away!
Dr. Teodor Costinel Popescu**, Hydraulics and Pneumatics Research Institute INOE 2000-IHP, Bucharest, **Romania

Henryk had one more characteristic, which was for us, for me, the most important – He simply liked people, people with all their shortcomings. One could feel it at every step.
***Darek Michalak** - Kielce Fair, Director*

I have had the great chance of meeting the now much regretted PhD. Eng. Henryk Chrostowski in international conferences on Hydraulics organized in Romania and Poland, and also at the Institute in Wroclaw. He has been a specialist with great expertise in his activity filed, extremely open to the future, agreeing to and promoting innovative technologies, concepts and ideas, also from connected areas, which led to his modern achievements with broad applicability. I have met a cheerful and optimistic person, who had great love for his native country, Poland. May God rest his soul in peace!
Prof. Erol Murad**, Politehnica University of Bucharest, **Romania
Testimonials in detail could be read on the web address http://hidraulica.fluidas.ro.

Determining the Times of Charging and Discharging of Hydro-pneumatic Accumulators

Prof. PhD Eng. **Dan PRODAN[1],** Prof. PhD Eng. **Anca BUCUREȘTEANU[2]**

[1] University POLITEHNICA of Bucharest, prodand2004@yahoo.com

[2] ancabucuresteanu@gmail.com

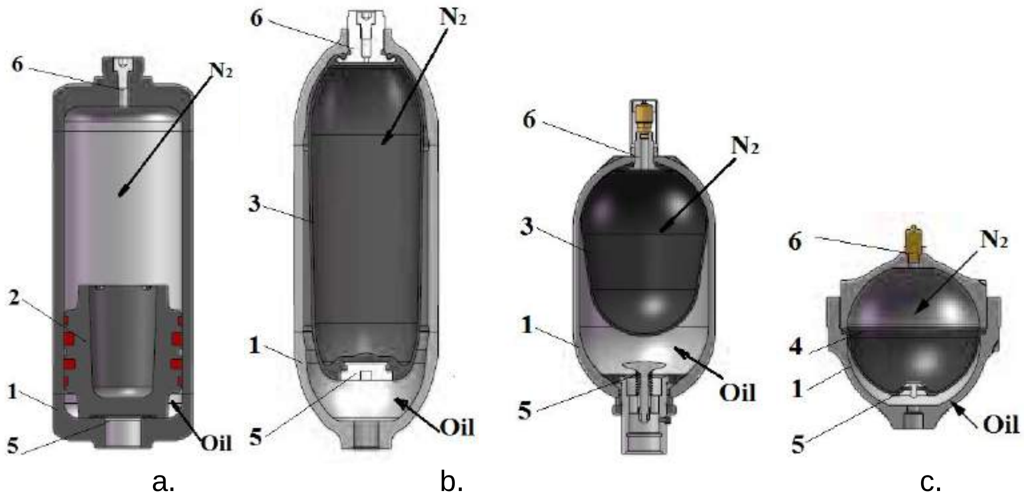
Abstract: *The authors present in this paper some research on the behavior of the hydro-pneumatic accumulators in the hydraulic installations of the machine-tools. They introduce the mathematical models that describe the phases of charging and discharging of these accumulators, taking into account the actual operation of the installation. There are also presented the results of some simulations of installations with accumulators, but also the hydraulic installation with which they checked the mathematical models developed. In this paper, the authors intended to determine the actual time in which the accumulators are able to take over the task of the pumps to supply an oil flow in a certain pressure range. Among the hydraulic installations with accumulators for which the obtained results could be used, we mention: hydrostatic suspension systems of heavy duty vertical lathes, lubrication systems of the bearings that operate on shafts with high inertial masses, hydraulic systems for safety, hydraulic systems that use accumulators as additional source of energy, hydraulic systems for balance etc.*

Keywords: *hydro-pneumatic accumulators, machine-tools, sources of hydraulic energy*

1. Introduction

Accumulators are hydraulic components that enable the reception, storage and transmission of hydrostatic energy under the form of liquid volumes under pressure. The low degree of compressibility of the liquids makes it difficult to store energy in small volumes, but allows the transmission of high efforts. Unlike liquids, gases have great possibilities in terms of compressibility, which allows the storage of a relatively high energy in low volumes. The association of liquids and gases, in special systems has led to the manufacture of hydro-pneumatic accumulators [1].

The currently used hydro-pneumatic accumulators are piston type (a.), bladder type (b.) or diaphragm type (c.). The structure of these accumulators is shown in Figure 1 [1, 2, 3, 4, 5].



a. b. c.

Fig. 1. Hydro-pneumatic accumulators: a.- piston accumulators, b.- bladder accumulators, c.- diaphragm accumulators

Notations in Figure 1 are the following ones: 1 - body of accumulator, 2 - piston, 3 - bladder, 4 - elastic diaphragm, 5 - oil inlet/outlet valve, 6 - nitrogen charging valve (N_2). In accumulator body 1, the oil and nitrogen are separated by means of the piston 2 or bladder 3 or, for smaller volumes, by an elastic diaphragm 4. Initially, the accumulator is charged with nitrogen at a predetermined pressure, by valve 6. If oil pressure is higher than nitrogen pressure, valve 5 opens and the accumulator takes over a certain amount of oil. If the pressure of nitrogen is higher than the pressure in the hydraulic installation, then the oil from the accumulator is sent to the installation (HI).
Keeping the same notations, the accumulators shown in the figure above can be presented schematically as in Figure 2.

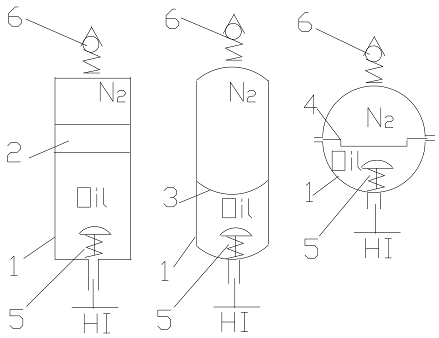


Fig. 2. Schematic representation of hydro-pneumatic accumulators

Elements 2, 3 or 4 separate the nitrogen (N_2) of the upper chamber from the oil of the hydraulic installation (HI). Further, for developing the mathematical models, we shall take into consideration the case of piston accumulators, but the results can be applied to all types of hydro-pneumatic accumulators.

2. Recharging the accumulator with oil

The accumulator defined by its total volume, denoted by V_0, is usually charged with nitrogen through valve 6 of Figures 1 and 2 at pressures p_0. If the installation where the accumulator will be assembled operates between a minimum pressure p_{min} and a maximum pressure p_{Max}, the nitrogen charging pressure checks the relations [1, 5]:

$$p_0 = k \cdot p_{min} \tag{1}$$

$$k \in [0.6 - 0.9] \tag{2}$$

When starting the hydraulic installation, the accumulator will be charged with oil as in Figure 3.

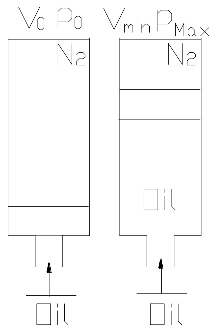


Fig. 3. Recharging the accumulator with oil

If considered that the accumulator circuit is supplied by a constant flow pump having its value Q_P and that nitrogen undergoes an isotherm transformation, it will be obtained:

$$p_0 \cdot V_0 = p_{Max} \cdot V_{min} \tag{3}$$

$$\Delta V = V_0 - V_{min} \tag{4}$$

$$Q_p \cdot t_I = \Delta V \tag{5}$$

$$t_I = \frac{V_0}{Q_P} \cdot \left(1 - \frac{p_0}{p_{Max}}\right) \tag{6}$$

In the relations (3) ÷ (6) it was also noted: V_{min} - minimum volume of nitrogen, ΔV - volume of oil that enters the accumulator, t_I - time of accumulator recharging.

3. Discharging the accumulator during operation

In the phases in which the accumulator is discharged, providing the circuit with oil, as shown in Figure 4, it can be considered that the existing nitrogen undergoes transformations in accordance with:

$$p_{Max} \cdot V_{min}^{\gamma} = p \cdot V^{\gamma} = p_{min} \cdot V_{Max}^{\gamma} \tag{7}$$

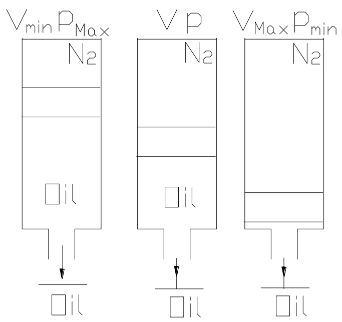


Fig. 4. Discharging of accumulator

In the equation (7) it was noted by γ the index of nitrogen transformation which has the values: $\gamma = 1$ - for the transformations that take place during tens of seconds or even minutes (isothermal) and $\gamma = 1.4$ - for the transformations taking place in matter of seconds (adiabatic).
From the relation (7) it results:

$$V = \frac{p_0 \cdot V_0}{p_{Max}^{1-\frac{1}{\gamma}}} \cdot p^{-\frac{1}{\gamma}} \tag{8}$$

The flow provided by the accumulator has the expression:

$$Q = \frac{dV}{dt} = \frac{dV}{dp} \cdot \frac{dp}{dt} \tag{9}$$

From the relations (8) and (9) it will be obtained:

$$Q = K \cdot p^{-1-\frac{1}{\gamma}} \cdot \frac{dp}{dt} \tag{10}$$

In the relation (10) it was denoted by K a constant specific to the accumulator of volume V_0, which is charged at pressure p_0 and runs at maximum pressure p_{Max}, with value:

$$K = -\frac{p_0 \cdot V_0}{\gamma \cdot p_{Max}^{1-\frac{1}{\gamma}}} \tag{11}$$

It may be considered that this discharge of accumulator is made through a throttle valve, cartridge throttle or a flow control valve towards a circuit where the useful pressure (p_U) is greater than or equal to atmospheric pressure. The condition of proper operation of the installation is that the useful pressure checks any time the relation below:

$$p_{min} \geq p_U \tag{12}$$

It is considered that accumulator discharging is performed by actuating the directional valve DV in Figure 5.

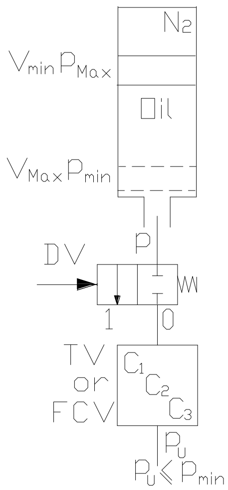


Fig. 5. Discharging of accumulator by throttle valve (TV) or by flow control valve (FCV)

In Figure 5, the constants specific to various modes of discharge (to be defined below) were denoted by C_1, C_2 and C_3.
The flow discharged by various hydraulic resistances depends on the pressure difference between input and output. In these conditions, the following three dependents can be taken into consideration:

$$Q = f_i(p) \tag{13}$$

$$f_1(p) = C_1 \cdot \sqrt{p - p_U} \tag{14}$$

$$f_2(p) = C_2 \cdot (p - p_U) \tag{15}$$

$$f_3(p) = C_3 \tag{16}$$

$$i \in \{1,2,3\} \tag{17}$$

Discharging through a throttle valve is made according to function f_1. Such discharges are found in the hydraulic installations for balance, compensation or hydrostatic [6, 7, 8, 9, 10]. If the discharging is performed through a coil, as in the case of hydrostatic guideways [11, 12], it can be considered that the flow is described by function f_2. If the discharge is made by a flow control valve [1, 11], it shall be considered that the pressure does not influence the value of flow, according to function f_3.

By equaling the expressions (10) and (13), we obtain finally, after integration:

$$K \cdot \int_{p_{Max}}^{p_U} \frac{dp}{p^{1+\frac{1}{\gamma}} \cdot f_i(p)} = \int_0^{t_U} dt = t_U \tag{18}$$

In the relation (18) we denoted by t_U - the useful discharge time of accumulator, namely the time for useful discharging of accumulator, thus the time when it covers, alone or together with the pump, the necessary of oil.
Relation (18) has general character and it will be used for making specific customizations as follows:

- Υ will be 1 for isothermal transformations (in tens of seconds or even minutes) and 1.4 in the case of adiabatic transformations (matter of seconds).
- p_U - is the pressure of the filled circuit (it may be considered as “0” if the spill is at atmospheric pressure).
- C_1, C_2, C_3 - are the constants whose value is usually given in the catalogues of the manufacturers of hydraulic equipment [4].

For example, when an accumulator is discharged in isothermal mode, towards the hydrostatic pockets of a machine tool, at atmospheric pressure, in case of failure of the pump, it will be obtained [10]:

$$t_U = \frac{2}{3} \cdot \frac{p_0 \cdot V_0}{C_1} \left(\frac{1}{p_{min} \cdot \sqrt{p_{min}}} - \frac{1}{p_{Max} \cdot \sqrt{p_{Max}}} \right) \tag{19}$$

In this case, constant C_1 has the expression:

$$C_1 = C_{TV} \cdot S_{TV} \cdot \sqrt{\frac{2}{\rho}} \tag{20}$$

In the relation (20) it was denoted: C_{TV} - constant of throttle valve (given in catalogue), S_{TV} - surface of throttling, ρ - density of the oil used.

4. Simulation of hydro-pneumatic accumulators operation [13, 14]

The following numerical values have been considered for these simulations: V_0 = 4 l, p_{min} = 100 daN/cm^2, p_{Max} = 200 daN/cm^2, p_0 = 90 daN/cm^2, Q_P = 2.4 l/min.
The accumulator is charged in conformity with the feature in Figure 6.
Charging time in these conditions is t_l = 60 s. If replacements are made in relation (6) we obtain t_l = 55 s.
To simulate accumulator discharging, it is considered that the discharge is made through a throttle valve with openings expressed in percentages related to the maximum opening. Figure 7 shows the evolution of flow and pressure when throttle valve opening is 70%.

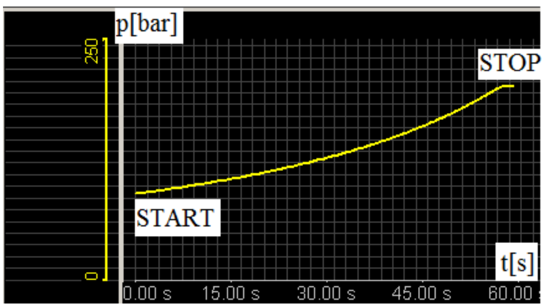


Fig. 6. Charging of accumulator

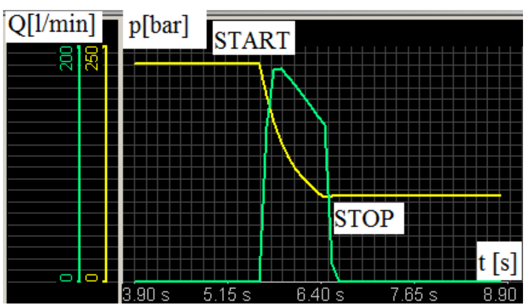


Fig. 7. Features for opening of 70%

Flow reaches values of 150 - 180 l/min for approximately 1 s.
If throttle valve is closed so the surface of throttling becomes 15% of the maximum surface, the

flow is approximately 12 - 15 l/min for about 9 s, according to the feature in Figure 8.
If the throttle valve is closed up to a value of 3% of the maximum value, we shall have the features in Figure 9.

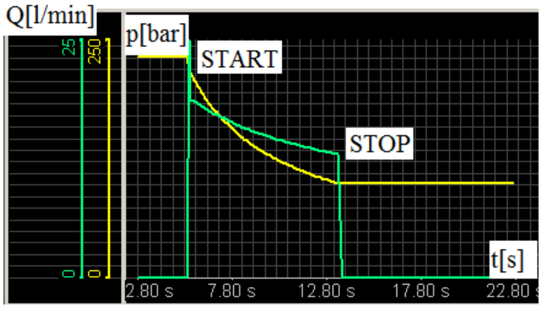


Fig. 8. Features for opening of 15%

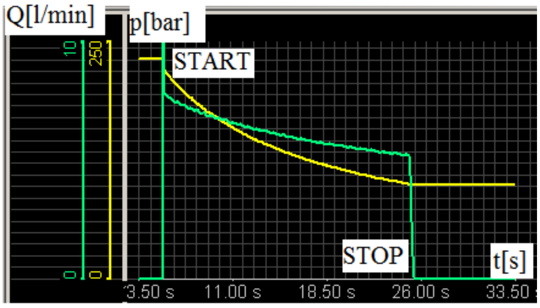


Fig. 9. Features for opening of 3%

In this case, the flow rate is between 8 l/min and 5.5 l/min for approximately 22 s.
If a flow control valve adjusted at the value Q_R = 4 l/min is used instead of the throttle valve, we shall obtain the feature of Figure 10.

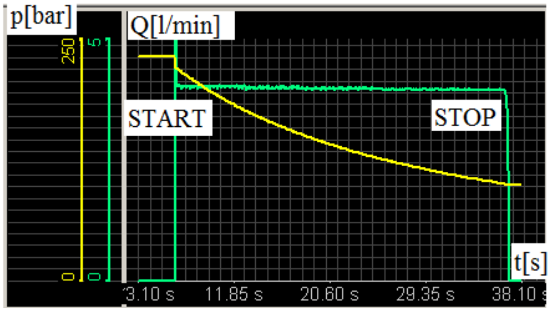


Fig. 10. Discharging of accumulator through a flow control valve adjusted at value Q_R = 4 l/min

In this case, the constant flow rate of 4 l/min is ensured for approximately 29 s.

5. Experimental determination of the times for charging/discharging of hydro-pneumatic accumulators

On the occasion of the remanufacture of a machine-tool, its hydraulic installation has been adapted so to be able to perform the measurement of times, pressures and flow rates. The hydraulic diagram of this installation is shown in Figure 11.
Notations in Figure 11 are as follows: 1 - suction filter (SF), 2 - pump (P), 3 - electric motor (EM), 4, 13 - manometers (1M, 2M), 5 - pressure valve (PV), 6, 16 - directional valves (DV1, DV2), 7 - return filter (RF), 8 - level gauge (IN), 9 - tank (T), 10 - check valve (CV), 11, 12 - pressure switches (PS1, PS2), 14 - safety block (SB), 15 - accumulator (Ac), 16 - pressure transducer (PT), 17 - throttle valve (TV), 18 - flow control valve (FCV), 19 - flowmeter (FI).
Pump P driven by the electric motor EM sucks oil via suction filter SF from the tank T. The working pressure of the pump is viewed on manometer M1. The value of this pressure is adjusted using the pressure valve PV. By actuating the electromagnet 1S of the directional valve DV1, the oil will be sent to the accumulator Ac via the check valve CV. The pressure in the circuit of accumulator is read by means of manometer M2 and pressure transducer PT. The accumulator is equipped with the safety block SB [4]. The pressure switches are adjusted as follows:

- PS1 is adjusted at minimum pressure (p_{min}) and controls the charging of accumulator circuit (1S+) and the uncoupling of the associated circuit (3S+,2S-).
- PS2 is adjusted at maximum pressure (p_{Max}) and controls the uncoupling of charging (1S-) and the coupling of the associated circuit (2S+,3S-).

By actuating the directional valve DV2 (2S+), the oil from the accumulator is sent to the throttle valve (TV) or to the flow control valve (FCV). The flow is measured by means of the flowmeter (FI). In Figure 11 it was also noted: NI - level gauge, TV - throttle valve, FCV - flow control valve.

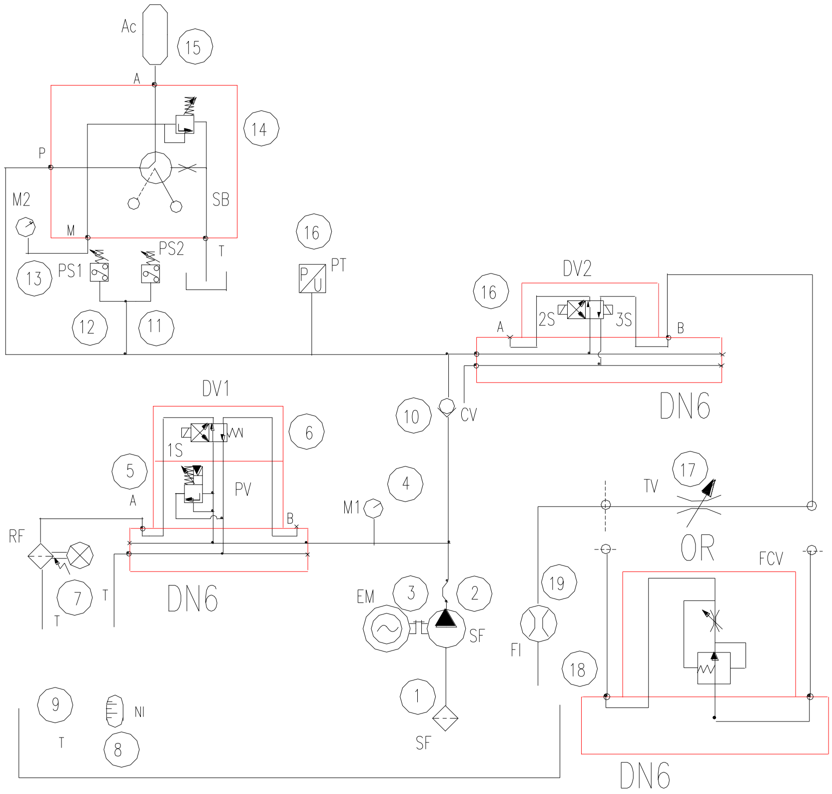


Fig. 11. Hydraulic diagram of the installation

The measured flow rates are average values. They were obtained by measuring the volume spilled over and the associated times.
For comparison, the results obtained by calculation, simulation and experiments are shown in Table 1.

Table 1: Time for charging and discharging the accumulator

	Analytical calculation	**Simulation**	**Measurements**
Charging time [s]	55	60	64
Time for discharging by open throttle valve 15% [s]	11 Flow 13.6 l/min	~9 Flow ranging from 12 to 15 l/min	~7 Average flow ~13 l/min
Time for discharging by flow control valve adjusted at 4 l/min [s]	27	~29	~26

6. Conclusions

The use of hydro-pneumatic accumulators in the hydraulic installations is more and more frequent. Among the advantages of using the hydro-pneumatic accumulators we can mention:
- diminution of the pumps flow;
- storage of hydraulic energy during the phases when the installations have low consumption;
- ensuring uninterrupted operation of the installations for a certain period, even in the event of pump stop.

Accumulators are usually charged in a longer period than the period necessary for discharging. To determine the time of discharge but also the flow rate usable in a certain range of pressures, in this paper were developed mathematical models that can be used in most hydraulic installations of different fields. According to the type of installation, one can use the mathematical model that corresponds to the discharge:
- flow depending on pressure drop;
- flow depending on the root of the pressure drop;
- flow that is not dependent on pressure drop if flow control valves are used.

In order to benefit from the mathematical models presented in the paper, one shall use the features specific to the equipment utilized, in conformity with catalogues of the manufacturers.
Specialized simulation programs can be used successfully provided that the features of the actual equipment are known.
The experimental measurements confirm the accuracy of the mathematical models developed.

References

[1] D. Prodan, M. Duca, A. Bucureşteanu, T. Dobrescu, "Hydrostatic drives-machine parts" („Acţionări hidrostatice - Organologie"), AGIR Publishing House, Bucharest, 2005;
[2] H. Exner, R. Freitag, H. Geis, R. Lang, J. Oppozer, P. Schwab, E. Sumpf, U. Ostendorff, "The Hydraulic Trainer" Volume 1, Mannesmann Rexroth GmbH, Wurzbug, 1991;
[3] T. Deaconescu, "Hydraulic Drives" („Acţionări hidraulice"), Publishing House of Transilvania University, Brasov, 2007;
[4] *** Bosch Rexroth, Berarma, Parker, Atos, Aron, Hydro Leduc, Fox catalogs;
[5] *** Hydro-Pneumatic Accumulators Nakamura Koki Co., LTD;
[6] D. Prodan, A. Bucuresteanu, E. Bălan, "Increasing the Operating Safety of the Machine-Tools by Using Hydro-pneumatic Accumulators. Modeling – Simulation", *Advanced Materials Research* Vols 463-464 Online: 2012-02-10 © (2012) Trans Tech Publications, Switzerland, pp. 1111-1115;
[7] D. Prodan, A. Bucureșteanu, E. Balan, "Heavy machine-tools. feed-positioning systems for cross-rails", *The 21st INTERNATIONAL DAAAM SYMPOSIUM "Intelligent Manufacturing & Automation: Theory, Practice & Education"* 20-23rd OCTOBER 2010, Zadar, Croatia, ISSN 1726-9679, ISBN 978-3-901509-73-5, Katalinic, B. (Ed.), Published by DAAAM International, Vienna, pp. 0273-0274;
[8] D. Prodan, E. Bălan, A. Bucureșteanu, G. Constantin, "Solutions for hydraulic balancing of the positioning and feed kinematic chains in portal milling machines", *Proceedings in Manufacturing Systems*, Volume 9, Issue 1, 2014, ISSN 2067-9238, Bucharest, pp. 25–30;
[9] D. Prodan, "Heavy machine tools. Mechanical and Hydraulic Systems" („Maşini-unelte grele. Sisteme mecanice si hidraulice"), Printech Publishing House, Bucharest, 2010;
[10] D. Prodan, A. Bucuresteanu, E. Bălan, "Safety Hydraulic System for Grinding Machines with Magnetic Table", *HERVEX 2007*, Published by INOE 2000-IHP, ISSN 1454 -8003, Râmnicu Vâlcea, pp. 334-337;
[11] B. Perovic, "Machine tools handbook" („Handbuch Werkzeug-Maschinen"), Carl Hanser Verlag, Munchen, 2006;
[12] P. H. Joshi, "Machine tools handbook", McGraw-Hill Publishing House, New Delhi, 2007;
[13] *** Software package: Automation Studio by Famic Technologies Inc.;
[14] D. Prodan, "Machine-Tools. Modelling and Simulation of Hydrostatic Elements and Systems" („Maşini-unelte. Modelarea şi simularea elementelor şi sistemelor hidrostatice"), Printech Publishing House, Bucharest, 2006.

