No.1-2 / June 2011
ISSN 1453 - 7303

# Hidraulica

Magazine of
HYDRAULICS, PNEUMATICS, TRIBOLOGY, ECOLOGY, SENZORICS, MECHATRONICS

## CONTENTS






Published by:
**Hydraulics & Pneumatics Research Institute INOE 2000 - IHP Bucharest-Romania**
Address:
14 Cuțitul de Argint,
district 4, Bucharest, cod 040557
ROMANIA
Phone: +40 21 336 39 90; +40 21 336 39 91
Fax:+40 21 337 30 40
E-mail: ihp@fluidas.ro
Web: www.ihp.ro
Web: www.ihp.ro/hidraulica
with support of:
**National Professional Association of Hydraulics and Pneumatics in Romania - FLUIDAS**
E-mail: fluidas@fluidas.ro
Web: www.fluidas.ro

## CONDITIONS FOR SUBMISSION OF PAPERS FOR "HIDRAULICA" MAGAZINE

1. Page format is A4 (210 x 297)
2. The text of an article, drawings included, shall not exceed 8 conventional pages.
3. The article shall be draft in two columns, using MS Office Word 97 to 2007, preferably with fonts Arial CE or Times Roman R, font size 12, at 1 (one) line in page; diacritical marks shall be used. Also, formulas shall be written in Arial CE too; Microsoft Equation is recommended.
4. Since no. 1 / 2008 issue of "HIDRAULICA" articles submitted for publishing shall be written in English.
5. Together with article shall be presented also an abstract of the paper in 10 rows and keywords; abstract and keywords shall be written in Romanian and English.
6. All figures and tables included in articles must be mentioned in text; figures and captions must have titles.
7. When authors belong to several institutions, each author is marked with a number of asterisks. Each number of asterisks will be associated with an institution. Caption of asterisks will be presented at bottom of page. For the first author or corresponding author is an e-mail address shall be provided.
8. Tables and figures will be presented in the following formats: .bmp, .jpeg, .tiff, .eps, .ai, .cdr;
9. Papers must be original contributions of the author, not remakes or translations of other papers.
10. Authors retain full responsibility for the accuracy of statements expressed, for calculations and results displayed.
11. Not allowed for publishing papers that were published in full or in part in other publications.
12. Review committee reserves the right to make recommendations for improving the articles or to not publish material regarded as inappropriate. Unpublished manuscripts are not returned.
13. Magazine does not grant copyrights.
14. Paper intended to be published shall be submitted via one of the e-mail addresses : hidraulica@fluidas.ro or fluidas@fluidas.ro.

EDITORIAL

# INNOVATION

Ph.D.Eng. **Petrin DRUMEA**
MANAGER
Hydraulics & Pneumatics Research Institute

The notion of innovation is included in recent years in all sorts of speeches, from the technical-scientific to economic and hence to political ones. Everyone agrees that without innovation there can be no discussion about exiting the crisis or recession, but when it is time to move on to facts some problems occur. Next we will discuss only about innovation in technology and only about innovative products and new innovative technologies. The first problem which I consider very important is who are the decision makers or those who consider themselves experts on this issue of national interest. Unpleasant and even dangerous is that the number of those who have never participated in creating products and technologies is too high and they are generally very aggressive in their insufficiency. The fact that these people have obtained some titles in other areas and for other activities does not help them at all to understand the phenomenon of innovation and make the best decisions. Perhaps it would be necessary in all committees and councils in the area to be included in the first place the specialists and only in second or third place those with claims that can not be supported by tangible achievements. The second question is for whom innovation is discussed. Interestingly, in this case everyone agrees that the beneficiary should be the Romanian economy especially SMEs, given the scarcity of large companies that have as their main activity manufacturing of industrial products. Of course that in the group of products can actually be included also the new technologies designed in our country at equipment level. If we accept that innovation is primarily for SMEs, then it should be examined seriously what they produce, how they produce and at what technical and technological level. Innovation, in my opinion, should address most of these companies and not only a few that work in top areas in which our access to innovation is limited.

The third problem is what kind of products should be offered to those companies considering the technology available, the staff they have and clearly also the financial capacity of each. I do not think there are of interest, as innovative products, the large scale ones, such as airplanes, electric cars, reactors for nuclear power plants, etc. because usually only technical novelties are not