Modeling of Flow through Cooling Plant with Sea Water

Prof. PhD.Eng. **Mariana PANAITESCU**[1], Prof. PhD.Eng. **Fanel-Viorel PANAITESCU**[2],
Master Degree Student **Gabriel TUDOSE**[3]

[1] Constanta Maritime University, marianapan@yahoo.com

[2] Constanta Maritime University, viopanaitescu@yahoo.ro

[3] Constanta Maritime University, tudosegabriel74@gmail.com

Abstract: *This paper aims at modeling of fluid flow through the installation of sea-water cooling system of an engine. The overall goal is to optimize the flow by modifying driving pipes and decreasing hydraulic losses. It has created a three-dimensional model to reproduce as accurately as the real facility using ANSYS FLUENT 13.0. There were calculated the variation of pressure and flow inside the plant. Flow model chosen is k-ε model. The results are presented graphically. In order to optimize the flow were analyzed the hydraulic losses both analytical and simulation in ANSYS FLUENT 13.0.The differences obtained ranged between 9...15%. For a result as close as possible to reality, the losses of pressure and flow rate variation are measured on a pipe segment as bounded. In practice, this study will help to correct the operation of the plant and its aggregates.*

Keywords: *Modeling, flow, cooling plant, sea, water, loss, pressure, simulation.*

1. Introduction

Cooling plant with sea water, contains a whole series of elements, by their construction, the material used and the types of pumps, leading to reliable operation of all equipments in the machine compartment, interdependent towards some.
Cooling facility is provided so many filtering systems back-up: main pumps, an emergency pump, coolers, valves "one-way", valves type "Butterfly", pressure and temperature transmitters (Fig. 1)[1, 2].

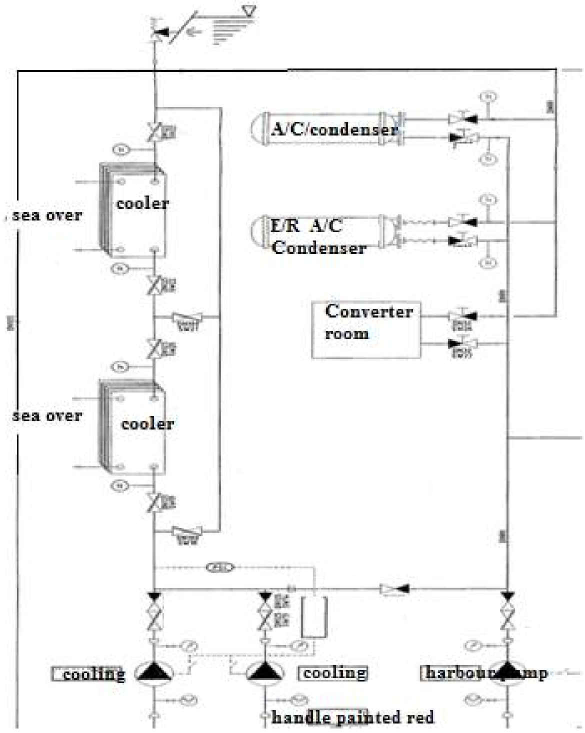


Fig. 1. Cooling plant scheme [2]

Pumps used are: water pump (Q = 125 m^3/h), fire and bilge pump, bilge pump and ancillary installations (Q = 5 m^3/h), foam pump (Q = 210 m^3/h), the ejector pump (Q = 36 m^3/h) and general service pump (Q = 60 m^3/h).

2. Method and research

For the modeling of fluid flow in the cooling plant ANSYS FLUENT 13.0 has been used [3,4].

2.1 Choice of model and mesh scope study

Build a three-dimensional model to reproduce as accurately as possible the real facility. The principle of construction itself was to merge some geometric shapes (square, sphere, cylinder, truncated cone) (Fig. 2) the initial diameter of the pipes is 100 mm, it is likely to drop to 60 mm by means of reductions [5,6]. After we have chosen the desired geometric form, we can define the whole beach of dimensions.

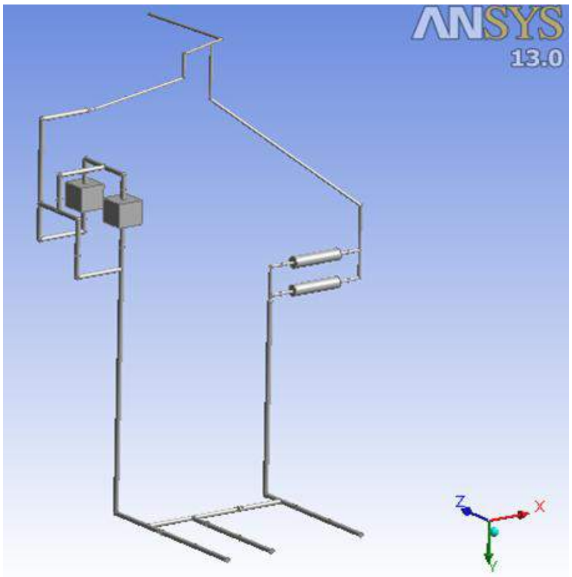


Fig. 2. Sea water cooling plant scheme in 3D [4, 5]

2.2 Mesh fields of study

There are meshing areas of plant study: pipes, valves, cooling (Fig. 3, Fig. 4).

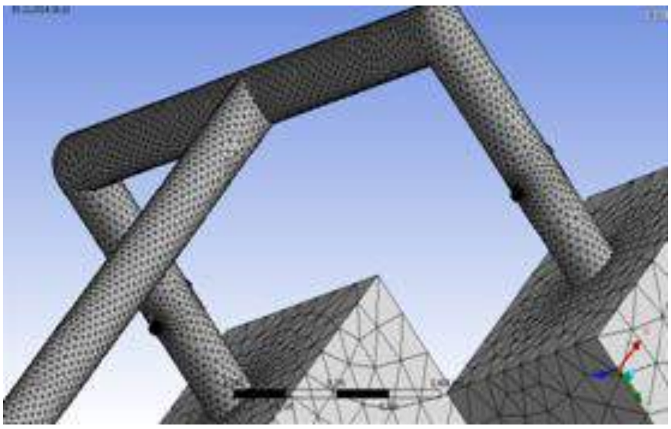


Fig. 3. Mesh of drive pipes

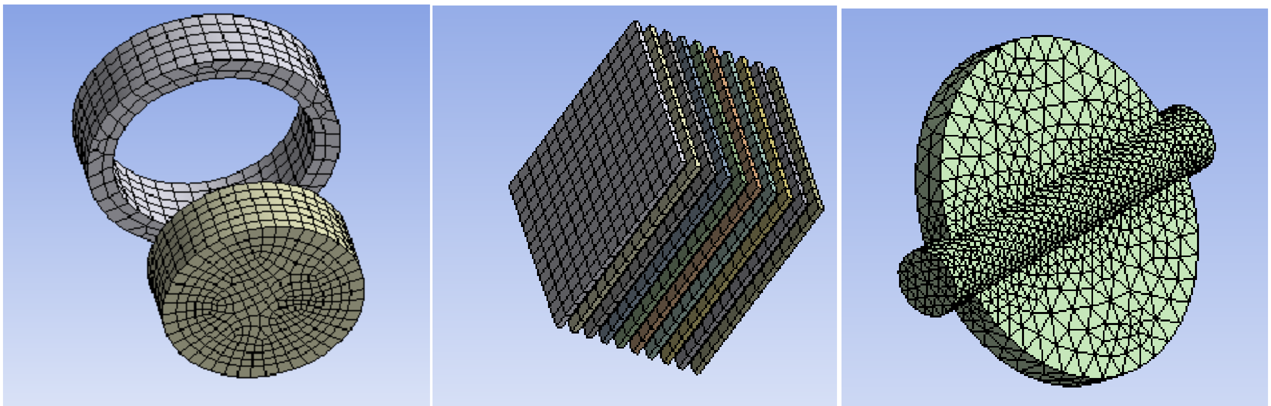


Fig. 4. Mesh of one-way valve, cooler and “butterfly” valve

2.3 Setting water flow parameters

Select the following variables: the working fluid (seawater), density (1015 kg/m^3), the acceleration due to gravity (9.81 m^2 /s), kinematics viscosity constant (0.001003 kg/m.s), mass flow (32.24 kg/s, respectively 22.5 kg/s, absolute speed, flow model k- ε,, transitional fluid, fluid discharge in atmosphere.

3. Results and interpretations

Further analyze the graphical results for chosen facility. In the following figures (Fig. 5, Fig. 6, and Fig. 7) can be seen falling pressure and speed changes that take place as the fluid flows through the pipes. Converges solutions aimed at 537 iterations.

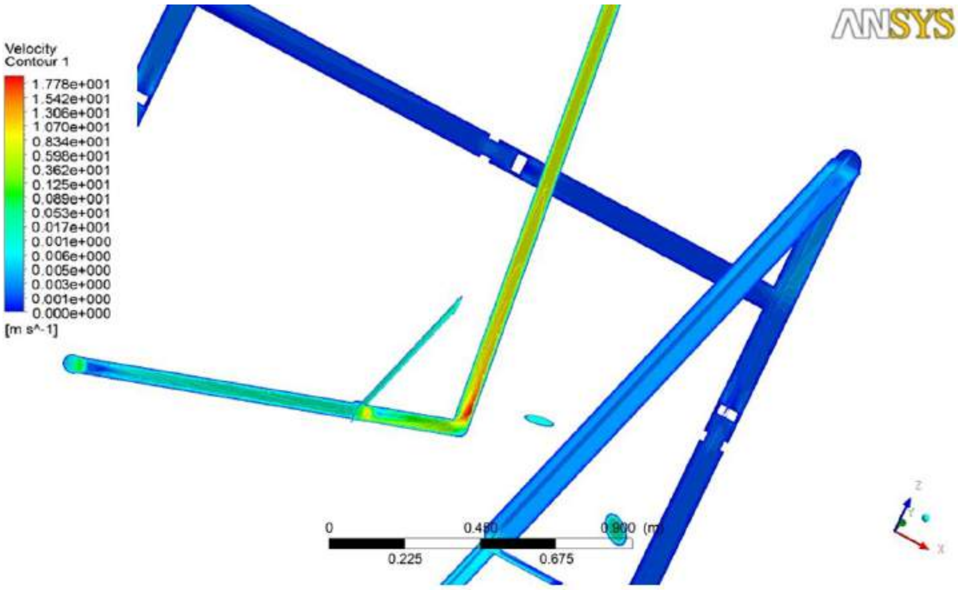


Fig. 5. Maximal speed through the pipes of plant

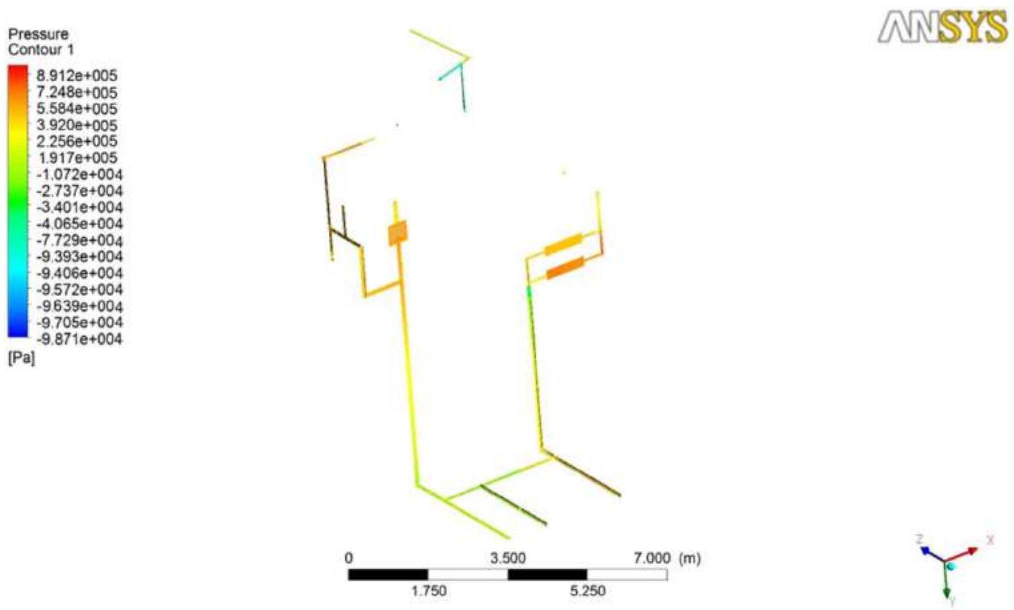


Fig. 6. Falling pressure through the pipes of plant

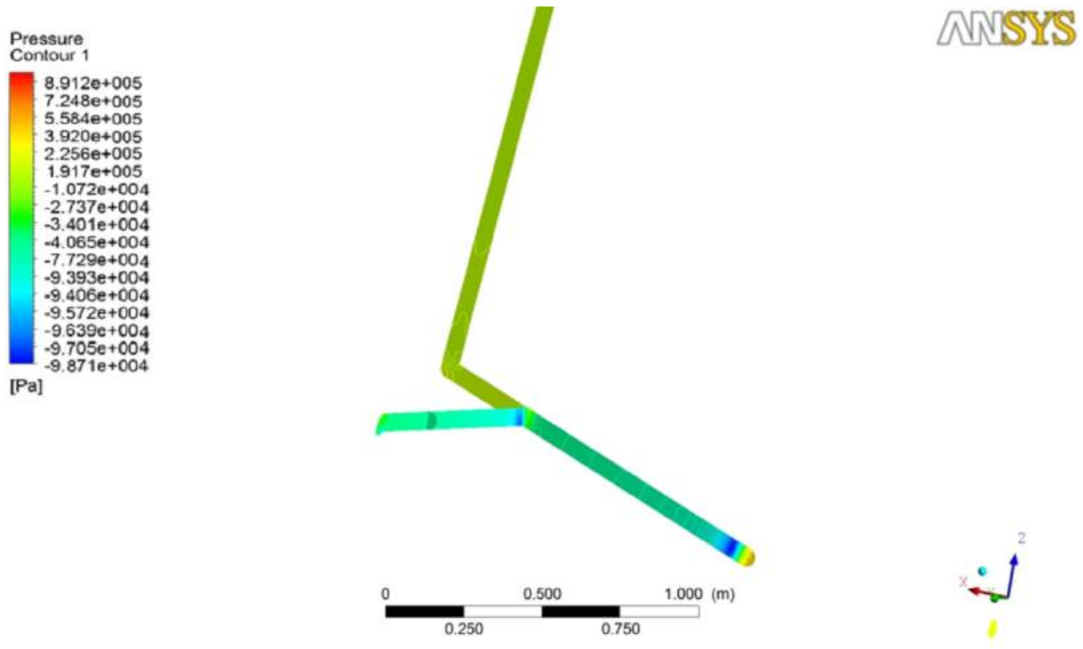


Fig. 7. The partial driving pipes with maximal falling pressure

In order to optimize the flow were analyzed the plant's losses, both analytical and hydraulic simulation in ANSYS FLUENT 13.0 (Table 1).
There have been compared hydraulic losses on several sections of pipes (1, 2, 3), on elbows and valves. After this, the total losses have been calculated and compared.

Table 1: Comparative study for hydraulic losses

Hydraulic loss	Calculated Value [Pa]	ANSYS FLUENT Value [Pa]
Loss section 1	$0.44*10^5$	$0.65*10^5$
Loss section 2	$0.89*10^5$	$0.93*10^5$
Loss section 3	$0.14*10^4$	$0.12*10^4$
Total linear losses	$1.34*10^5$	$1.59*10^5$
Elbow loss on 90^0	$1.66*10^5$	$1.54*10^5$
Loss on "butterfly" valves	$1.83*10^5$	$1.62*10^5$
Loss on "one-way" valves	$0.70*10^5$	$0.93*10^5$
Total local losses	$8.02*10^5$	$8.91*10^5$

The values obtained using ANSYS FLUENT 13.0 are not identical, but are plausible. For precision of study, pressure losses or cha nges in flow rate shall be measured on a limited sections of pipes.

4. Conclusions

For analysis of graphics were inserted various sections of the plane built, observing thus eddy currents. ANSYS FLUENT values were taken with the option "variable sampling" in the points of interest.
The main purpose of this work was to carry out the construction of a segment of the plant as close to the real.
It has also pursued the fluid flow around fittings, elbows's T, taking into account linear and local losses of pressure.
The margin of error was received between 8-15%, about what it means that the estimates were done correctly.

Acknowledgments

Acknowledgments of the crew and management team from LS Jamie Oil Tanker.

References

[1] A. Costică, Machines and plants for naval propulsion" ("Mașini și instalații navale de propulsie"), Technical Publishing House, Bucharest, 1991;
[2] http://www.marinetraffic.com/pl/ais/details/ships/236507000;
[3] http://148.204.81.206/Ansys/150/ANSYS%20Fluent%20Tutorial%20Guide.pdf;
[4] http://www.ansys.com/Products/Fluids/ANSYS-Fluent;
[5] D. Dinu, "Fluids Mechanics for seafarers" ("Mecanica Fluidelor pentru navigatori"), NAUTICA Publishing House, Constanta, 2010;
[6] M. Panaitescu, F.V. Panaitescu, „Fluids Mechanics and Hydraulic machines for Environmental Engineering" („Mecanica Fluidelor si Masin Hidraulice pentru ingineria mediului"), volume 1, 2, NAUTICA Publishing House, Constanta, 2016.

Discreet Flow Distribution in Central Pivot

Prof. Dr. **Podalyro Amaral de SOUZA**[1], Dr. **Eber Lopes de MORAES**[2]

[1] Departamento de Engenharia Hidráulica e Sanitária / Escola Politécnica / USP
Av. Prof. Lúcio Martins Rodrigues, 120 – CEP 05508-9000 São Paulo, SP
podalyro@usp.br-(Prof. Titular.)

[2] Departamento de Engenharia Hidráulica e Sanitária / Escola Politécnica / USP
Av. Prof. Lúcio Martins Rodrigues, 120 – CEP 05508-9000 São Paulo, SP
eberdemoraes@hotmail.com-(Post-Doctoral.)

Abstract: *In the search of a Central Pivot's best operational performance, the equipment's electric and mechanical characteristics, plus the realistic knowledge of the hydraulic relationships to which the "system" as a whole is submitted, is a must for its proper sizing in function of its components adequate choice. The presence of different sprinklers, featuring diverse diameters and hydraulic characteristics, pressure regulators and the possibility of the presence of a larger sprinkler with a booster on the network, allied to the necessity of attending the physical-natural restrictions which wrap-up agriculture production technology, properly express the arduous task of developing irrigation and drainage projects seeking a harmonic arrangement between a uniform application and the efficient use of water catchment's reserves.*
Leaning towards the discussion of this subject and seeing the importance of the presence of automatic irrigation systems on a large scale agricultural productivity, the present paper presents its basic considerations under the mathematical modeling scope of the sprinkler's flow distribution along the Central Pivot's line, so as to harmonize both their continuous and discrete flows, aiming at a better aspersed water uniformity application.

Keywords: *Discrete distribution, discrete flow.*

1. Introduction

The Central Pivot is a mechanized aspersion system, idealized by Fank Zybach (EUA) in the '70s, whose main fascination lies in it being an irrigation equipment endowed with self-movement.
In the late '70s, at the height of the "green revolution" proposed by the economic strength of the Brazilian military regime, the merging of the equipment's operational versatility with the possibility of service to larger areas, sealed for the first time the interest of national agriculture in 'automated" and "dynamic" irrigation, a feature extremely attractive in a country with such continental dimensions and tropical climate as Brazil. [1]
Effectively, this is an equipment made up of "an irrigational radial line bearing a fix extremity, from where it is supplied with water and electric energy and around which it describes a spinning circle, having as a main feature, the fact of moving itself while irrigating [1].
On the other hand, although bearing sophisticated performance characteristics, once hydraulically poorly dimensioned, those devices mostly present a high water and electricity consumption, resulting in a sharp increase in agricultural production costs, enabling the occurrence of runoff and consequent fertile soil erosion, affecting definitely the cultivated area by the indiscriminate use of high intensity applications for long periods, which usually occurs due to technical conception failures. [3]
This framework, however, is a practice no longer allowed nowadays, where the shortage of water in the original watersheds, justified by the necessity of raising up new areas of cultivation and the ongoing climate irregularity, caused by years of reprehensible environmental practices is felt in a Brazil that awakens to a new consciousness based on three points of interest: the competitive use of water, preservation of natural resources and concern about the immediate need for sustainable development of irrigated agriculture. [1]

2. Applying intensity x continuous distribution

In pipeline outflows with multiple exits, known as those with distribution in motion, inherent to the Central Pivot irrigation equipment, the flow that goes through a stretch of the pivot generic line is, in a first moment, treated mathematically as a variable which varies continuously along the outflow pressure. [8]

This hypothesis is assumed in this first step because of the interest on knowing the relationship between required flow and wetted circular area with different lengths of line, "R" and "r"respectively. Thus, the intensity of sprinkled "rain" "I" in a given area, Figure 1, is described analytically for the two radius lengths as:

$$\frac{Q_0}{\pi R^2} = \frac{Q}{\pi(R^2 - r^2)} = I \tag{1}$$

or rewritten evidencing the residual flow in the "$R-r$" length stretch.

$$Q = Q_0\left[1 - \left(\frac{r}{R}\right)^2\right] \tag{2}$$

being:

- Q - the flow that meets the downstream stretch of the corresponding length "$R-r$";
- Q_0 - the entrance flow in the equipment (required by the system);
- R - the radius effectively moistened;
- r - the generic radius.

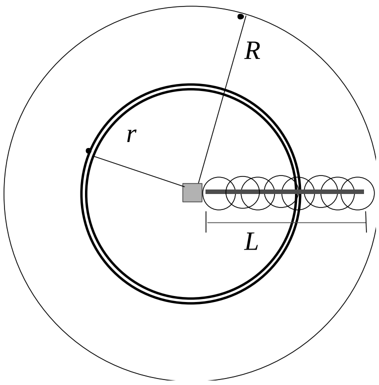


Fig. 1. Area irrigated by a generic Pivot

$$Q_0 \left(\frac{r^2}{R^2}\right) = Q_0 - Q \tag{3}$$

The principle of mass conservation, applied to flows with multiple derivations, Figure 2, ensures that the sum of the sprinklers flow to a position "r" of the line, should meet the relation:

$$\sum q_i = Q_0 - Q \tag{4}$$

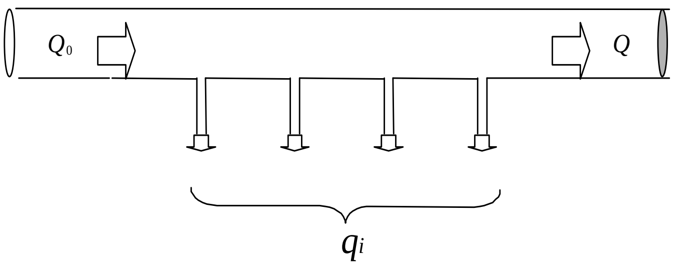


Fig. 2. A Central Pivot generic stretch

3. Continuous distribution x discreet distribution

The Equation 2, which models the flow that will sprinkle the final stretch of the downstream cultivation area, corresponding to the circular crown's difference radius "$R-r$" carries in its wake the hypothesis of continuous water distribution along the line. In practice this is not the case, being effectively a discrete distribution of water through equally spaced nozzles. [2]

This difference between the continuous theoretical model, Eq 2, and the effective functioning of the discrete distribution of water in the actual performance of a Central Pivot must be properly bypassed, at the risk do not following the principle of mass conservation, which bears on errors in the necessary numerical calculation results for a Central Pivot careful design. [6]

The compatibility between these ways of conceiving the distribution can be obtained by accepting that the discrete flow "q_i" released by a sprinkler located in a given generic position "i" must water the ring area bounded by the sprinklers located between positions "i" and "$i+1$", distant from "Δx".This reasoning requires that the flow released by the last sprinkler (downstream) will water, in addition, a circular crown radius "Δx", this crown, located beyond the effective length "L" of the Central Pivot's line.

Considering the "Q_i" flow in the position "i" of the Pivot line, flow which should meet all the sprinklers demand of that "i" position until the downstream's end plus any further demand from a large sprinkler (with booster) corresponding to a "Q_c", flow designed to water the Crown between "L and R", where "L"is the effective pipe length and "R" is the radius of the total watered area, "$R-r$", one can therefore rewrite the water distribution model as:

$$Q_i - Q_c = I\ \pi \left[(L+\Delta r)^2\right] \tag{5}$$

$$Q_i - Q_c = I\ \pi \left[(L+\Delta r)^2 - r_i^2\right] \tag{6}$$

This expression Eq. 6, when written for the first line sprinkler, "x_1", takes the form:

$$Q_1 - Q_c = I\ \pi \left[(L+\Delta r)^2 - r_1^2\right] \tag{7}$$

With Eqs 6 & 7, formulated to express the moving flow (discreet outputs) along the line of a pivot, one can write the application intensity bearing in mind the hypothesis of continuous outputs as:

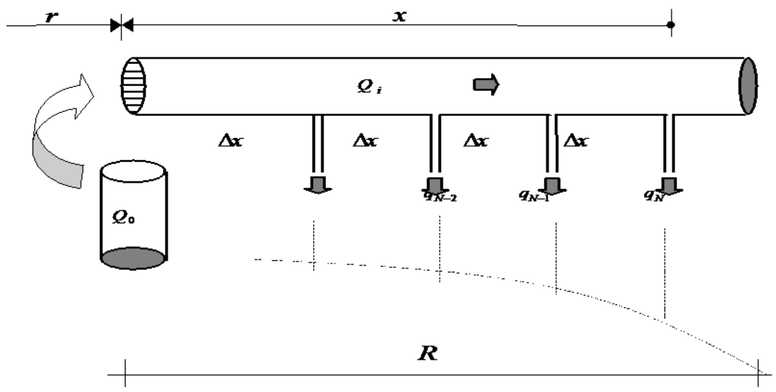


Fig. 3. A generic Central Pivot stretch (identification of flows in motion)

$$\frac{Q_i - Q_c}{Q_1 - Q_c} = \frac{(L+\Delta r)^2 - r_i^2}{(L+\Delta r)^2 - r_1^2} \tag{8}$$

whose line flow, made explicit for a generic sprinkler "i",can be written as:

$$Q_i - Q_c = \left[\frac{(L+\Delta r)^2 - r_i^2}{(L+\Delta r)^2 - r_1^2}\right](Q_1 - Q_c) \tag{9}$$

and to the next downstream sprinkler "$i+1$", as:

$$Q_{i+1} - Q_c = \left[\frac{(L+\Delta r)^2 - r_{i+1}^2}{(L+\Delta r)^2 - r_1^2} \right] (Q_1 - Q_c) \tag{10}$$

Thus, identifying the flows present in a pivot's main line, at the borders of a generic sprinkler "i", the principle of mass conservation ensures that this sprinkler's flow is expressed by $q_i = Q_i - Q_{i+1}$, Figure 5, where the elimination of "Q_i" and "Q_{i+1}" occurs with the interaction of Eqs 11 and 12 respectively. This facilitates the obtaining of a "q_i" flow rate law, expressed by:

$$q_i = \left[\frac{r_{i+1}^2 - r_i^2}{(L+\Delta r)^2 - r_1^2} \right] (Q_1 - Q_c) \tag{11}$$

By expanding this expression's numerator, Eq 13, one obtains:

$$r_{i+1}^2 - r_i^2 = (r_i + \Delta r)^2 - r_i^2 = 2\, r_i^2 \Delta r + \Delta r^2 \tag{12}$$

With an expanded numerator, we can write Eq 16 as:

$$q_i = \left[\frac{2 x_i \Delta x + \Delta x^2}{(L+\Delta x)^2 - x_1^2} \right] (Q_1 - Q_c) \tag{13}$$

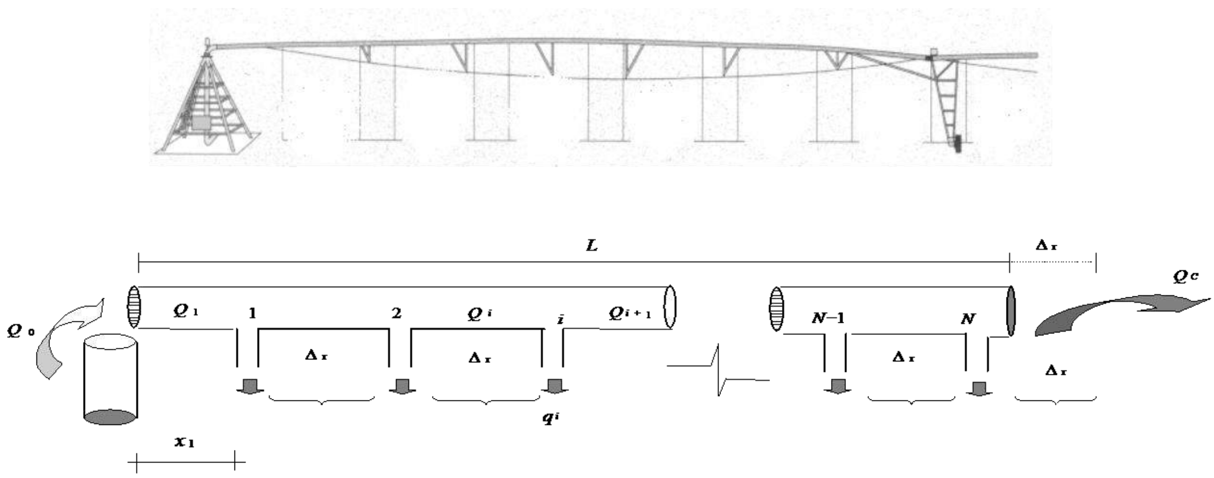


Fig. 4. A generic stretch of a Central Pivot (continuous treatment of withdrawals under way)

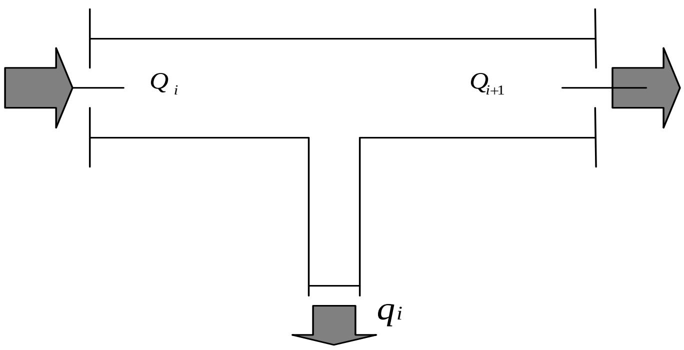


Fig. 5. Schematic representation of a generic sprinkler "i"

Thus, working out the algebraic relations between "L" and "r_i" it gives:

$$L = r_i + (N-i)\,\Delta r \tag{14}$$

$$r_i = (L - (N-i)\,\Delta r) \tag{15}$$

Where the expression for the "q_i" flow can be written as:

$$q_i = \left[\frac{2(L-(N-i)\,\Delta r)\,\Delta r + \Delta r^2}{(L+\Delta r)^2 - r_1^2}\right](Q_1 - Q_c) \tag{16}$$

or still further as:

$$q_i = \left[\frac{2L\Delta r - (2(N-i)-1)\,\Delta r^2}{(L+\Delta r)^2 - r_1^2}\right](Q_1 - Q_c) \tag{17}$$

4. Conclusions

In possession of the relationships shown in Eqs 14 and 15, and developing the denominator of Equation 17 in the attempt of obtaining an expression that is involved only with the magnitudes "L", "N" and "Δr",and it is possible to eliminate the term "r_1^2" in this part of the equation, writing:

$$(L+\Delta r)^2 - r_1^2 = (L^2 + 2L\Delta r + \Delta r^2) - r_1^2 \tag{18}$$

$$= (L^2 - r_1^2) + 2L\Delta r + \Delta r^2$$

$$= (2L(N-1)\Delta r - (N-1)^2\Delta r^2) + 2L\Delta r + \Delta r^2$$

$$= 2\;L\;N\;\Delta r - 2\;L\;\Delta r - N^2\Delta r^2 + 2\;N\;\Delta r^2 - \Delta r^2 + 2\;L\;\Delta r + \Delta r^2$$

$$= 2\;L\;N\;\Delta r \;- N^2\Delta r^2 + 2\;N\;\Delta r^2 \tag{19}$$

Finally, rewriting Eq. 17 with the denominator expression presented by Eq. 19, we have modeled the final equation for calculating the flow rate of a generic sprinkler, "i", in a pivot's line, described as:

$$\boxed{q_i = \left[\frac{2\;L\;\Delta r - 2\,(N-i)\,\Delta r^2 + \Delta r^2}{2\;L\;N\;\Delta r - N^2\Delta r^2 + 2\;N\;\Delta r^2}\right](Q_1 - Q_c)} \tag{20}$$

References

[1] R.D. Coelho**,** "Caracterização do sistema pivô central (LEPA), operando em condição de microrelevo condicionado"**,** 1996, 181p., Tese (Doutorado) – Escola de Engenharia de são Carlos, Universidade de São Paulo, São Carlos;
[2] L.C. James, "Principles of farm irrigation system design", Washington: Krieger Publishing Co, 1988, 543p.;
[3] J. Keller, R. Bliesner, "Sprinkle and trickle irrigation systems", New York: Blackburn Press, 1980, 652p;
[4] A. Lencastre, "Hidráulica geral", Lisboa, Edgard Blücher Ltda., 1983, 411p.;
[5] J.M.T. Martin-Beníto, "El riego por aspersion y su technolojia", Madrid, 406p., 1999;
[6] E.J. Scaloppi, R.G. Allen, "Hydraulics of center pivot laterals", Journal of Irrigation and Drainage Engineering, v.119, n. 3, p. 554-567, May 1993;
[7] E.J. Scaloppi, R.G. Allen, "Hydraulics of irrigation laterals: Comparative analysis", Journal of Irrigation and Drainage Engineering, v.119, n. 1, p. 91-115, Jan. 1993;
[8] E.J. Scaloppi, "Sistemas de irrigação e seus componentes", Brasília, 1987, 67p.

Some Analyze Considerations of the Kerf Variation for Abrasive Water Jet Cutting of a Steel Material

PhD. Stud. Eng. **Paul A. BASARMAN**[1]**,** PhD. Stud. Eng. **Bogdan CIORUŢA**[1]

[1]Technical University of Cluj-Napoca – North University Center at Baia Mare, Faculty of Engineering, Victor Babeş str., No. 62A, 430083, Baia Mare, Romania
adrian.basarman@cunbm.utcluj.ro / bciorutza@yahoo.com

Abstract: *A modern method of cutting materials is considered to be the abrasive water jet (AWJ) cutting method. It is a method that is in full ascension nowadays regarding, especially, the level of usability in production. In order to have the adequate setup for abrasive water jet cutting on a certain system, the surface quality, the kerf aspect, the shape and respectively the form of the obtained part have to be carefully analysed. This paper presents the results obtained after cutting a square shaped part, made of S355 material. In the present study, there were analysed both the inside and the outside of the cut, the kerf width, the aspect of the taper and the profile deviation, in accordance with a computer numerical control (CNC) attached to the cutting machine used.*

Keywords: *abrasive water jet (AWJ), surface quality, 3D measuring of surface, kerf variation.*

1. Introduction

The abrasive water jet (AWJ) cutting method is as mentioned in the abstract a modern method for cutting different types of materials. There is no relevant literature that specifies a formula for calculation of the stock left for machining of metallic parts. In this case, the first thing that had to be studied was the surface aspect of the steel parts after cutting and measuring the kerf size and the profile deviations, both operations being carried out in relation to and as adaptation to other similar research. In the process of cutting the materials using abrasive water jet (AWJ), the surface resulted can be very rough or very fine, the main difference between them would be the combination of the next variables: *thickness of the steel part*, *hardness of the material*, *water pressure used for cutting*, *type and quality of the abrasive particles* and *the cutting speed* [8].
By reference to the scientific literature consulted, Valíček et al. [7] divided the resulted surface, after cutting it with abrasive water jet (AWJ), based upon *the surface roughness*, in three different sectors: *primary impact zone, smooth zone* and *rough zone*. Regarding the way of calculating the cutting feed rate, other researchers interested in the same issues proposed a mathematical modell that had a big number of parameters as independent variables [4]. Researches in this field have been made by authors like Fowler [1], Popan et al. [5], for the purpose of milling the surface of metal parts using this technology. Hlaváček et al. [2], Hloch et al. [3] and Valíček et al. [6] investigated the way of turning materials using abrasive water jet (AWJ) cutting method.

2. Work method and experimental setup

The experiments were conducted using an abrasive water jet (AWJ) cutting machine - *Bystronic ByJet Pro L* - presented in Figure 1a. The advantage in using this kind of method is that the cutting element is the abrasive water jet (AWJ) - distinct in relation to the conventional cutting methods used in production, where there is a contact of the solid tool with parts that need to be cut. Another important advantage to be mentioned of this method is that during the cutting process, the materiall is cooled down just by the water, acting like a coolant liquid.
The abrasive water jet (AW) cutting machine has several components which are presented, in detailed, in Figure 1b.
The main components of the abrasive water jet (AWJ) machine are the followings: 1) catch tank; 2) workpiece; 3) cutting head; 4) abrasive delivery system; 5) work motions system; 6) grill for supporting the parts. This abrasive water jet (AWJ) cutting machine is a computer controlled machine (CCM), assisted easily only by an operator.

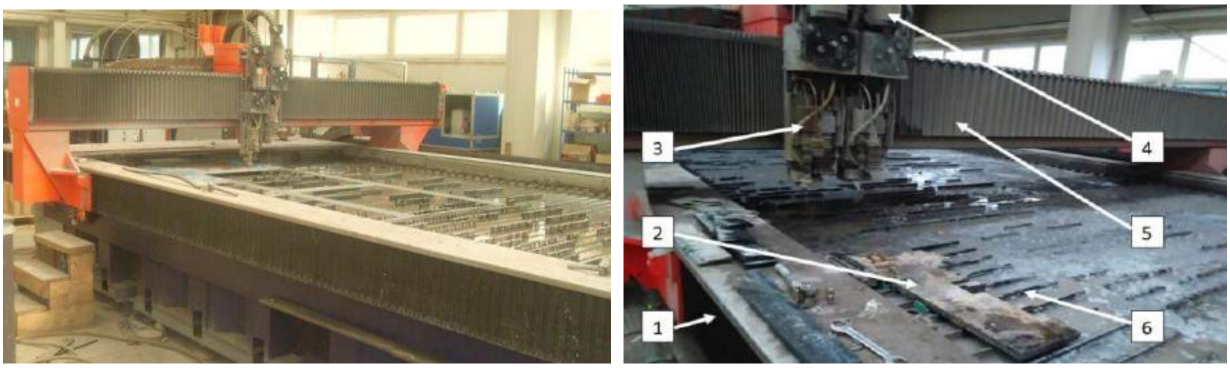


Fig. 1. a) The Bystronic ByJet Pro L machine; **b)** Elements of the considered abrasive water jet machine

In order to take notice of how the surface has modified after cutting, we had to analyse, in parallel, both the interior and the exterior of the kerf, as presented in Figure 2a and also, in that way, by measuring the kerf width, we can experimentally determine the stock left for machining, needed in order to have a correct and more accurate cut.

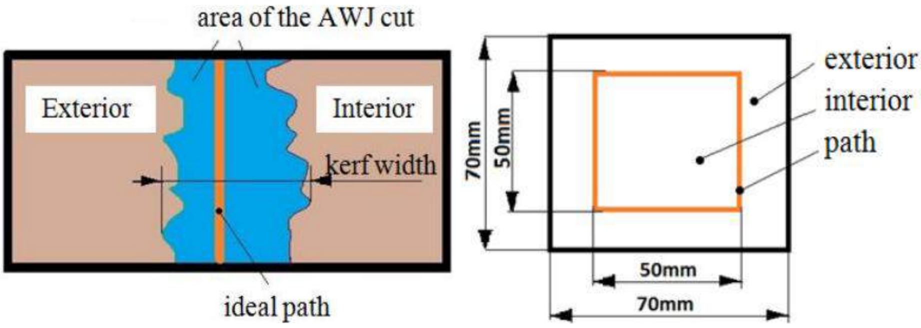


Fig. 2. a) General aspect of the abrasive water jet cut; **b)** Design for cutting

These experiments were conducted on a S355 material plate, of 30 mm thickness. The design on how the cut was made is presented above, in Figure 2b. From the list of cutting conditions offered by the machine manufacturer, there were selected the slowest, that can possibly offer the best results regarding the quality of the cut. The selected parameters are presented in Table 1.

Table 1: The parameters used for the cutting regime

Parameters	Value selected
Breakthrough time	19 s
Breakthrough pressure	3600 bar
Abrasive material used	GMA Garnet 80 Mesh (150-300 micron)
Quantity of abrasive material	342 g/min
Cutting pressure	3600 bar
Cutting speed	27 mm/min
Interior diameter of sapphire nozzle	0.28 mm
Exterior diameter of nozzle	0.8 mm

We considered that the first step in conducting the experiments is putting the steel part (S355) on the machine's grill support (6 – from Figure1b) and fixing it down, using the machine clamping system. After this step, the program of cut is loaded in the software of the machine, the next step including the selection of the cutting conditions. Subsequently, the part is cut.
The part that has been cut is presented right after cutting, in Figure 3a, still on the machine, and after taking out of the base plate, in Figure 3b.

Fig. 3. a) Part after cutting still on the machine; **b)** Same part after taking out of the base plate

After cutting the steel part, for both parts resulted, as in Figure 3b, in order to determine the exact aspects all the surfaces, we focus on splitting the surface in three rows. Further, using a 3D measuring arm, as presented in Figure 4a, there were taken a number of 10 measurements on each row, having a total of 30 points measured on a surface, as presented in the Figure 4b.

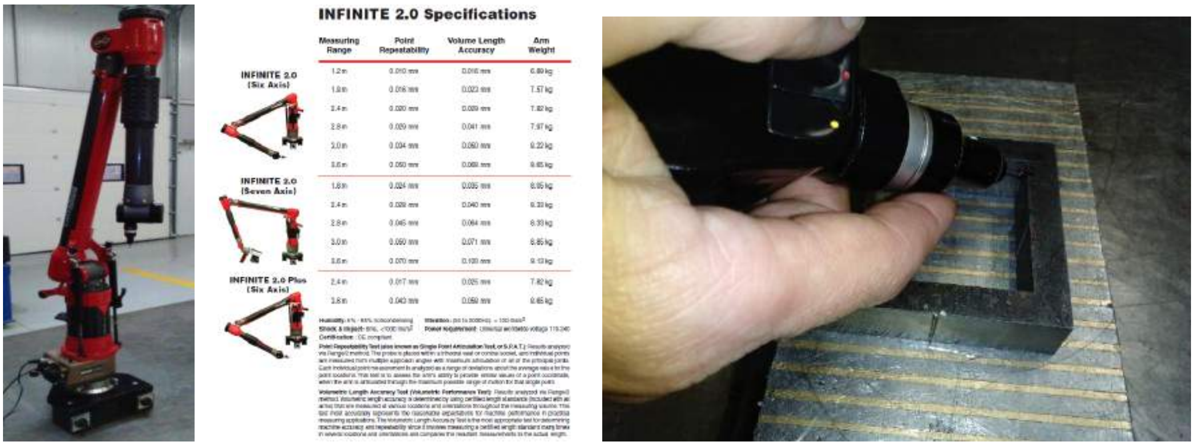


Fig. 4. a) Cimcore Infinite 3D measuring arm, v.2.0; **b)** The surfaces analysed after cutting operation

The rows were measured separated and analysed one with each other, both for the interior and for the exterior of the cut. Being a square part, that meant having four exterior surfaces (E1-4) and four interior surfaces (I1-4), they were analysed together (e.g. E1 with I1), as in the Figure 5a. As previously specified, for every surface measured independently we obtain a set of values, as in Figure 5b.

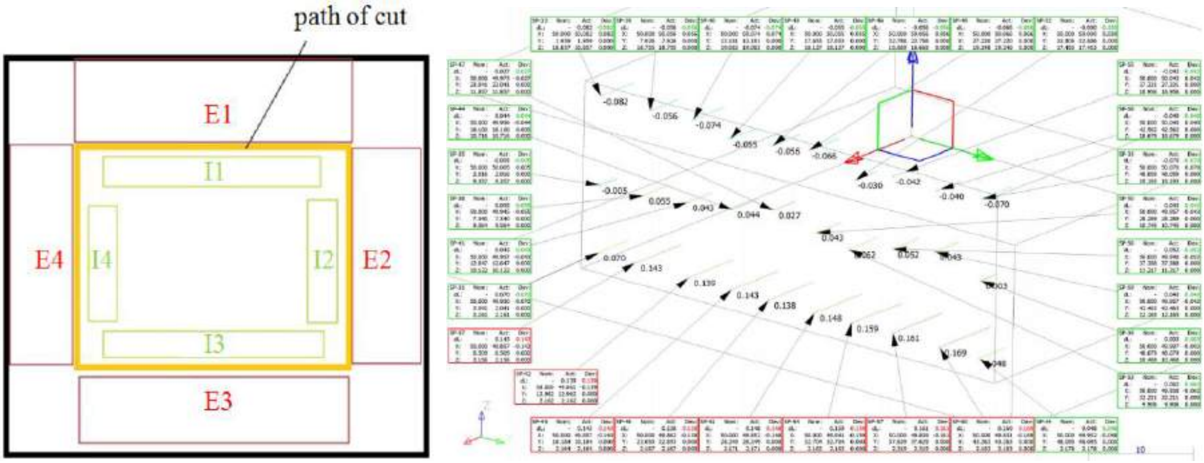


Fig. 5. a) The way of measuring the points; **b)** The measuring set values obtained for the cutting part

After measuring every surface, the values were included in Table 2.

Table 2: Values measured on every surface on the steel part cut using abrasive water jet (AWJ) method

Surface I	Interior	1	2	3	4	5	6	7	8	9	10
	Ideal surface	0	0	0	0	0	0	0	0	0	0
	up	0.910	0.879	0.872	0.870	0.923	0.922	0.900	0.910	0.900	0.920
	middle	0.787	0.746	0.747	0.750	0.854	0.812	0.765	0.764	0.759	0.789
	down	0.632	0.576	0.563	0.603	0.688	0.645	0.587	0.566	0.608	0.644
	Exterior	1	2	3	4	5	6	7	8	9	10
	up	-0.097	-0.083	-0.080	-0.084	-0.120	-0.069	-0.058	-0.056	-0.058	-0.078
	middle	-0.286	-0.217	-0.239	-0.213	-0.398	-0.292	-0.204	-0.177	-0.182	-0.301
	down	-0.399	-0.325	-0.334	-0.358	-0.524	-0.450	-0.310	-0.297	-0.328	-0.486
Surface II	Interior	1	2	3	4	5	6	7	8	9	10
	Ideal surface	0	0	0	0	0	0	0	0	0	0
	up	0.939	0.909	0.907	0.894	0.903	0.904	0.908	0.909	0.906	0.937
	middle	1.161	1.072	1.075	1.076	1.074	1.072	1.078	1.067	1.079	1.161
	down	1.264	1.225	1.190	1.199	1.199	1.174	1.191	1.196	1.214	1.268
	Exterior	1	2	3	4	5	6	7	8	9	10
	up	-0.067	-0.050	-0.061	-0.058	-0.059	-0.054	-0.057	-0.048	-0.058	-0.071
	middle	0.022	0.110	0.100	0.115	0.110	0.096	0.142	0.114	0.099	0.035
	down	0.165	0.285	0.295	0.299	0.290	0.288	0.289	0.282	0.259	0.213
Surface III	Interior	1	2	3	4	5	6	7	8	9	10
	Ideal surface	0	0	0	0	0	0	0	0	0	0
	up	0.856	0.844	0.850	0.860	0.881	0.882	0.872	0.889	0.889	0.924
	middle	0.799	0.739	0.718	0.724	0.730	0.749	0.759	0.766	0.763	0.829
	down	0.685	0.565	0.546	0.583	0.553	0.570	0.568	0.597	0.590	0.706
	Exterior	1	2	3	4	5	6	7	8	9	10
	up	-0.136	-0.100	-0.099	-0.095	-0.089	-0.089	-0.091	-0.092	-0.101	-0.122
	middle	-0.300	-0.190	-0.192	-0.202	-0.208	-0.205	-0.197	-0.202	-0.193	-0.261
	down	-0.443	-0.326	-0.332	-0.329	-0.309	-0.329	-0.314	-0.328	-0.327	-0.379
Surface IV	Interior	1	2	3	4	5	6	7	8	9	10
	Ideal surface	0	0	0	0	0	0	0	0	0	0
	up	0.965	0.936	0.930	0.914	0.906	0.897	0.899	0.887	0.880	0.885
	middle	1.139	1.051	1.036	1.015	1.006	0.979	0.998	0.976	0.989	1.062
	down	1.211	1.105	1.098	1.072	1.098	1.075	1.054	1.055	1.056	1.140
	Exterior	1	2	3	4	5	6	7	8	9	10
	up	-0.030	-0.026	-0.021	-0.028	-0.036	-0.048	-0.038	-0.058	-0.060	-0.099
	middle	0.007	0.056	0.078	0.051	0.052	0.035	0.046	0.027	0.019	-0.057
	down	0.127	0.193	0.181	0.147	0.174	0.158	0.139	0.143	0.149	-0.007

3. Analysing the results

After measuring every surface, as mentioned before, the values were presented both referring to the ideal path of cut and on each side of the path. In some cases, as presented in Table 2, some of the resulted values were considered as negative. These being stated, regarding the kerf width, the values from each side of the ideal path were summed together, thereby hoping to form the resulted measurement of the kerf width, as presented in the Table 3.

Table 3: The value of the kerf width on every surface

Surface I	No. of meas.	1	2	3	4	5	6	7	8	9	10
	up	1.007	0.962	0.952	0.954	1.043	0.991	0.958	0.966	0.958	0.998
	middle	1.073	0.963	0.986	0.963	1.252	1.104	0.969	0.941	0.941	1.090
	down	1.031	0.901	0.897	0.961	1.212	1.095	0.897	0.863	0.936	1.130
Surface II	No. of meas.	1	2	3	4	5	6	7	8	9	10
	up	1.006	0.959	0.968	0.952	0.962	0.958	0.965	0.957	0.964	1.008
	middle	1.139	0.962	0.975	0.961	0.964	0.976	0.936	0.953	0.980	1.126
	down	1.099	0.940	0.895	0.900	0.909	0.886	0.902	0.914	0.955	1.055
Surface III	No. of meas.	1	2	3	4	5	6	7	8	9	10
	up	0.992	0.944	0.949	0.955	0.970	0.971	0.963	0.981	0.990	1.046
	middle	1.099	0.929	0.910	0.926	0.938	0.954	0.956	0.968	0.956	1.090
	down	1.128	0.891	0.878	0.912	0.862	0.899	0.882	0.925	0.917	1.085
Surface IV	No. of meas.	1	2	3	4	5	6	7	8	9	10
	up	0.995	0.962	0.951	0.942	0.942	0.945	0.937	0.945	0.940	0.984
	middle	1.132	0.995	0.958	0.964	0.954	0.944	0.952	0.949	0.970	1.119
	down	1.084	0.912	0.917	0.925	0.924	0.917	0.915	0.912	0.907	1.147

4. Analysing the aspect of surface

The values presented in Table 3 are correspondent directly proportional with the kerf width. We can observe that the *smallest kerf width* from all the surfaces analysed is *0.862 mm* (surface III) and the *biggest kerf width* is *1.252 mm* (surface I).
Because the part was cut with the same cutting parameters, on every surface, the values should be approximately the same; thus, we will present only two cases: first one, when the cutting head moved longitudinally and second one, when the cutting head moved transversally. We will take the first two surfaces for analysing, by analogy being able to debate the other remaining surfaces.

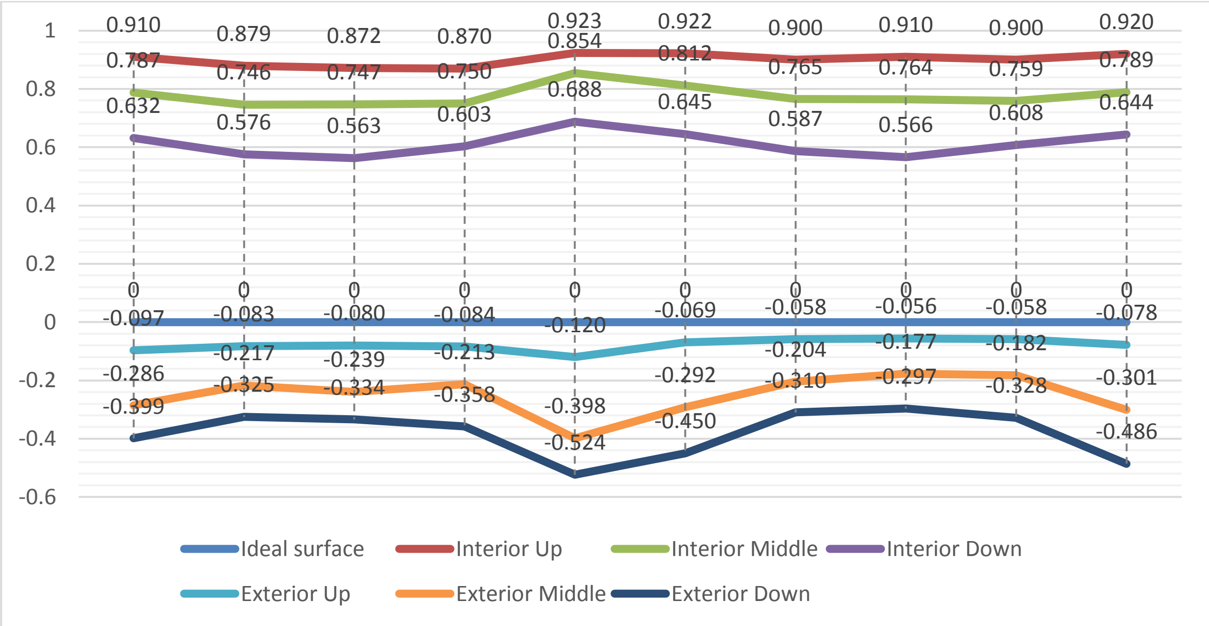


Fig. 6. Top view for the aspect of the surface I

In Figure 6, it can be seen the aspect of the cut, and also of the surfaces, both for the interior of the cut and also for the exterior. One can see that the surface is not straight, presents a V shape taper, which slides in a certain direction, towards the interior and the lines have an irregular aspect. Putting the values together, it was generated the aspect of the width of cut, shown in the Figure 7.

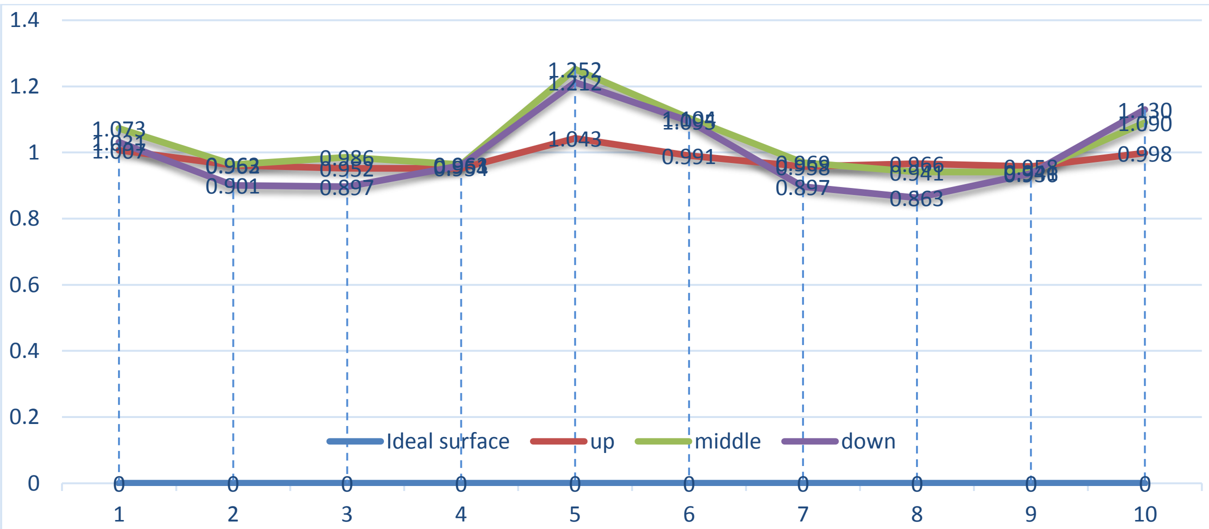


Fig. 7. The width of cut for surface I

As it can be observed in Figure 7, the bulging aspect of the surface is accentuated in the middle, between point 4 and point 7, and even more up in the corners, from point 9 to 10. That may result because of the machine error while switching the movement direction from one axis to another.

Analysing the next surface, presented in Figure 8, different aspects can be seen.

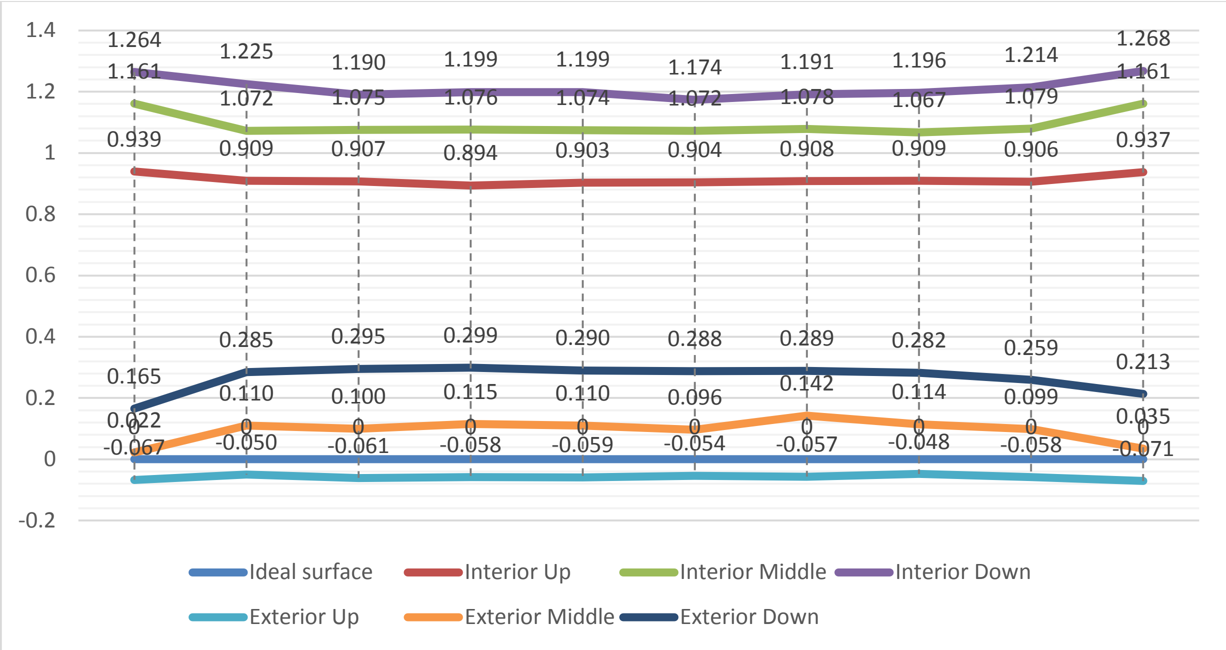


Fig. 8. Top view for the aspect of the surface II

In Figure 8, a major difference in relation to the surface I can be observed. The ideal surface is not incorporated in the exterior surface anymore. The surface presents a small V shaped taper, but in this case, the sliding is towards the exterior.

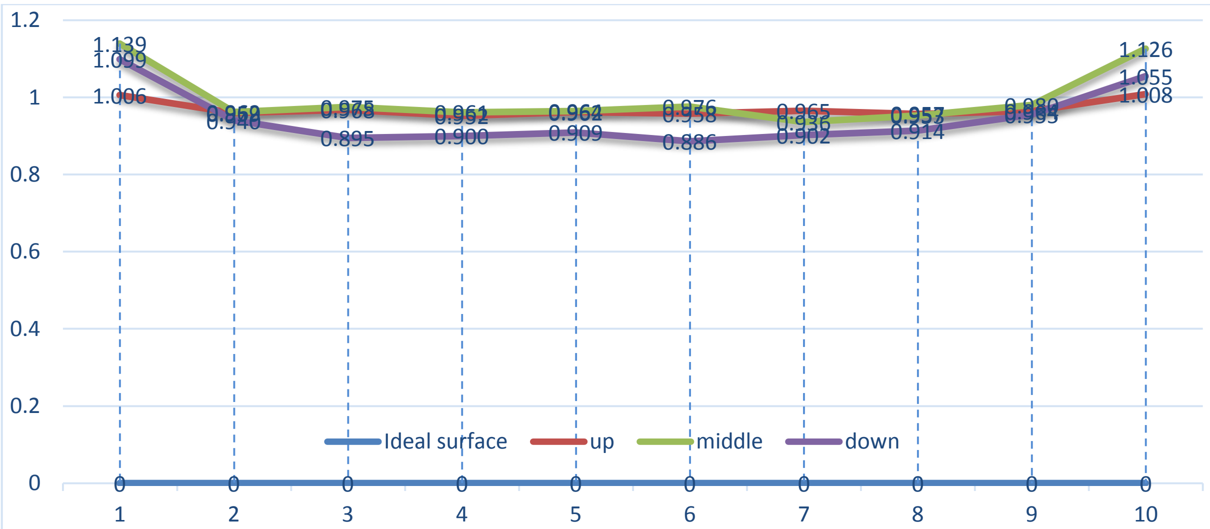


Fig. 9. The width of cut for surface II

As it can be observed in Figure 9, the surface has not the same bulging aspect in the middle, as in the Figure 7, the measured values outlining with great smoothness a surface that has a regular shape, compared to the width of cut for surface I.

5. Conclusions and future research directions

After analysing all the results of the described experiments, realised for a part cutted from a specified material (S355), the following conclusions can be highlighted:

a) The surface aspect has a slight deviation of profile, because of the error of the machine. The *smallest kerf width* vs. the *biggest kerf width* resulted is *0.863...1.252 mm* (surface I), *0.886...1.139 mm* (surface II), *0.862...1.128 mm* (surface III) and *0.907...1.147 mm* (surface IV). This confirms that the error of the machine is the one causing this, while it's switching the movement direction from one axis to another, but taking a look at the values of the kerf width on both surfaces analysed, the difference between the values isn’t very high. In order to verify this hypothesis, more experiments are required, for different cutting conditions.

b) The maximum stock needed to be left for machining with abrasive water jet (AWJ) is *1.252 mm* for a *30 mm* thickness steel S355 part.

c) The aspect of the surface after cutting it with abrasive water jet (AWJ) has a bulging in the middle (surface I) and it is slightly pointier in the corners (surface II).

References

[1] G. Fowler, “Abrasive water jet: controlled depth milling of titanium alloys“ (PhD. Thesis), University of Nottingham, 2003;
[2] P. Hlaváček et al., “Sandstone Turning by Abrasive Water jet“, Rock Mechanics and Rock Engineering,, vol. March, pp.1-5, 2015;
[3] S. Hloch et al., “Abrasive water jet (AWJ) titanium tangential turning evaluation“, Metalurgija, vol. 53,, no.4, pp. 537-540, 2014;
[4] J. H. Olsen, “Abrasive Jet Machining“, Manufacturing Engineering Handbook, chapter 38, pp. 1-8, 2004;
[5] I. A. Popan et al., “Research to improve the surface roughness of the parts made by water jet milling“,, Academic Journal of Manufacturing Engineering, vol. 11, no.2, pp. 105-109, 2013;
[6] J. Valíček et al., “Analogy between flexible abrasive water jet technology and traditional chip - Machining technology“, Materialwissen-schaft und Werkstofftechnik, vol. 46, no. 4-5, pp. 401-413, 2015;
[7] J. Valíček et al., “Study of surface topography created by abrasive water jet cutting“, Slavonski Brod, Strojarski fakultet u Slavonskom Brodu Sveučilišta J.J. Strossmayera u Osijeku, 2011;
[8] S. Ravai-Nagy, N. Medan, “Study of Surface Roughness for Steel Parts Cut with Abrasive Water Jets“, Magazine of Hydraulics, Pneumatics, Tribology, Ecology, Sensorics, Mechatronics (HIDRAULICA), no. 4, ISSN 1453-7303, pp. 12-17, 2016.

Design Details and Fluid Flow Analysis for the Centrifugal Pump with Special Rotor Pattern

Assistant professor **Fănel Dorel ȘCHEAUA**[1]

[1]"Dunărea de Jos" University of Galați, MECMET Research Center, fanel.scheaua@ugal.ro

Abstract: *The operating principle of a pump is represented by the conversion of mechanical energy taken from an energy source such an electric or thermal engine in hydraulic energy of the circulated fluid. By means of a pump, an energy amount is generated being uniformly distributed within the fluid mass that can defeat the hydraulic resistances and reaching a maximum height to which the fluid can be lifted by the pump. The centrifugal pumps are considered as hydrodynamic units that can perform the circulation of a fluid flow rate at a certain height. The primary functional sub-assembly found in the composition of a centrifugal pump is represented by the impeller which through the rotational movement inside the pump body performs the fluid transport from the aspiration region to the pump discharge region. Between the rotor blades and the working fluid some interaction forces are arising due to which the mechanical rotational energy is converted into flow energy of the fluid. In this paper the general considerations regarding centrifugal pump are presented and analyzed in terms of the functioning of a constructive type. Thus, a three-dimensional assembly of centrifugal pump was built having an impeller inside and analyzed with the ANSYS CFX program for emphasizing of fluid flow parameters. The results are presented in terms of pressure and flow velocity of the fluid regions.*

Keywords*: centrifugal pump, impeller model, fluid flow, 3D modelling, computational fluid dynamics (CFD)*

1. Introduction

The operation principle of centrifugal pumps is based on the fact that the working fluid is conveyed and delivered to a certain difference level ensured through the pump.
In the composition of a centrifugal pump enters the fixed part or pump casing having a specific design and the mobile part or rotor being positioned inside the casing.
In order to circulate the fluid the rotor has several blades through which the fluid can be driven from the pump aspiration area to the discharge area.
Centrifugal pump operation is represented by the conversion of mechanical energy taken from the power source represented by a thermal or electric engine in energy in hydraulic energy of the driven fluid.
The centrifugal pumps are considered as part of hydrodynamic or kinetic units capable of providing kinetic energy to the transported liquid characterized by mass flow rate and flow velocity.

2. General considerations regarding the centrifugal pumps

The operation of the centrifugal pump is ensured by the rotational movement of the rotor within the casing. As a result of this movement, the fluid particles are entrained from the aspiration region where a depression is created and sent to the peripheral region of the casing, by means of centrifugal force field that arises at the high velocity rotational motion of the rotor.
The fluid is continuously aspirated due to the continuous rotational motion of the drive shaft being conveyed through the discharge pipe. In the outlet region it has a high velocity, which means a high kinetic energy, but as it moves into the discharge pipe the motion velocity value gradually decreases and the kinetic energy is converted into pressure energy.
The rotor construction involves a number of angled shape blades being mounted on a shaft within the pump casing. The rotational movement of the impeller inside the pump casing is forming a field of centrifugal forces whose action causes fluid transport in a vortex-type movement depending on the blades profile.
By continuously rotational motion ensured by the rotor inside the pump casing the energy is generated within the fluid region which allows overcoming the hydraulic resistances and ensuring a height value at which the fluid is circulated.

The energy generated is uniformly distributed in the fluid mass within the pump and his specific increase describes a linear evolution represented by the load of the pump or the pump head, which can be calculated by the relationships: [5]

$$H_{pc} = \left(\frac{p_1}{\gamma_f} + \theta_1 \frac{v_1^2}{2g} \right) - \left(\frac{p_0}{\gamma_f} + \theta_0 \frac{v_0^2}{2g} \right) \tag{1}$$

$$H_{pc} = \frac{p_1 - p_0}{\gamma_f} + \frac{\theta_1 v_1^2 - \theta_0 v_0^2}{2g} \tag{2}$$

where:
- γ_f - fluid specific weight;
- g - gravitational acceleration;
- v - fluid velocity.

The pump head can be presented as a sum of total increases regarding pressure (static) and kinetic energy (dynamic).
The functioning of centrifugal pumps within an fluid pumping installation presents a dependency relationship between functional parameters represented by the fluid flow rate, pumping height, power consumption, pump efficiency and rotor velocity: [7]

$$f\left(Q_p, H, P_a, \eta_p, n\right) = 0 \tag{3}$$

where:
Q_p - fluid flow rate;
H - pump head;
P_a - absorbed power;
η_p - pump yield;
n - rotor rotational velocity.

$$Q_p = \frac{dV}{dt} \tag{4}$$

In order to determine the pumping height is applied the first principle of thermodynamics to the fluid unit mass circulated inside the pump casing: [7]

$$L = \Delta E_c + \Delta E_p + \Delta F_f \tag{5}$$

where:
ΔE_c - kinetic energy variation;
ΔE_p - potential energy change of position and pressure;
ΔF_f - energy amount dissipated and transformed into heat.

$$\Delta E_c = d\left(\frac{v^2}{2}\right) \tag{6}$$

$$\Delta E_p = gdz + \int V dp = gdz + \int \frac{dp}{\rho} = gdz + \frac{p_1 - p_0}{\rho} \tag{7}$$

For both values sets at the pump inlet and outlet can be written: [7]

$$L = \frac{v_1^2 - v_0^2}{2} + g\left(z_1 - z_0\right) + \frac{p_1 - p_0}{\rho} + F_f \tag{8}$$

Thus, the useful work done is described by the relation: [7]

$$L_0 = \frac{v_1^2 - v_0^2}{2} + g\left(z_1 - z_0\right) + \frac{p_1 - p_0}{\rho} \tag{9}$$

Dividing the useful work relationship with the gravitational acceleration can be achieved the pumping height (*H)*: [7]

$$H = \frac{v_1^2 - v_0^2}{2g} + z_1 - z_0 + \frac{p_1 - p_0}{\rho g} \tag{10}$$

The useful power can be defined as the total strength transferred to the working fluid that is transported through the centrifugal pump casing:

$$P_u = \rho g H Q \tag{11}$$

The absorbed power represents the power applied to the motor shaft in order to achieve impeller rotary motion within the pump casing and entrainment of the working fluid:

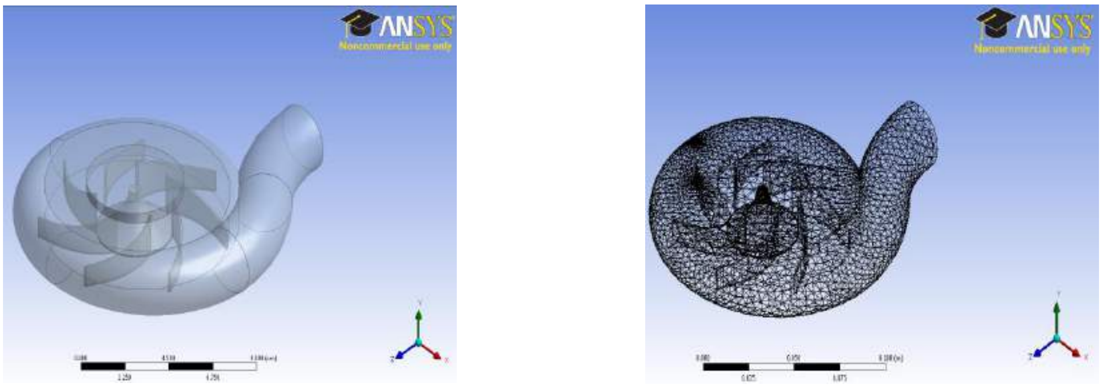
$$P_a = EI \tag{12}$$

Thus the pump efficiency can be calculated as the ratio between the useful power and the absorbed power:

$$\eta_{pc} = \frac{P_u}{P_a} \tag{13}$$

3. CFD analysis for the centrifugal pump with impeller model

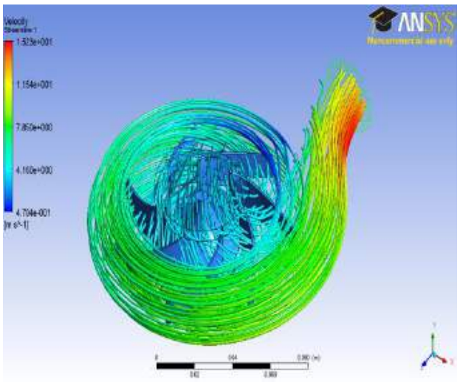
Two three-dimensional assembly models are constructed for the centrifugal pump assembly model and analyzed using ANSYS CFX in order to emphasize the fluid flow parameters represented by water velocity and pressure. In figure 1 and 3 are presented the analized models having the inlet diameter of 50 and 30 mm, while the outlet diameter is mantained at constant value of 30 mm. The working fluid is declared as water. There have been declared a rotational velocity of 1500 rot/min at the rotor and velocity at the inlet of 4 m/s. The pump impeller have a special design with seven angled blades capable of taking over the fluid, conduct it along the outer walls of the pump housing and then send it to the output region with a particular velocity.



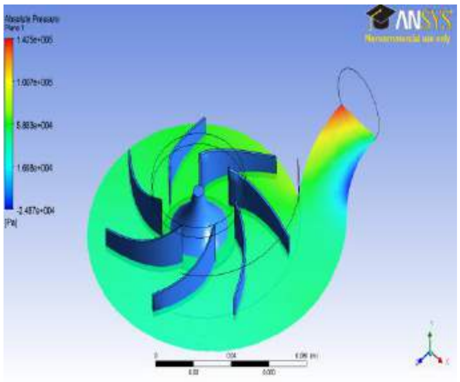
a) centrifugal pump model assembly **b)** mesh model of 11931 nodes and 58880 elements of triangle surface

Fig. 1. The centrifugal pump assembly model (D1=50 mm) imported in Design Modeler (ANSYS CFX) and mesh details

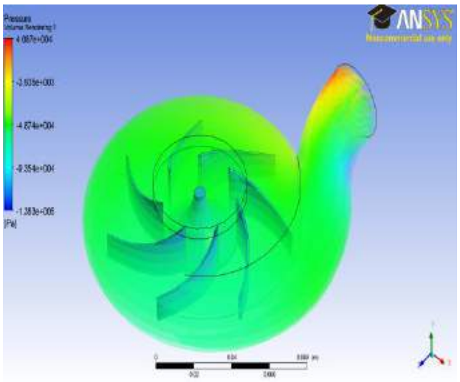
The obtained results from the analysis are presented in figure 2 and 4, showing the fluid streamlines and pressure values in the fluid region.



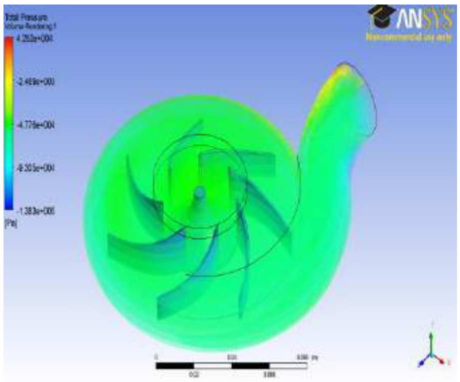
a) Fluid velocity [m/s] as fluid streamlines



b) Absolute pressure [Pa] in ZX plane

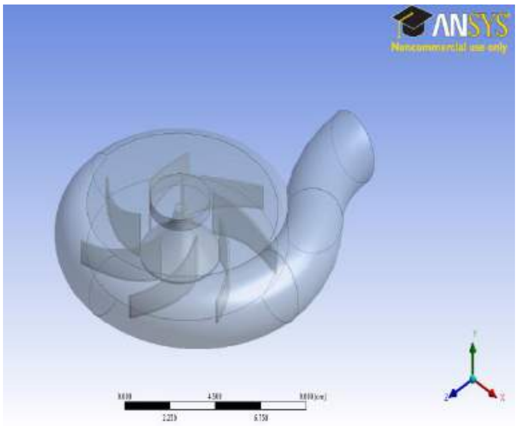


c) Static pressure [Pa]
(fluid volume rendering)

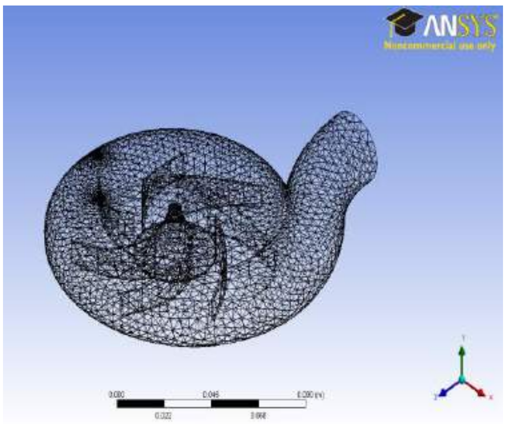


d) Total pressure [Pa]
(fluid volume rendering)

Fig. 2. The CFD analysis obtained results for model with inlet of D1=50 mm

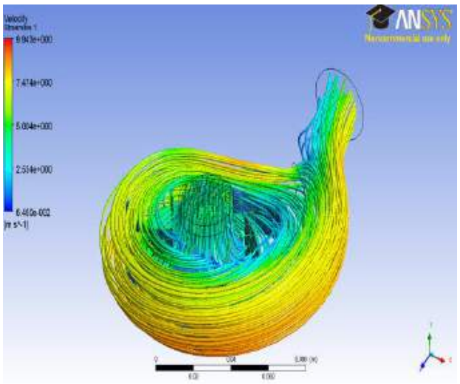


a) the centrifugal pump model (D=30 mm)

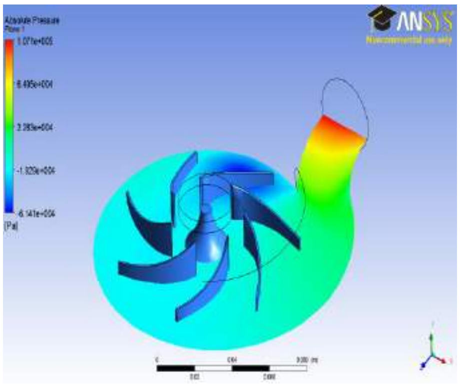


b) mesh model of 12183 nodes and 60280 elements of triangle surface

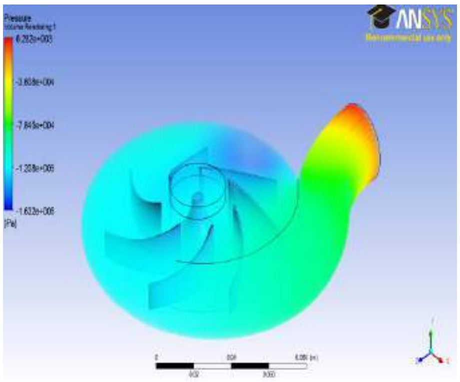
Fig. 3. The centrifugal pump assembly model (D2=30 mm) imported in Design Modeler (ANSYS CFX) and mesh details



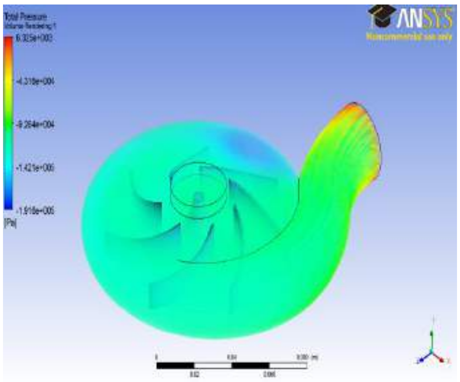
a) Fluid velocity results [m/s] as fluid streamlines



b) Absolute pressure [Pa] in ZX plane



c) Static pressure [Pa]
(fluid volume rendering)

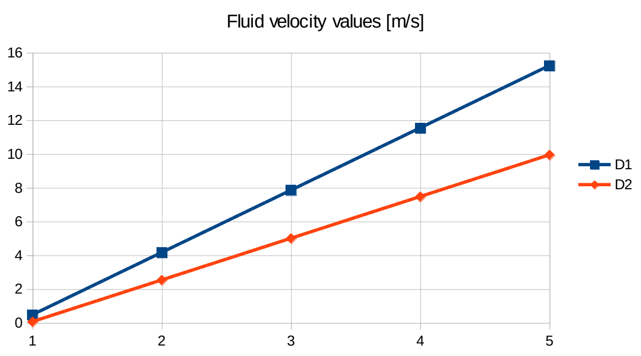


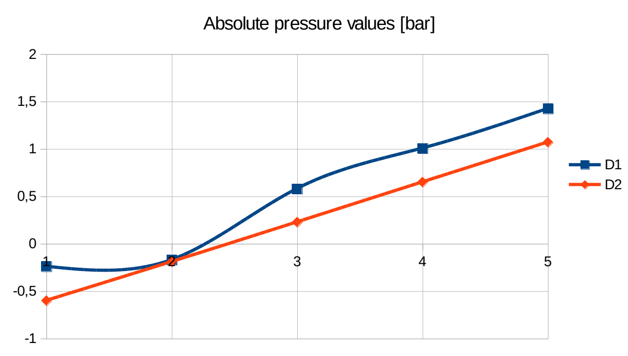
d) Total pressure [Pa]
(fluid volume rendering)

Fig. 4. The CFD analysis obtained results for model with inlet of D2=30 mm

The centrifugal pump model working principle was analyzed of fluid flow for the two values of inlet diameter (D1 = 50 mm, D2 = 30 mm). It was declared the fluid region having considered the impeller as inner solid rotating after OY axis. The reference pressure of the fluid in the region of 1 atm.

Table 1: The diagrams for fluid velocity and absolute pressure values for the two analyzed models





In Table 1 are presented the results diagrams for the theoretical fluid velocity values according with the inlet diameter of the centrifugal pump casing model. Also are presented the diagrams regarding the absolute pressure values for the two analyzed cases.

4. Conclusions

In this paper it was presented and analyzed in terms of the functioning a centrifugal pump model. The assembly consists of the fixed part - pump housing and the mobile part represented by impeller, located inside the pump casing.

The pump housing has been special designed having variable geometry at the outlet region. The casing was modelled for the two inlet diameter values of 30 and 50 mm, while the diameter of the fluid outlet was kept constant at 30 mm.

The numerical analysis performed using the ANSYS CFX program aimed mainly to emphasize the operating characteristics of centrifugal pump assembly when the impeller is performing a rotational motion within the casing with a velocity of 1500 rev/min. The results obtained from carried out analyzes shows the Eigen values for the working fluid velocity and fluid pressure created in the considered fluid region and specific fluid particles trajectory carried from the inlet region, along the pump casing walls inside and finally to the output region. For the carried on analyses the declared working fluid was water.

Such types of centrifugal pumps are used in particular for the construction of specialized circuits intended of water conveying or other types of fluids to a certain specific height depending on the pump construction type and dimensions.

References

[1] G. Axinti, A.S. Axinti, "Acționări hidraulice și pneumatice", Editura Tehnica-Info, Chișinău, 2009;
[2] A.S. Axinti, F.D. Șcheaua, "Introducere în hidraulica industrial", Editura Galați University Press, 2015;
[3] Al. A. Vasilescu, "Mecanica fluidelor", Ministerul Educatiei și Învățământului, Universitatea din Galați, Galați, 1979;
[4] F.D. Șcheaua, "Aspects Regarding the Special Hydraulic Distributors Operation", Hidraulica Magazine, ISSN 1453-7303, no.4, 2016;
[5] https://www.termo.utcluj.ro;
[6] https://biblioteca.proiecte/pompele-centrifuge;
[7] www.termo.utcluj.ro.

Experimental Determination of Pressure Losses when the Working Fluid Passes through Hydrostatic Directional Control Valves

PhD. Eng. **Corneliu CRISTESCU**[1]**,** PhD. Eng. **Cătălin DUMITRESCU**[1]**,**
Ph.D. Eng. **Radu RĂDOI**[1]**,** Dipl. Eng. **Alina Iolanda POPESCU**[1]

[1]Hydraulics and Pneumatics Research Institute INOE 2000-IHP, Bucharest, Romania,
cristescu.ihp@fluidas.ro

Abstract: *The article makes some general comments on the energy losses from hydrostatic drive system in order to increase their energy efficiency, as well as some theoretical consideration targeting the theoretical possibility of evaluating them, and in the second part of the article there is presented an experimental laboratory research to determine pressure losses when the working fluid passes through a hydraulic directional control valve commonly used in hydrostatic drives. The paper shows the schematic diagram, the experimental stand, as well as some graphical results achieved which, by comparison to the ones in the specialized literature, validate the qualitative and quantitative variation of pressure losses, and also validate correctness of the experimental testing methodology which has been used.*

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

1. Introduction

Modern hydrostatic systems are characterized by special technical performances, flexibility and low energy consumption because they are made using some modern components, with proper static and dynamic parametres, but also based on a **modern conception** in developing the schematic diagram of the hydrostatic drive. When designing them, there is performed a very careful analysis on mass and energy flow, but also on informational flow, identifying all critical points which can cause flow and pressure losses **in order to increase the energy efficiency** of the drive system.

A hydrostatic drive system consists of various components, through which the working fluid flows, and therefore this flow process through hydrostatic systems is always accompanied by **load losses**. In the design stage it is intended that these load (pressure) losses to be as small as possible, becauseall these load losses in hydrostatic drive systems implicitly lead to **high, unnecessary energy consumption**, that the power source of the equipment / machine must cover in addition to **useful consumption** that generates / ensures the required technical performances.

Load losses in hydrostatic drive systems reify in the end in **dissipating of part of the working fluid energy**, as this energy is transformed into heat energy, which is taken by the fluid, heating it, the system requiring sometimes cooling plants, which in their turn consume an extra energy to cool it. The higher **pressure drop** across each component of the hydrostatic system is, the more accentuated the above mentioned process will be.

The above presented facts are phenomena that exist and **always accompany flow phenomena** of working fluid, but the value of losses in the system should be as small as possible for **hydrostatic drive system** to be **efficient in terms of energy** [1].

That is why, when designing a hydrostatic drive system, it is necessary to use modern hydraulic components (pumps, directional control valves, motors, etc.), with high performances known from the related technical documentation, to assess, early in the design stage, pressure and flow losses, energy losses generally speaking, to guarantee high energy efficiency and high technical performances.

Worldwide one can notice intense concerns to develop hydrostatic drive systems with **power consumption as low as possible**. Since huge powers are circulated in the hydrostatic drive systems, the issue of energy consumption is particularly significant [2].

2. Some theoretical aspects regarding the flow of working fluid through hydraulic equipment

In the structure of all components of hydraulic systems, we meet, invariably, some specific elements in common: fixed or variable orifices, single or multiple: fixed or variable slits; variable **hydraulic contacts**, single or multiple [3].

A flow rate $Q\ [cm^3/s]$ passing through an **individual *hole*** of section $A\ [cm^2]$ is due to a hydraulic "tension" called pressure drop $\Delta p = p_i - p_e$ [daN/cm^2] and is generally characterized by a law of the following form:

$$Q = C_d A \sqrt{\frac{2}{\rho}}\ (p_i - p_e)^m = k\ (p_i - p_e)^m, \qquad (1)$$

where: C_d [-] = $\sqrt{1 / \xi}$ at ζ = 1.8 (typical value), C_d = 0.7, and $\rho\left[\frac{daNs^2}{cm^3}\right]$ is fluid density.

If a fluid flows through a number of *n multiple* holes, in order to determine the unknown elements of flow it is necessary to solve a system of similar equations.

A flow rate $Q\ [cm^3/s]$ passing **through a slit** of some profile is also due to a pressure drop $\Delta p = p_i - p_e$, but it is generally characterized by a linear law of the following form:

$$Q = \delta\ (p_i - p_e) \qquad (2)$$

δ values, for usual cases, are indicated in the literature [3].

In drive systems there also occur the so called "hydraulic contacts".

Hydraulic contacts are flow spaces between the fixed and moving parts of hydraulic devices, for example between the body and the slide valve of a hydraulic directional control valve, and they can be individual or multiple.

Individual* hydraulic contact** is characterized by a **flow law** of the form of equation (1), in which, this time, flow area *A* is a function of opening ***x of the contact, and the flow is a function not only of Δp but also of *x*:

$$Q = C_d A(x) \sqrt{\frac{2}{\rho}}\ (p_i - p_e)^m = C_d \sqrt{\frac{2}{\rho}}\ A\ (x)(\Delta p)^m \qquad (3)$$

Equation (3) expresses a non-linear connection between Q, x, Δp, even if *m* would be equal to 1, due to the product between the variables *A(x)* and Δp.

***Multiple* hydraulic contact** characterizes, in fact, **hydraulic directional control valves** used in hydraulic drive systems or in automatic control systems. Unlike the single contact, in this case there occur in a single functionality **simultaneous flows through a set of contacts** connected together in series and / or parallel. To determine the relationship of interest, between the flow and pressure, of the form: $Q = f\ (x, \Delta p)$, where $\Delta p = p_i - p_m$ (and p_m= pressure drop in the driven hydraulic motor) there should be taken into account possible flows not only through the contact of the directional control valve, which connects the pump and motor input, but through all the contacts that change jointly (with the same variation of opening *x*) with the mentioned contact, upstream or downstream from it. The situations encountered in practice are numerous, and they depend on the following main factors: type of power supply (with constant pressure or constant flow), type of directional control valve (with slide valve - 1, 2, 4 active edges - or clack valve fitting), type of driven motor (differential or non-differential), slide valve type (with positive coverage, null or negative coverage, symmetric or asymmetric) [3].

***Unlinearized* characteristic equation of directional control valves** with **negative coverage** (x_a), supplied at constant pressure, figure 1, is very complicated, but it can be expressed in **the linearized form** as follows:

$$\Delta Q_m = A\ C_0 \Delta x_m + \frac{A^2 C_0}{E_0}\ \Delta p_m \qquad (4)$$

where C_0 is the coefficient of flow amplification, and E_0 is the coefficient of force amplification.

Figure 2 shows a flow / working diagram through a directional control valve, 4-way, 3-position, with **positive coverage** and closed center, on this position practically flow is null, as well as pressure drops / losses. On a side position, fluid flow causes pressure loss / drop, and flow gets to the motor

/ consumer, almost entirely, except flow losses / leakages occurring through the gap between the body and slide valve / rod of directional control valve, q_{sd}, which are very low. When the fluid passes through the hydraulic rotary motor MHR, besides pressure losses / drops, there also occur internal flow losses / leakages, which go directly to the tank, but are also low.

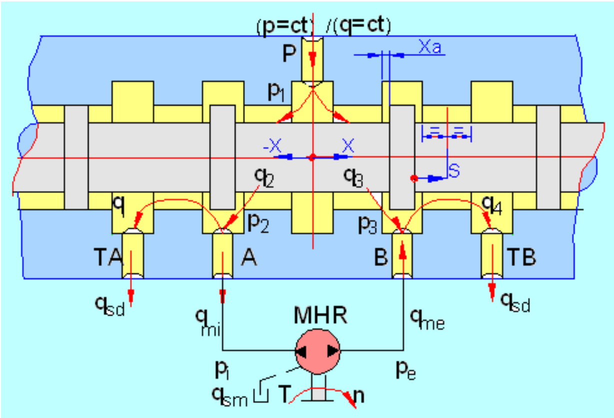


Fig. 1. Flow diagram through a 4-way open center (negative coverage) directional control valve

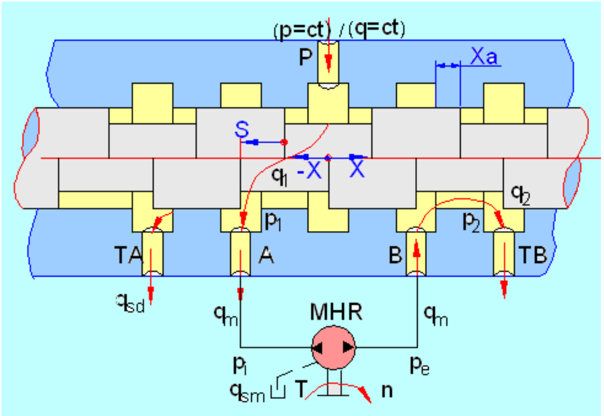


Fig. 2. Flow diagram through a 4-way, close center (positive coverage) directional control valve

As one can seen from the above, the **issue of determining theoretically** the flow flowing through a directional control valve is very complicated, in fact, the same as the issue of computing pressure losses or drops, depending on the flow which passes through the directional control valve in question. Therefore, the only accurate method for determination of pressure losses when a fluid flow circulates is the **experimental method**, which, indeed, requires adequate laboratory infrastructure. The following shows an experimental research to **determine pressure losses** as function of flow; this research has been conducted in the laboratories of INOE 2000-IHP, [4], [5].

3. Presentation of infrastructure for experimental determination of pressure losses

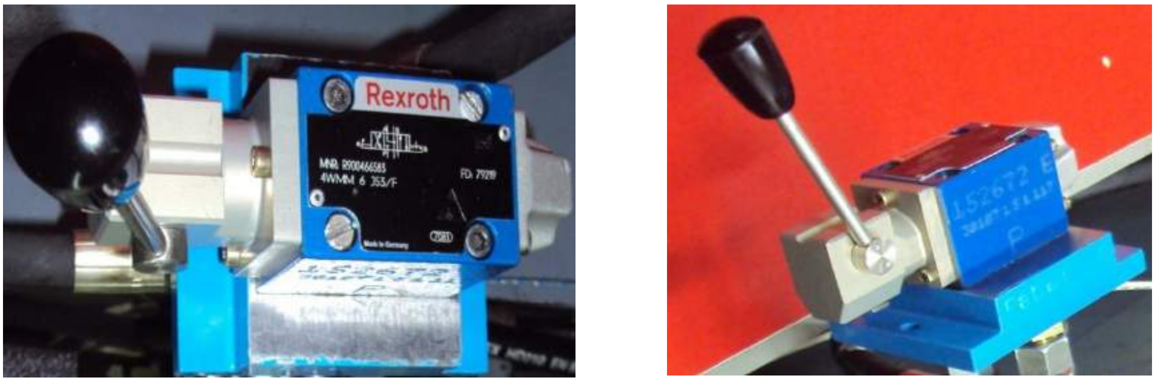
Besides lowering hydraulic drive system efficiency, losses in the system can also lead to degraded technical performances. The designer of drive systems must **identify the source of those losses** and deepen the underlying phenomena, and, even more, has to know how to limit the effects of such losses. One of the assessment methods of energy losses through different devices in the structure of hydrostatic drive systems is the experimental testing on appropriate stands. The following shows a methodology for experimental determining of pressure losses when the working fluid is passing through a hydraulic directional control valve currently used in hydrostatic drives.

3.1. The object under experimental research

As the object of experimental research there has been selected a directional spool valve, directly operated, with manual actuation, manufactured by REXROTH BOSCH GROUP company, code 4 WMM 53 6 J / F, shown in figure 3 [6]. From working diagram of the device, displayed on its label, figure 3a, there results that it is a 4-way, 3-position directional control valve; positions were obtained by manual control of the lever, having the central position with ports A and B to the tank, port P - closed. By pulling the lever, figure 3b, the result is connecting the port P to port B, this being the working position on the stand for determining pressure losses through the hydraulic directional control valve.

The main technical characteristics of the directional control valve are the following:

- Size 6, 4/3 directional design;
- Component series 5X;
- Maximum operating pressure 315 *[4569 psi*;
- Maximum flow 60 l/min *[15.8 US gpm]*;
- Types of actuation: and lever.



a) Top view **b)** Side view

Fig. 3. Directional spool valves, directly operated, with manual actuation, Rexroth Bosch Group Co. [6]

3.2. Presentation of experimental stand

To conduct the experimental research, aiming to determine the pressure losses when the working fluid passes through the hydraulic directional control valve presented, it was necessary to design and construct a test stand. The stand should make it possible to carry out imposed tests for measuring pressure at the device input, and respectively output. It also must allow flow variation in the working range of the directional control valve, having thus the possibility of recording, storing and processing the experimental data. The ultimate goal of the experimental research is to plot the variation diagram of pressure losses depending on the flow of fluid which passes through / transits the directional control valve.
Based on these requirements, there has been designed a functional diagram, shown in figure 4, based on which there has been constructed a test stand, primarily based on components existing in the Laboratory of Hydraulic Servosystems at INOE 2000-IHP, but also based on some components especially procured for that end.

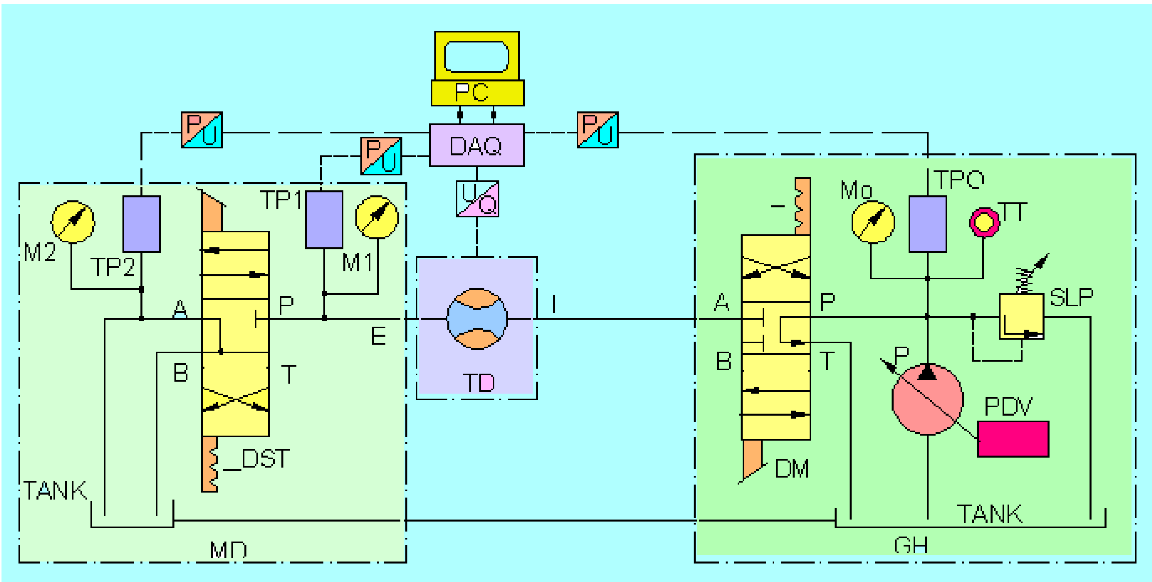


Fig. 4. Conceptual diagram of the test stand

The experimental stand consists of four main parts, namely:
- **Source / station of hydraulic power generation GH,** which consists mainly of a pump P, variable displacement, equipped with device for varying the flow PDV, assisted by a pressure limit valve SLP, flow fluid being sent to a manually operated 4-way, 3-position hydraulic directional control valve DM, on the central position the oil being returned to the tank. Oil / fluid pressure can be visually traced on a manometer M_0 and taken by the pressure transducer TP0, in order to be acquired, recorded and processed by the computer system. Oil temperature is visualized and

taken by the temperature transducer TT, in order to be acquired and registered by the computer system;
- **Module for determining the pressure losses MD**, which consists primarily of hydraulic directional control valve being tested DST, which is equipped on the inlet circuit, with a manometer M1 for viewing the oil pressure at the directional control valve input, as well as a pressure transducer TP1, for pressure signal to be taken over by the computer system. Moreover, on the output circuit, there is provided a pressure gauge M2 for viewing the oil pressure at the directional control valve output, as well as a pressure transducer TP, for pressure signal to be taken over by the computer system. After passing through the hydraulic directional control valve output, the fluid /oil reaches the second tank, which communicates, through a pipe, with the first one;
- **Flow transducer TD**, mounted between the source of working fluid generation GH and the module for determining the pressure losses MD. Flow transducer sends a signal of flow variation to the computer system, to record and process data;
- **Data acquisition, recording and processing computer system**, composed mainly of an acquisition board DAQ and a PC-type computer, and also processing **software** appropriate for the application.

Figure 5 shows a general view of the experimental stand, and figure 6 presents a view on location of the hydraulic devices that really put in practice the diagram for experimental determining of pressure losses when the working fluid passes through the directional control valve under tests.

Fig. 5. General view of experimental stand

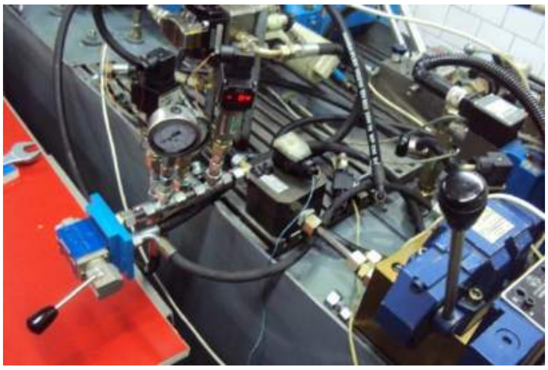


Fig. 6. View on location of hydraulic devices

A top view on location of hydraulic devices is presented in figure 7, and mounting of manometers and pressure transducers on the board of directional control valve can be seen in figure 8.

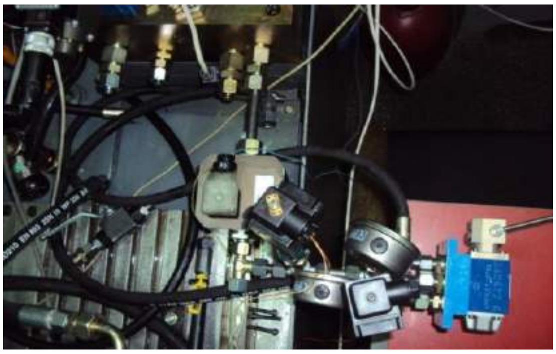


Fig. 7. Top view on location of devices

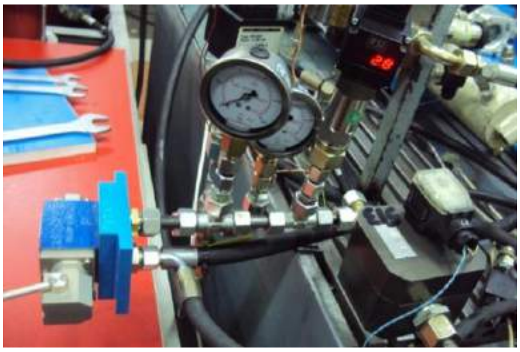


Fig. 8. Mounting of devices at directional control valve

As one could see from the description of the schematic diagram of the test stand, for the pressure, flow and oil temperature variation signals to be taken over, there have been used high performance transducers, namely: figure 9 presents pressure transducers of the type Genspec GS200, with instantaneous display of pressure value, mounted on the inlet, and respectively outlet circuits of hydraulic directional control valve, and figure 10 presents the gear flow transducer, type KZA-1865R20. Figure 11 shows both the temperature transducer manufactured by the company

TURCK and the proportional pressure control valve manufactured by the company REXROTH BOSCH GROUP.

Fig. 9. Pressure transducers with display

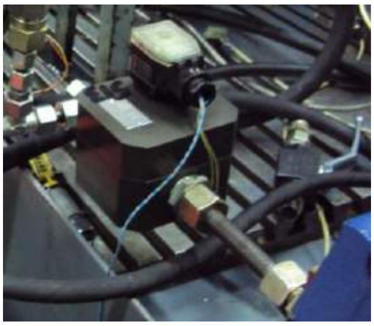


Fig. 10. Gear flow transducers

Fig. 11. Pressure transducers and pressure control valve

In order to acquire and process the signals from the transducers, there has been used **an electronic computer system**, which besides the electric panel for control, display and tracking of tested parameters, is also equipped with a PC–type computer, as one can see in figure 12, inclusively supply sources and electronic modules for signal accommodation, as well as an NI USB – 6218 **acquisition board** manufactured by National Instruments, shown in figure 13.

Fig. 12. General view of the computer system

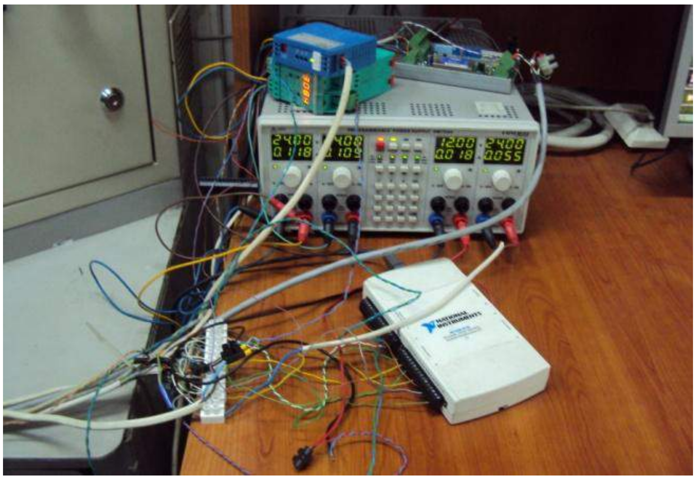


Fig. 13. Supply source and aquisition board

The methodology of conducting the experimental research

In order to determine the pressure losses when the fluid passes through hydraulic directional control valve, the infrastructure presented above has the one being used, and, generally, we proceeded as follows:

- Using the components presented above, we have done their mounting according to the schematic diagram of the stand, shown in fig. 4;
- After the mounting of hydraulic components was completed, there has been done a checking of all hydraulic circuits and sequential tests have been carried out, with manual controls, on operation;
- There have been created all electric and electronic connections imposed by the schematic diagram and by the actual requirements of each individual component, in order to assure electric supply, necessary controls, as well as processing / acquisition of signals from transducers, to store and process them by computer;
- There have been elaborated **special software dedicated to this application**, in the graphical programming environment LabVIEW; its schematic diagram is presented below, in figure 14;
- Determinations, in semiautomatic mode, of pressure loss variation diagrams, depending on the flow that transits the device, have been based on a **proportional electric control** of the PDV device, shown in figure 4, for variation of the pump P displacement, in order to achieve variable flow rate at the pump P, in the working range of the directional control valve. Proportional electrical control, presented in the fig.15, from 0 to 5A, and then from 5 to 0 A, aimed both at continuous increase in the flow up to the maximum value considered as acceptable and at continuous decrease of it, from the maximum value down to the minimum respective value;
- Data regarding variation of interest parameters have been acquired and processed by computer, based on the elaborated software, and presented both in graphical form and as folders of numerical values. Thus, we have achieved variation of pressure losses, in both flow variation directions, and the final diagram resulted revealed the occurrence /existence of **hysteresis**, a phenomenon which is known / present in technical systems.

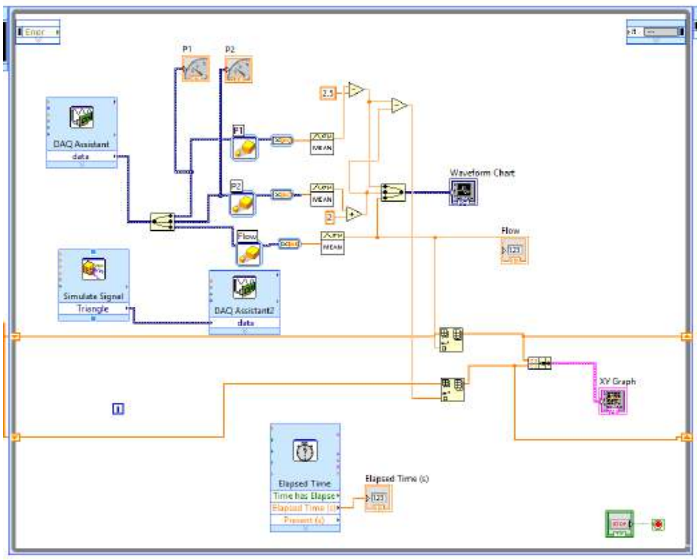


Fig. 14. Logical diagram of the application-dedicated software

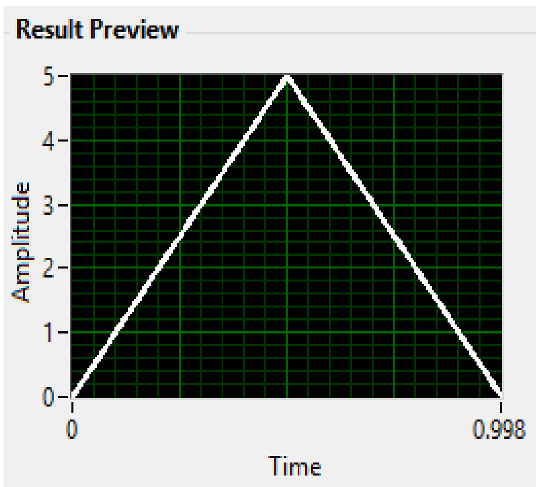


Fig. 15. Proportional control chart of pump flow

The use of testing infrastructure, presented above has resulted in obtaining several sets of extremely interesting experimental results, which allowed plotting the variation curve of pressure losses, when the working fluid passes through the hydraulic directional control valve under tests, depending on the oil flow that transits the device, an example being presented below.

4. Presentation of several experimental results achieved

As a result of conducting the experimental research concerning determination of pressure losses when the working fluid passes through hydraulic directional control valves, we have achieved **a set of complex** data **folders**, registered and processed by computer, which include: a numerical text folder, a folder with plots on the screen, regarding variation of pressure p1, at directional control valve input, variation of pressure p2, at directional control valve output, variation of flow Q, and also **the goal chart of the research**, namely **variation of pressure losses through the directional control valve depending on the** oil **flow variation**, both at increase in flow rates and at decrease in the rates. All these diagrams are obtained automatically on computer, based on the **special software elaborated** for this application. From this set of complex diagrams achieved, figure 16 presents **an example of computer screen** with variations of the interest parameters mentioned above.

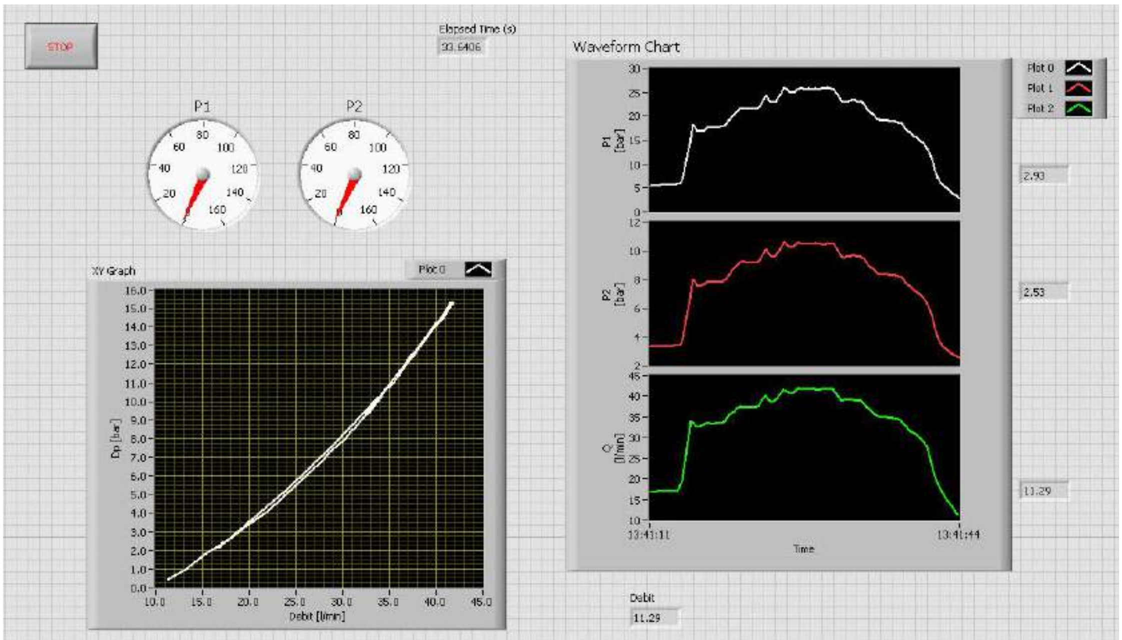


Fig. 16. Pressure and flow graphical variations on the computer screen

For a numerical assessment of pressure loss variation depending on flow variation, from the "txt" numerical folder there have been selected the values shown in table 1, when flow increases, and in table 2, when flow decreases.

Table 1: Variation of pressure losses when flow increases

Q [l/min]	17	19	24	32	34	36	38	40	42
Δp [bar]	2.3	3.2	5.3	9.3	10.2	11.t	13.2	14	15.3

Table 2: Variation of pressure losses when flow decreases

Q [l/min]	40	38	36	34	32	30	28	25	20
Δp [bar]	14.0	13.1	11.0	10.3	9.0	8.0	7.0	5.6	3.3

Figure 14 also shows the existence of **the goal chart of the research**, which presents **variation of pressure losses through the directional control valve depending on the** oil **flow variation**, both at increase in flow rates and at decrease in these rates; this chart after being processed by computer, looks like in fig. 17.

For validation of the results achieved and confirmation of the testing methodology used, figure 18 shows a multiple diagram, obtained / conceived by the company REXROTH BOSCH GROUP.

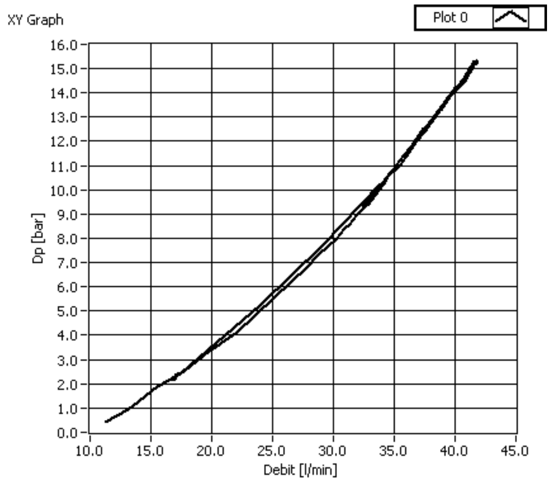


Fig. 17. Variation of pressure losses depending on flow

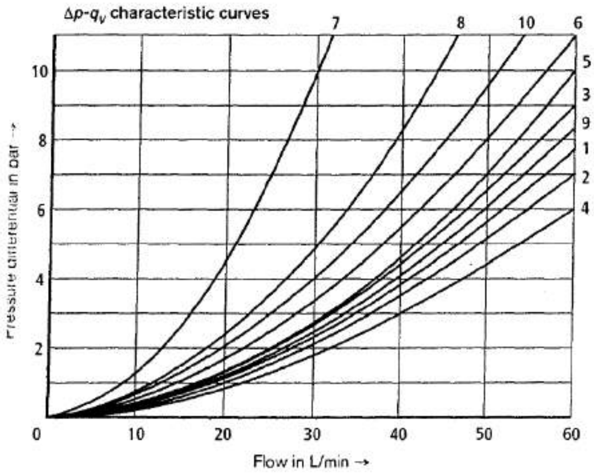


Fig. 18. Variation of pressure losses in literature [6]

From the **comparative analysis** of the diagram obtained in the laboratory experimental research, presented in fig. 17, and the multiple diagram in fig.18, which is elaborated for various schemes of directional control valves, we can conclude that the shape of pressure loss variation is identical, and in terms of value, the resulted graph is placed between graph 7 and graph 8 in the diagram in fig.18.
On the other hand, the experimental research conducted in the institute INOE 2000-IHP reveals existence of a hysteresis between flow increase phase and flow decrease phase, actually an expected phenomenon, and the obtained graph quantifies exactly its value in each point.

5. Conclusions

In the first part of the article, there are presented some general theoretical considerations concerning quantification / assessment, in theoretical mode, of pressure losses when the fluid flow passes through the hydraulic directional control valves used in hydrostatic drive systems.
In the second part of the article, there is presented a laboratory experimental research, concerning determination of the pressure losses when the fluid flow passes through a hydraulic directional control valve with rated diameter of 6 mm, for which there has been designed and developed a complex test stand that includes a computer system for data acquisition, registration and processing, facility that made it possible to obtain of a complex graph, that includes both variations of input and output flow pressure rates and a final graph of variation of pressure losses when the variable flow passes through the directional control valve under testing.
In comparison to the data from literature, we conclude that the shape of the pressure loss variation curve is identical to the known one, and in terms of value the resulted graph is placed in normal domain, corresponding to various schemes of functionality of hydraulic directional control valves of the given size.
The original part of this research is the fact that, based on special software elaborated for this application, in the resulted graph **there also occurs hysteresis** between the flow increase and decrease phases.
Based on the achieved graphical and numerical results, but especially by comparison to the data in the specialized literature, we can conclude that **the methodology used is correct**, it corresponds to the intended purpose and can be extended also to other sizes and dimensional types of hydraulic directional control valves.

Acknowledgement

This paper has been developed in INOE 2000-IHP, with financial support of ANCSI, under the national research *Programme NUCLEU-2016,* project title: *Physics of processes for reducing energy losses and developing renewable energy resources by use of high-performance equipment,* project code PN 16 40.03.01, financial agreement no. 5N/2016, Additional act no. 1/2017.

References

[1] G. Axinti, “Acţionări hidraulice şi pneumatice. Vol. I. Componente şi sisteme, funcţii şi caracteristici” (Fluid Power systems. Vol. I. Components and systems, functions and features), Tehnica-Info Publishing House, Chisinau, 2008;
[2] M. Avram, “Acţionări hidraulice şi pneumatice” (Fluid Power systems), University Publishing House, Bucharest, 2005;
[3] V. Marin et al., “Sisteme hidraulice de acţionare şi reglare automată” (Automatic drive and control hydraulic systems), Technical Publishing House, Bucharest, 1981;
[4] C. Cristescu, C. Dumitrescu, G. Vranceanu, L. Dumitrescu, ”Considerations on Energy Loses in Hydraulic Drive Systems”, In: Hidraulica Magazine, no. 1/2016, pp. 36-46, ISSN 1453 – 7303;
[5] P. Drumea, C. Dumitrescu, Al. Hristea, A.-M. Popescu, ”Energy Use in Hydraulic Drive Systems Equipped with Fixed Displacement Pumps**”,** In: Hidraulica Magazine, no. 2/2016, pp. 48-57, ISSN 1453 – 7303;
[6] Rexroth Bosch Group, “Directional Spool valves, direct operated with manual actuation”, On: https://www.boschrexroth.com/en/us/products/product-groups/goto-products/goto-hydraulics/directional-valves/4wmm-and-4wmr/index.

Considerations regarding Aerodynamic Interaction between Two Wind Turbines. Case of Study: Two Wind Turbines with Rotor Diameter of 6 Meters

PhD. Eng. **Adrian POPA**[1], Prof. PhD. Eng. **Beazit ALI**[2], PhD. Eng. **Ionut Cristian SCURTU**[3],

[1,2,3] Naval Academy "Mircea cel Batran", Constanta, Romania, scurtucristian@yahoo.com

Abstract: *In this paperwork are presented results of a study regarding the aerodynamic interactions between two turbines. Each turbine is a 3 blades turbine, with the rotor diameter of 6 meters. This study concludes that the optimum distance between wind turbines should be about 6 times the diameter of the rotor (36 meters in this case).*

Keywords: *Fluid flow engineering, Structural engineering, Wind energy*

1. Introduction

Nowadays, there are a lot of wind farm sites on development. The wind energy is one of the cleanest energy in the world, with a minimum effect on the environment. In order to obtain the maximum power on a available area, we need to know how the turbine interact with each other. In this paperwork, there are studied 2 turbines, having the rotor diameter 6 meters. Increasing the distances between them the authors found the distance where the aerodynamic influence is neglectable.

2. Wind turbine geometry

The studied type of the wind turbine is presented below in a 3D cad geometry:

Fig. 1. The studied wind turbine

During the study, there are considered two turbines, having different distances. The distances between turbines are measured starting from the hub's rear end of the first one, up to the hub's front end at the second one. These two turbines are situated in a 50m cylindrical air domain. The length of the cylindrical varies, and will be presented in the following sections of this paperwork.

3. Study cases

For this paperwork, there are considered 9 study cases (abbreviated as SC), with different distances between rotors according to Table no.1.

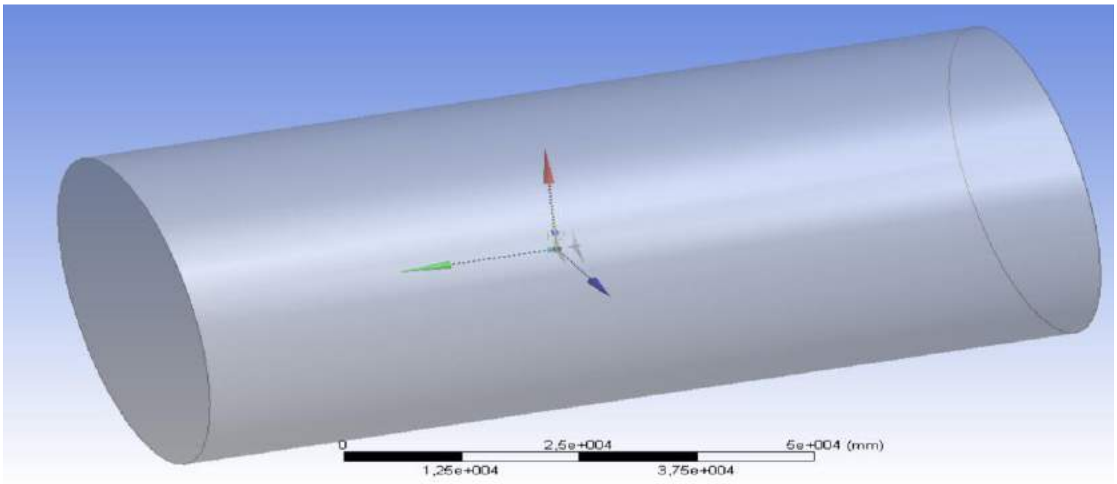


Fig. 2. SC1 Geometry

4. Mesh

The mesh was generated automatically, as a CFD mesh [1,2,3] with a relevance parameter of 100 (the biggest value for relevance).

Statistics for mesh are presented in the bellow table:

Table 1: Geometry details and mesh statistics for each study case

Study case	Distance between rotors	Number of Nodes	Number of Elements
SC1	0	260254	1535479
SC2	3	280742	1657594
SC3	6	290952	1718499
SC4	9	280294	1654775
SC5	12	266769	1575242
SC6	18	253733	1498090
SC7	24	255032	1505591
SC8	30	242813	1433612
SC9	36	220523	1301465

Mesh discretization networks are presented below, in figure 3 as example.

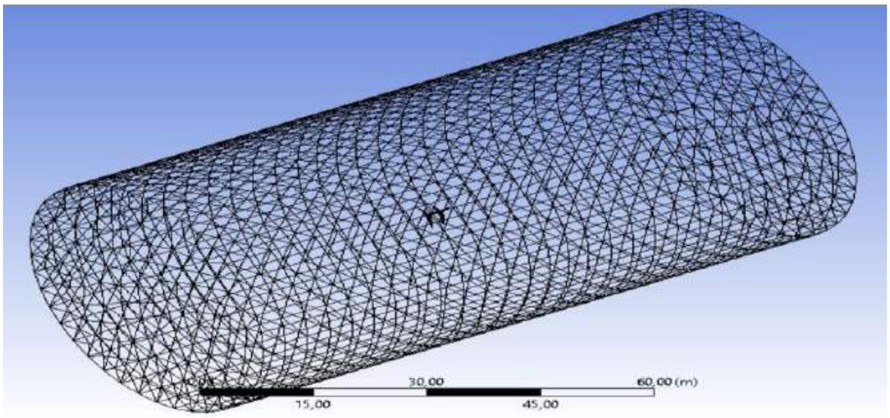


Fig. 3. SC1 Discretization network

5. Analysis setups and boundary conditions

The carried-out analysis is transient analysis, with a total time of 300 seconds, with 1 second time step.

The turbulence model is considered to be the Shear Stress Transport.

Boundary conditions are:

- The inlet of the cylindrical domain is a constant speed one, speed is 10 [m/s];
- The outlet of the cylindrical domain is a constant pressure one, with relative pressure 0 [atm];
- The cylindrical border is defined to be "free sleep wall";
- The turbines are defined to be "smooth wall"

Graphical representations of the boundary conditions and axes definitions are presented in the following picture:

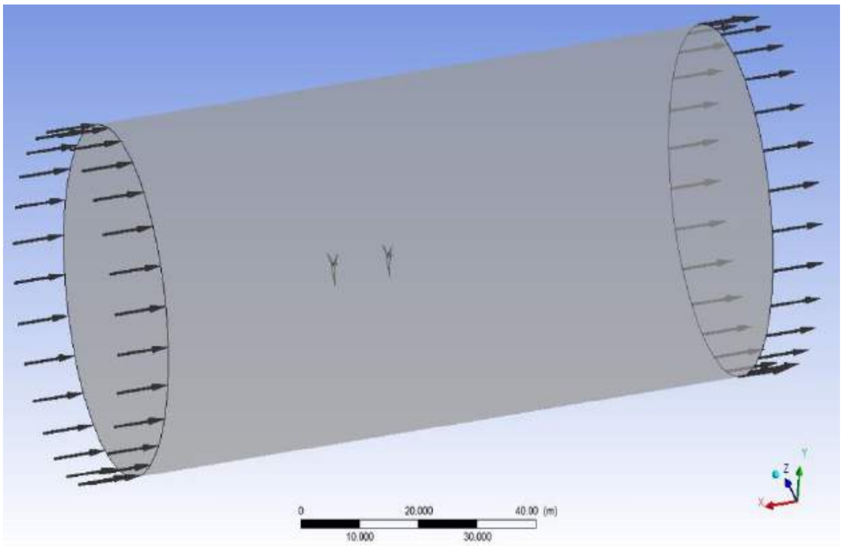


Fig. 4. SC4 Graphical representation for the boundary conditions, and axes definitions

6. Graphical results

In the bellow picture are presented graphical results regarding the flow around the two turbines and the pressure distribution diagram on the turbines:

6.1 Speed lines around turbines

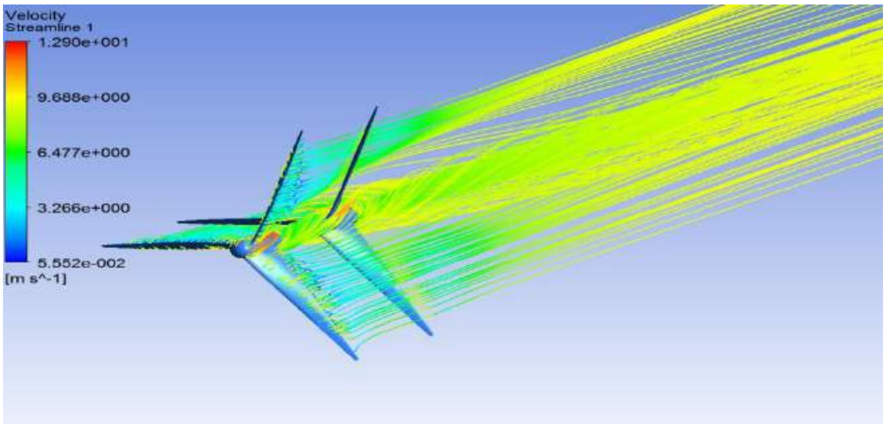


Fig. 5. SC1 Speed lines around turbines

6.2 Relative pressure distribution on turbines

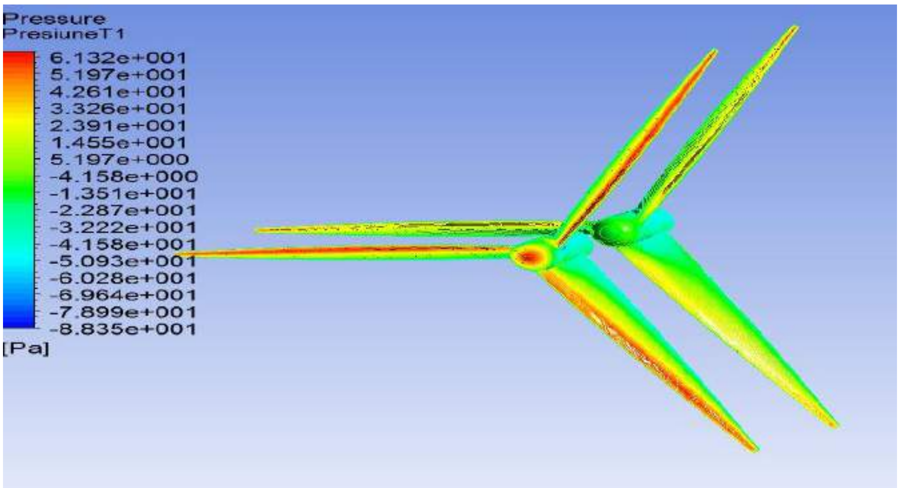


Fig. 6. SC1 Relative pressure on turbines

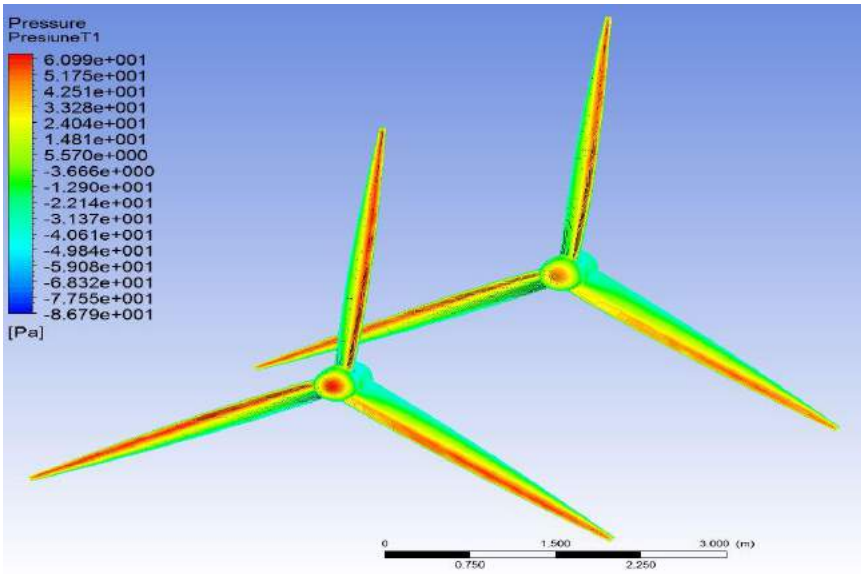


Fig. 7. SC5 Relative pressure on turbines

7. Numerical results

For numerical results, this study is considering the torque on the turbines. For this study, the Ox Torques are relevant [4,5,6].

The axes are defined in Fig. **4**.

The relevant results are presented below:

Table 2: Absolute values of the Ox torques on the wind turbines

Distance between hubs [m]	Ox torques (absolute values) [KNm]	
	Turbine 1	Turbine 2
0	1513.4	916.37
3	1549.4	1165
6	1558.6	1317.6
9	1564.4	1347.2
12	1578.5	1389
18	1604.6	1440
24	1602.1	1471.3
30	1619.7	1513.2
36	1647.8	1564.4

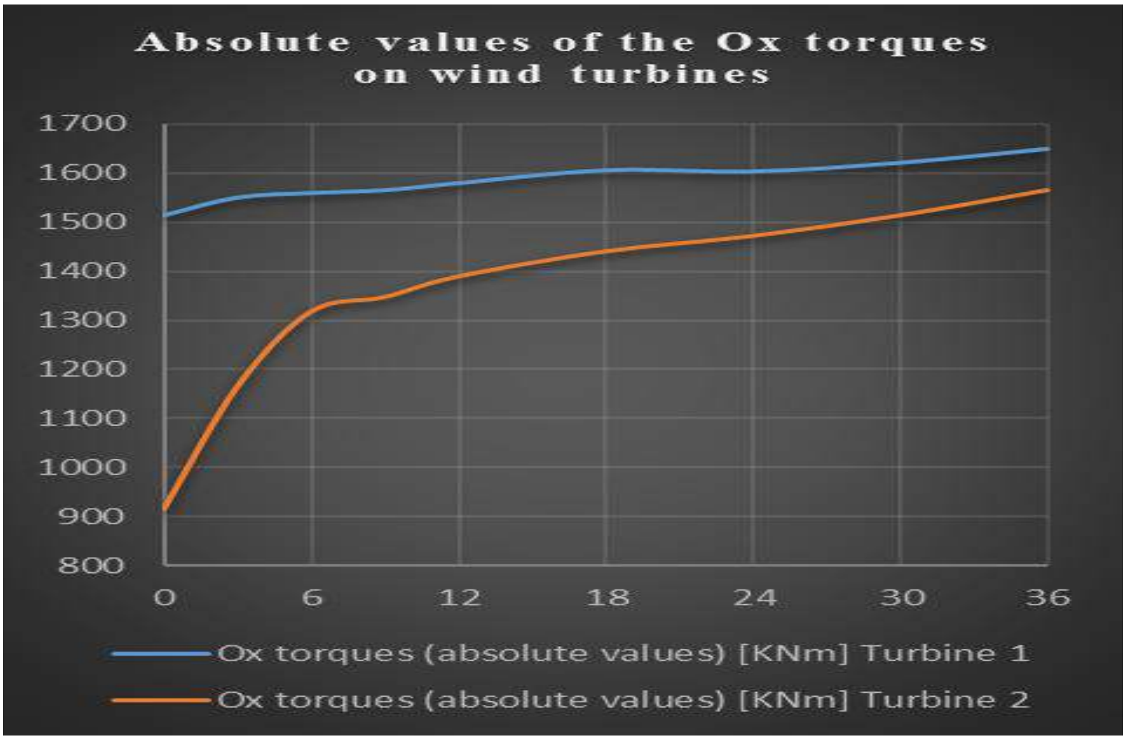


Fig. 8. Absolute values of the Ox torques on the wind turbines

8. Conclusions

Considering graphical variation presented in Fig. **8**, we can see that both turbines are influencing each other.
The most affected one is the Turbine 2, because Turbine 1 is in front of it and "takes" the air.
Also, the Turbine 1 is affected by the second turbine, which puts a backpressure and affects in a negative way the flows around it.

Table 3: Percental differences between toques on wind turbines

Distance between hubs [m]	Percental differences between toques on wind turbines
0	39.45
3	24.81
6	15.46
9	13.88
12	12.01
18	10.26
24	8.16
30	6.58
36	5.06

As it can be seen in Table **3**, at the distance of 36 meters (6 times the diameter of the turbine rotor) between hubs, the percental differences between turbines goes to about 5% and bellow. This value can be acceptable in most cases, considering the limited area available for the wind farm. As a final conclusion, the authors of the study recommend the wind farm developers to plant the wind turbine at intervals equal with 6 times diameter of the rotor.

References

[1] www.ansys.com;
[2] L . Domnisoru, E . Găvan, O Popovici, "Analiza structurilor navale prin metoda elementului finit", Editura Didactica si Pedagogica, Bucuresti 2005, ISBN 973 – 30 – 1075 – 8;
[3] Wei Tong, ed., "Wind Power Generation and Wind Turbine Design", WIT Press, 2010, ISBN 978-1-84564-205-1;
[4] I. Călimănescu, L. C. Stan, "Computer fluid dynamics (CFD) study of a micro annular gear pump", ATOM 2016, Conference Paper;
[5] L. C. Stan, I. Călimănescu, "A new innovative turbocharger concept numerically tested and optimised with CFD", 2016, Conference Paper;
[6] P. Jamieson, "Innovation in Wind Turbine Design", Wiley & Sons 2011, ISBN 978-0-470-69981-2.

Mechatronic Systems Embedded into the Electrohydraulic Control Equipment for Industrial Applications

Ph.D. Eng. **Radu RĂDOI[1],** Dipl. Eng. **Alexandru HRISTEA[1],** Dipl. Eng. **Bogdan TUDOR[1]**

[1] INOE 2000 – IHP Bucharest, radoi.ihp@fluidas.ro

Abstract: *Electrohydraulic control equipment manufacturers keep pace with technological evolution and incorporate in these devices the latest technologies. Thus it was switched from analogue technology to digital and from analog control signals to the data communication protocols (e.g. Profibus, CANopen). The latest models of such equipment contain microprocessors and sensors and can be parameterized using software provided by manufacturers. Installation and maintenance of such equipment is recommended to be performed by qualified personnel only. These devices are used in various equipment for industries such as mechanical processing, metallurgy, construction, etc.. These allow interfacing with automation systems based on industrial computers or PLCs. The article presents the description and characteristics of such equipment.*

Keywords: *servohydraulics, mechatronic system, proportional control, characteristic curves*

1. Introduction

Modern machinery have reached a very high degree of automation today.These require besides computerized management system some controlpartsfor subsystems or working bodies [1]. The hydraulic control equipment must be reliable,to ensure accuracy, good dynamics and to can be interfaced with the computerizedmanagement system.
A mechatronic system can be defined as a technical system consisting of mechanical components, electronics and intelligent software necessary for control of movements for obtaining certain functions.
The modern hydraulic control equipments incorporates besides the mechanical part, electro mechanical converters, sensors and electronic modules driven with microprocessors.These equipments are manufactured in variants with data communication (e.g Profibus) that can be connected easily with the process computer [2]. Industrial applications include the machine tools with numerical control, injection machines, turbines control and control of precise movement in centers of simulationand for car components tests. Electrical feedback for position of such equipment enables higher gains control loops, improving the dynamics and eliminating errors caused by hysteresis and temperature variations

2. Types of electrohydraulic control equipment which embed mechatronic systems

A classic servovalve for controlling a hydraulic motor is shown in Figure 1. This servovalve with two stages with mechanical feedback is a model produced by Moog. This is a valve whose spool is positioned proportionally to the amount of input electrical signal and internal movement of the spool is accomplished using hydraulic fluid under pressure.This model does not have embedded sensors and electronics.
Block diagram of a mechatronic system of an hydraulic control equipment can be seen in Figure 2.
The modern electro hydraulic control equipment (proportional directional valves or servo valves) have incorporated such a system. They have an actuator that moves a mechanical element which is a distribution spool, a position sensor and an electronic module that processes the signal from the positionsensor based on a control loop with PID controller and allows giving external commands to the device and monitoring of realized value (actual value).

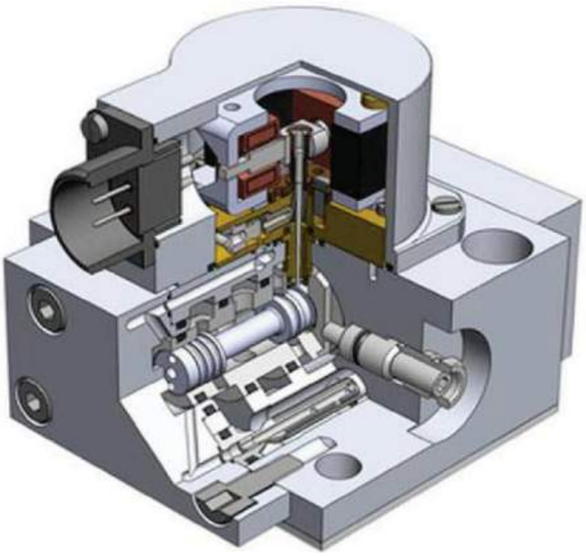


Fig. 1. Servovalve without electronics

Mechanical element is interconnected with the hydraulic subsystem that controls a drive system (e.g. hydraulic motor).

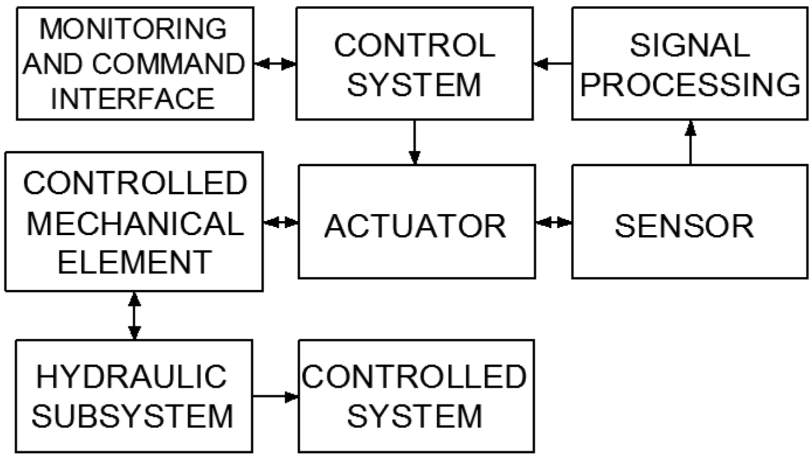


Fig. 2. Block diagram of a hydraulic control equipment

2.1 Proportional directional valve with two internal control loops

In Figure 3 there is a piloted proportional directional valve with two control loops one for control of movement for the pilot spool and one for main spool of valve.After receiving the command signal, the pilot send pressurized oil in one room from the ends of the main spool of the directional valve in the direction P→A or P→B.Stroke of the spool, implicitly flow, depend of the pressure acting on the surface from the end of the spool. Hydraulic pressure produced by the pilot with the proportional solenoid can be achived with a pressure reducing valve or pressure relief valve.
Without command signal the main spool is maintained centered by two coilsprings located at its ends. With the help of an electronic controller the spool position is maintained precisely according to the control signal [3].The electronics ensure good repeatability, accuracy and reduced hysteresis. The directional valve dynamics is influenced by the control system with proportional electromagnet.

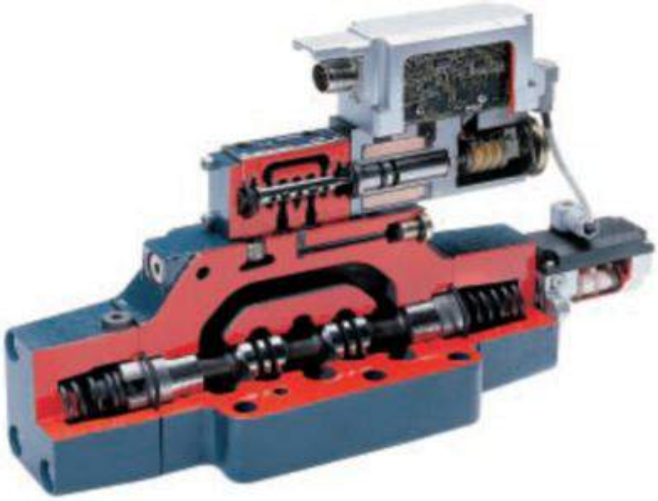


Fig. 3. Proportional directional valve with two LVDT's

2.2 Proportional directional valve with pilot with proportional solenoids

A high response piloted proportional directional valve having proportional pilot with proportional solenoids is the one in Figure 4 [4]. It has a pilot with two proportional solenoids and a loop position control only for the main spool. The transducer for moving the main spool for hydraulic fluid distribution is housed alongside with the circuit board into a case.

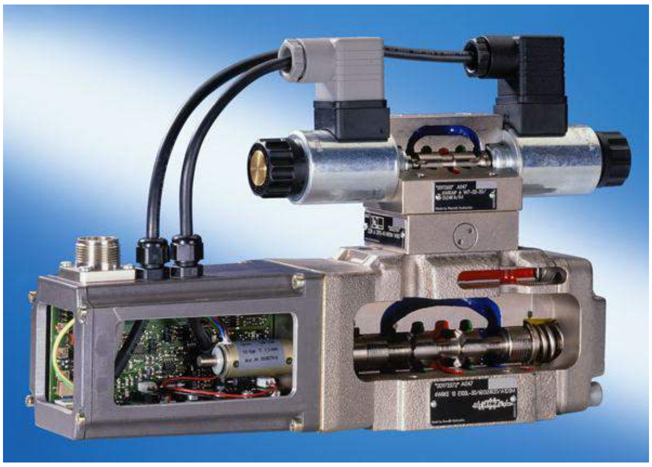
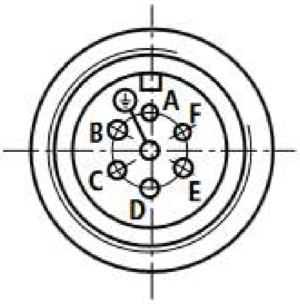


Fig. 4. Piloted proportional directional valve

In Figure 5 can be seen allocation of pins from electric connector of such equipment.



Pin	Allocation
A	24 V
B	0 V
C	Ref. for actual value
D	Command value
E	Ref. for com. value
F	Actual value

Fig. 5. Valve connector and pins

Between the D-E pins of directional valve can give command signal in ±10 V domain and between pins C-F can be read the signal value of the realized position of the spool also in the range of ± 10V.There are versions with 0 ... 10 mA or 4 ... 20 mAsignal, and other with digital communication options EtherCAT, Profinet, etc.

2.3 Proportional directional valve direct commandedwith linear force motor

In Figure 6 there is another type of directional valve with direct drive linear motor [5] with permanent magnets.The actuator with mobile coil has better linearity than proportional solenoid one.
At this directional valve, linear motor is located on one side and the electronics and the stroke transducer on the other side. Centering springs for spool are incorporated in linear force motor with permanent magnet.These directional valves have high dynamics and accuracy and are preferred for accuracy in positioning axes and hydraulic pressure control and speed. These directional valves reach frequency response of real servovalves. Electronics are made with microcontroller and have the possibiliity to be parameterized using a dedicated software.

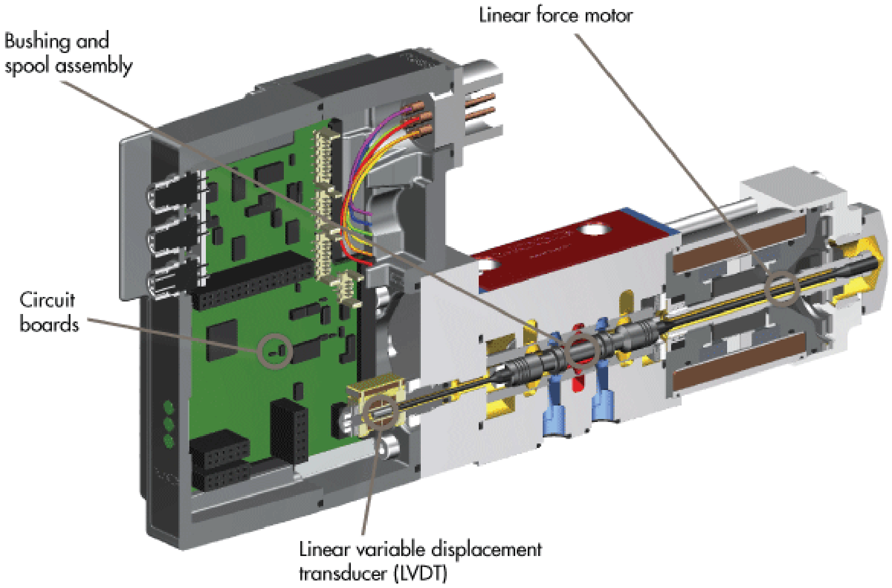


Fig. 6. Proportional directional valve with linear motor with permanent magnets

Table 1: Hysteresis by type of valve

	Direct operated			Two-stage	
Valve actuation	Open loop proportional solenoid	Closed loop Proportional solenoid	Voice coil linear motor	Hydraulic with mechanical feedback	Hydraulic with electrical feedback
Hysteresis	5 %	2 %	0,2 %	2 %	0,2 %

2.4 3 stage servovalve with on board electronics

The Servo valves are built also within 3 stage block. Such a servovalve is found in Figure 7[6]. To achieve a centre of the main spool it uses coil springs located in the side covers. Attached to the right cover there is a box containing the electronic control and transducer of LVDT type for the position of the spool. This servo valve has implemented a "fail safe" function. Between pilot body and main body it lies an on-off valvewith electric command to bring the main tray in a safe position depending on the version at voltage supply cutting. Anotherversion can bring to the center the spool where all hydraulic ports are closed with over laps. Another version can bring A→T after cut off of pilot pressure. These valves are suitable for the position, velocity, pressure and force control.

Fig. 7. 3 Stage servovalve

In fig. 8 can be seen a block diagram of the electronics incorporated in the body of the servo valve.

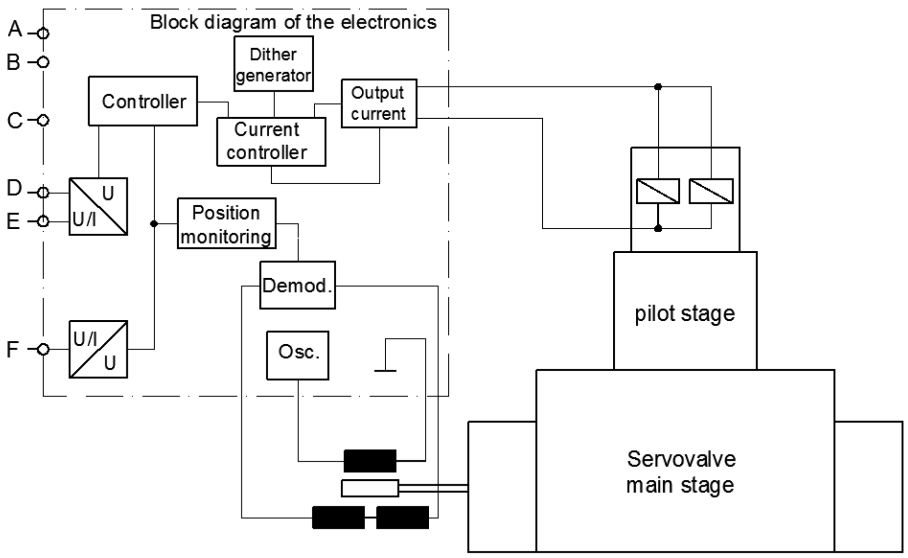
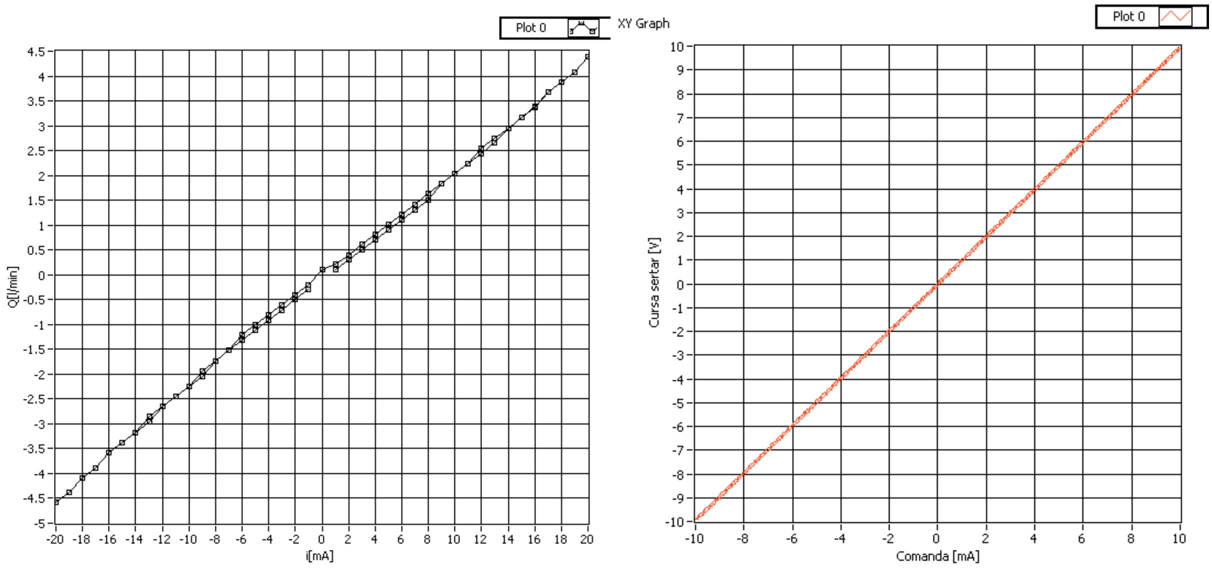


Fig. 8. Block diagram of the electronics of the servo valve

Testing this equipment is made by qualified personnel using specialized test stands. Characteristic curve of such equipment can be drawn in the laboratory using an application developed in LabVIEW. Such an application was used to obtain the characteristics of Figure 9. In figure 9 A was drawn a diagram for a classic servo valve without integrated electronics by recording flow values using a flow transducer. B chart in Figure 9 was obtained by recording the signal "actual value" provided to terminal F of a servo valve with integrated electronics. Comparing the two charts can see the difference between linearity and hysteresis of the two servo valves. The diagram of equipment with internal control loop (B) is linear and has a very low hysteresis.



A B

Fig. 9. Valves characteristic curves

3. Conclusions

Such equipment’s include microcontrollers, sensors and high precision mechanical components. These devices can be used in industrial machinery for regulating the position, pressure, speed and force control.
Compared to classic proportional servo equipment, equipment with electric positioning system allows higher control loops, improving the dynamics and eliminating errors caused by hysteresis.
Maintenance of this equipment must be carried by specialized personnel or specialized companies, otherwise it may exist the risk of irreparable damage.

Acknowledgment

This paper has been developed in INOE 2000-IHP, with financial support of ANCSI, under the national research *Programme NUCLEU,* project title: *Physics of processes for reducing energy losses and developing renewable energy resources by use of high-performance equipment,* project code PN 16 40.03.01, financial agreement no. 5N/2016, Additional act no. 1/2017.

References

[1] T.C. Popescu, A. I. Popescu, “Two operating modes electro-hydraulic servo valve”, Hidraulica No. 4/2014 Magazine of Hydraulics, Pneumatics, Tribology, Ecology, Sensorics, Mechatronics, Bucharest, 2014,ISSN 1453 – 7303, pp. 51-55;
[2] M. Comes, P. Drumea, A. Mirea, G. Matache, “Intelligent servohydraulic device for the control of the motion”, *24th International Spring Seminar on Electronics Technology ISSE 2001*;
[3] V. Balasoiu, I. Bordeasu, “Electrohidraulic Proportional Servo Valves – Electrohidraulic Servo Distributors”, Hidraulica No. 4/2013 Magazine of Hydraulics, Pneumatics, Tribology, Ecology, Sensorics, Mechatronics, Bucharest, 2013,ISSN 1453 – 7303, pp. 37-51;
[4] http://www.germangulf.com/02_hydraulic_products_hydraulic.html;
[5] http://machinedesign.com/motion-control/digital-direct-drive-valves-upgrade-industrial-machinery;
[6] http://www.designworldonline.com/building-a-bridge-for-wind-energy/#_.

ISSN 1453 – 7303 "HIDRAULICA" (No. 1/2017)
Magazine *of Hydraulics, Pneumatics, Tribology, Ecology, Sensorics, Mechatronics*

An Introduction to Hoisting Machines

Sunny NARAYAN[1], Aman GUPTA[2]

[1]Indus University, India, sn2008@rediffmail.com

[2]Indus University, India

Abstract: *A mechanical device or apparatus designed for lifting heavy objects or people is known as hoisting equipment. Structurally it is of diverse nature and may be classified based on the construction and application. In this work a brief introduction to these devices has been presented.*

Keywords: *Hoisting machines*

1. Introduction

Three groups of hoisting equipment's are mainly known as [1-10]:
a. Hoisting Machines: a group of periodic action devices designed as self-lifting gear for hoisting and moving loads.
b. Cranes: a combination of separate hoisting mechanism with a frame structure for lifting and/or moving loads
c. Elevating Equipment: a group of periodic action machine intended for raising loads with guide-ways

A device or apparatus designed for lifting heavy objects or people. Three groups of hoisting equipment are as follows. These equipment items help to transfer loads to cover a definite area. They are non-continuous operation so risk of beak down is high.

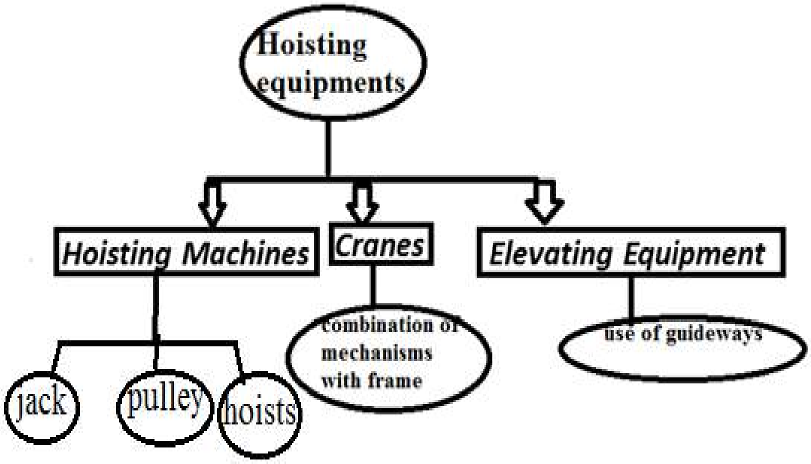


Fig. 1. Types of hoisting equipment

2. Types of hoisting Machinery [11-20]

Jacks-These are directly lifting simple devices suitable for short distance use. Jacks may be categorized as follows:

- Rack and pinion type jack
- Screw jacks
- Lever type jack
- Hydraulic action jack

A) Rack and pinion type jack-They have Significant lifting height as well as ability to operate in vertical and horizontal positions, These have Compact design, excellent ergonomics, ease of operation, High maintainability, Smooth lifting ability and ability to accurately position the load at a certain height. Rotation of handle causes pinion to rotate that in turn causes reciprocating motion of rack (on which load is placed) and hence motion of load.

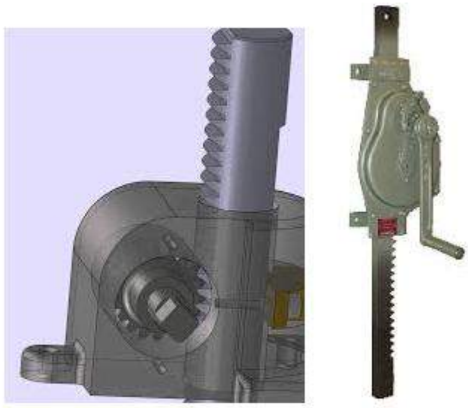


Fig. 2. Rack and pinion jack

B) Screw Jack-This jack has a Spindle and nut mechanism. Rotation of nut causes reciprocating motion of spindle on which load is placed and hence lift of it as seen in next figure. Mechanical advantage of a jack may be written as:

$$\frac{W}{E} = \frac{2\pi L}{P} \qquad (1)$$

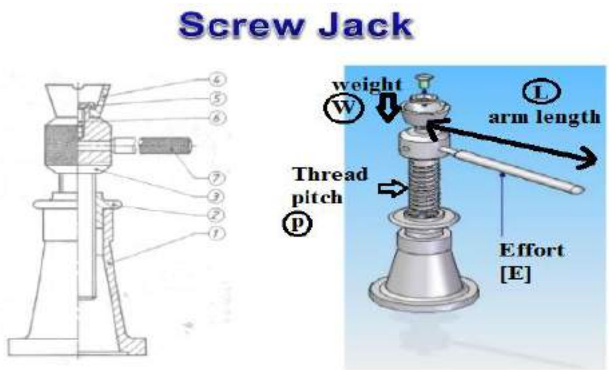


Fig. 3. Screw jack

C) Lever jack

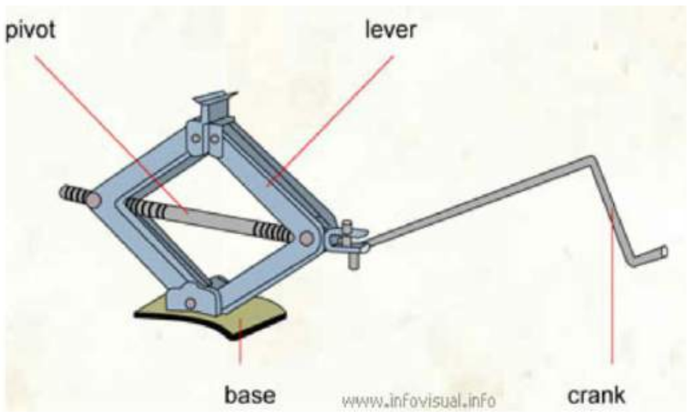


Fig. 4. Lever jack

This jack has following parts:
1. Automobile Jack: a device equipped with a crank that is used to raise an automobile.
Pivot: axis of rotation.
2. Lever: solid movable part attached to a fixed point, used to increase an applied force.
3. Crank: arm perpendicular to an axle, used to create circular motion.
4. Base: foot on which the jack rest.

D) Hydraulic Jack
This type of ram uses hydraulic pressure. A key principle of working with hydraulic systems is that the pressure is the same throughout. The pressure in the motor is the same as the pressure in the pump. Also the hydraulic pressure acts equally in all directions.

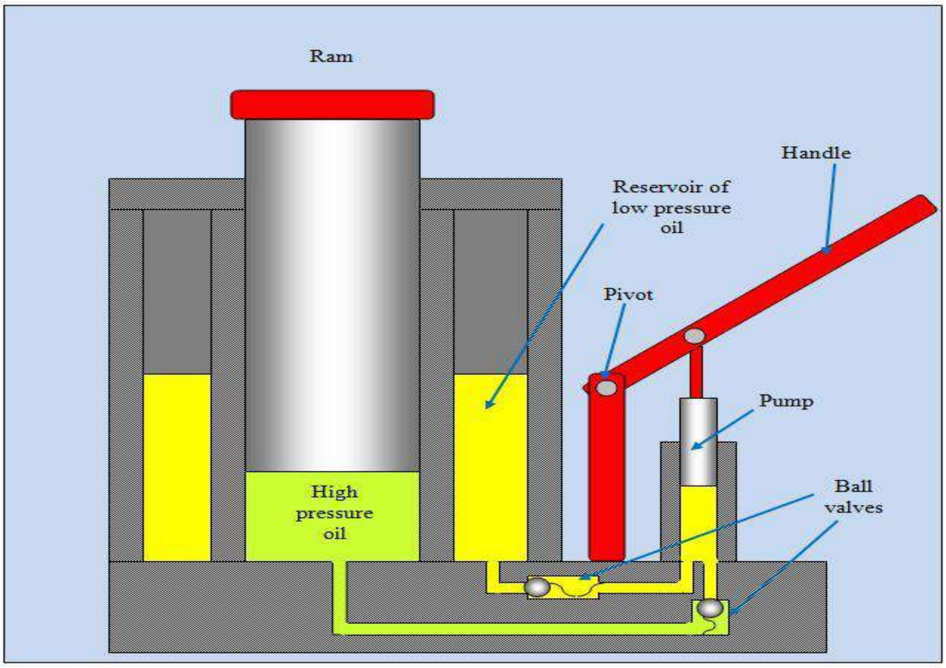


Fig. 5. Hydraulic jack

Hydraulic jack has following parts:
1. plunger cylinder: it is provided with a pivot and handle at the top and fits at the bottom into the bore of jack body filled with fluid.
2. Ram cylinder: feeds the high pressure fluid into the lower portion of the jack body through a system of ports.

Rate of lowering or rising of load is regulating by levels of extra oil present in reservoir. This ram has high Capacity of about 200 ton and efficiency with limited lift height, smaller weight and limited lifting speed.

Application:
Hydraulic system of stackers
Low and high lift trucks
Work levelers
Hydraulic lifts

2. Pulleys [21-24]

A pulley wheel is a mechanism which helps move or lift objects. They have small sheave or wheel with a grooved rim mounted on a pin on which it turns. A frame or block is used to mount sheaves with a flexible rope, cord or chain passing over groove. Three types of pulleys are as follows:

1. Fixed
2. Movable
3. A block and tackle

a) Fixed pulley: - fixed with fulcrum

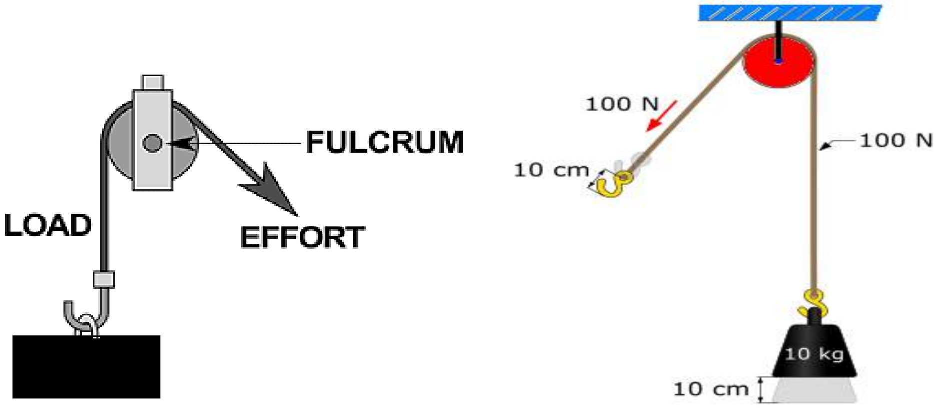


Fig. 6. Fixed pulley

In this simple pulley system, the force is equal to the load, so the
Mechanical Advantage is 1:1 or 1.
The Mechanical Advantage is calculated like so:
Mechanical Advantage = Load / Effort = 100 N / 100 N
Mechanical Advantage = 1:1 or 1

b) Moveable pulleys - rises and falls with the load being moved. Each side of the rope carries half the load. Therefore, the force required by the person to keep the load in equilibrium is also half the load. This system has a Mechanical Advantage of 2:1 or 2.
The Mechanical Advantage is calculated like so:
Mechanical Advantage = Load / Effort = 100 N / 50 N
Mechanical Advantage = 2:1 or 2

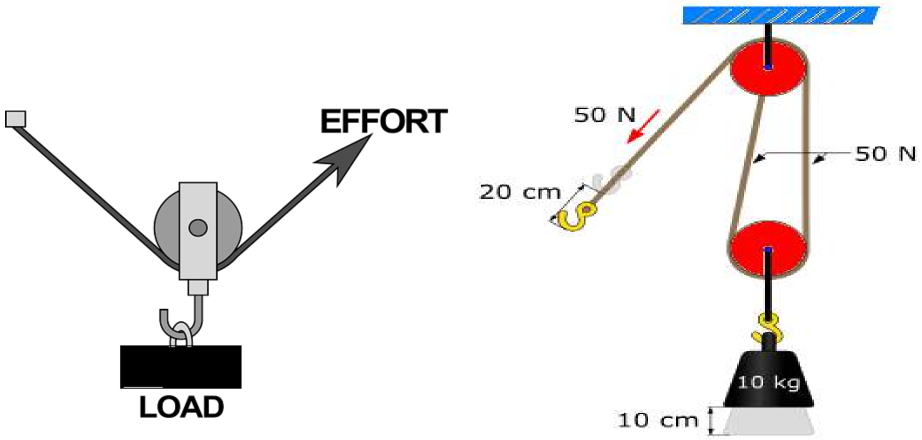


Fig. 7. Movable pulley

c) Block and tackles - Consists of two or more pulleys (fixed and moveable).

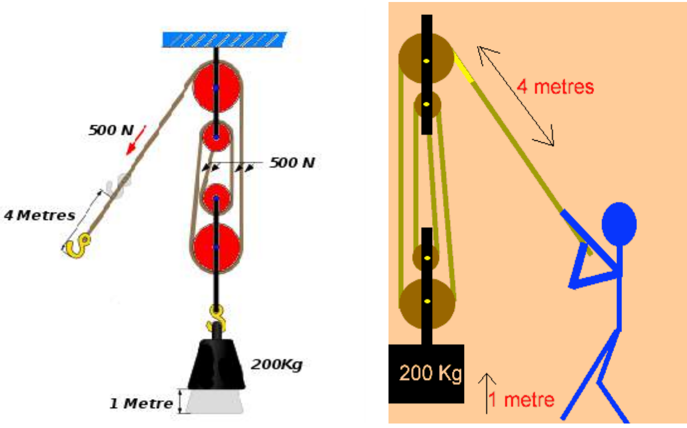


Fig. 8. Block and tackle pulley

This system has a Mechanical Advantage of 4:1 or 4
The Mechanical Advantage is calculated like so:
Mechanical Advantage = Load / Effort = 100 N / 25 N
Mechanical Advantage = 4:1 or 4

The Mechanical Advantage is calculated like so:
Mechanical Advantage = Load / Effort = 2000 N / 500 N
Mechanical Advantage = 4:1 or 4

Multiple Pulley drives can be used with belts, as seen in figure no 9.

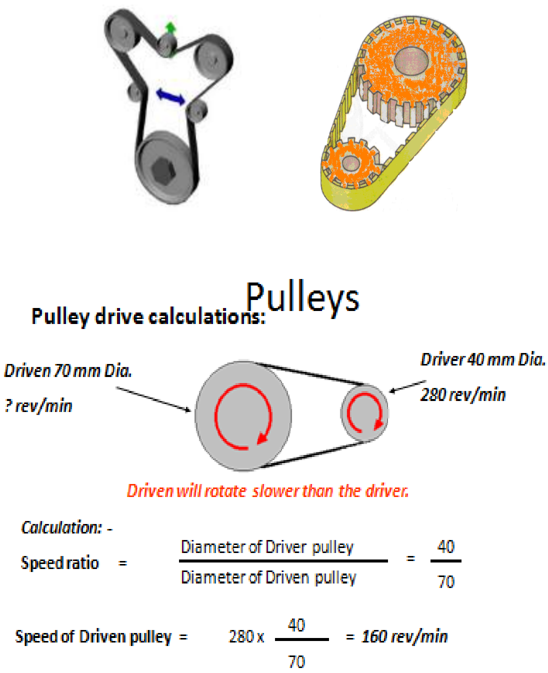
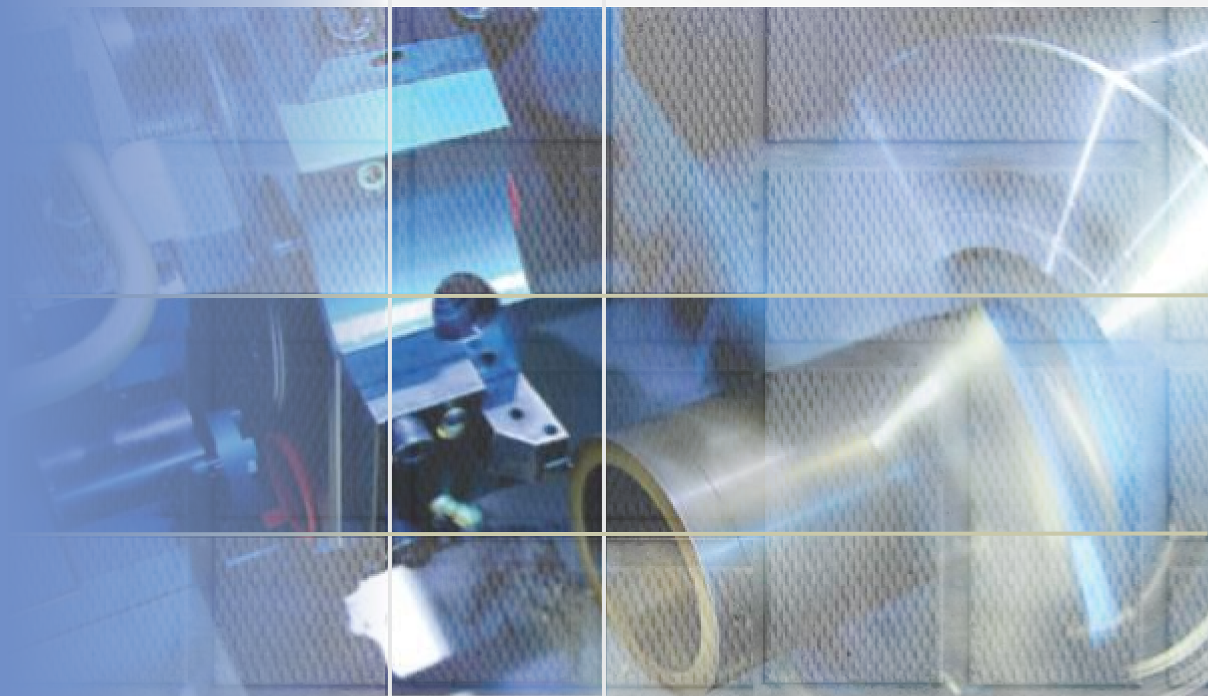


Fig. 9.

References

[1] C. Tung Simon, M.l L. McMillan, “Automotive tribology overview of current advances and challenges for the future”, Tribology International, 2004, 37:517–35;
[2] D.F. Li, S.M. Rhode, H.A. Ezzat, “An automotive piston lubrication model’, ASLE Transactions 1983, 26(2):151–60;

[3] V.W. Wong, T. Tian, H. Lang, J.P. Ryan, Y. Sekiya, Y. Kobayashi, et al., "A numerical model of piston secondary motion and piston slap in partially flooded elasto hydrodynamic skirt lubrication", SAE paper no. 940696, 1994;
[4] S. Jiang, J. Cho, "Effects of skirt profiles on the piston secondary movements by the lubrication behaviors", International Journal of Automotive Technology, 2004, 5(1):23–31;
[5] A. Gupta, S. Narayan, "Electrical Analogy of Liquid Piston Stirling Engines, Hidraulica Magazine no. 2 (2016): 58;
[6] S. Narayan, V. Gupta, "Overview of working of Stirling engines", Journal of Engineering Studies and Research, 21.4 (2015): 45;
[7] A. Gupta, S. Narayan, "A Review of Heat Engines", Hidraulica Magazine no. 1 (2016): 67;
[8] S. Narayan, "Analysis of noise emitted from diesel engines", Journal of Physics: Conference Series, Vol. 662, No. 1, IOP Publishing, 2015;
[9] A. Gupta, S. Narayan, "Effects of turbo charging of spark ignition engines", Hidraulica Magazine no. 4 (2015): 62;
[10] S. Narayan, "Designing of liquid piston fluidyne engines", Hidraulica Magazine no. 2 (2015): 18;
[11] S. Narayan, "Effects of Various Parameters on Piston Secondary Motion", No. 2015-01-0079, SAE Technical Paper, 2015;
[12] S. Narayan, "Analysis of Noise Radiated from Common Rail Diesel Engine", Tehnički glasnik 8.3 (2014): 210-213;
[13] S. Narayan, "Time-frequency analysis of Diesel engine noise", Acta Technica Corviniensis-Bulletin of Engineering 7.3 (2014): 133;
[14] S. Narayan, "Wavelet Analysis of Diesel Engine Noise", Journal of Engineering and Applied Sciences 8.8 (2013): 255-259;
[15] S. Narayan, V. Gupta, "Motion analysis of liquid piston engines", Journal of Engineering Studies and Research 21.2 (2015): 71;
[16] S. Narayan, "Modeling of Noise Radiated from Engines", No. 2015-01-0107, SAE Technical Paper, 2015;
[17] S. Narayan, A. Gupta, R. Rana, "Performance analysis of liquid piston fluidyne systems", Mechanical Testing and Diagnosis 5.2 (2015): 12;
[18] V. Gupta, S. Sharma, S. Narayan, "Review of working of Stirling engines", Acta Technica Corviniensis-Bulletin of Engineering 9.1 (2016): 55;
[19] A. Singh, S. Bharadwaj, S. Narayan, "Review of how aero engines work", Tehnički glasnik 9.4 (2015): 381-387;
[20] S. Narayan, "A review of diesel engine acoustics", FME Transactions 42.2 (2014): 150-154;
[21] S. Narayan, "Noise Optimization in Diesel Engines", Journal of Engineering Science and Technology Review 7.1 (2014): 37-40;
[22] A. Singh, S. Bharadwaj, S. Narayan, "Prikaz rada motora zrakoplova", Tehnički glasnik 9.4 (2015): 381-387;
[23] X. Meng, X. Youbai, "A new numerical analysis for piston skirt–liner system lubrication considering the effects of connecting rod inertia", Tribology International, 47 (2012): 235-243.



http://hidraulica.fluidas.ro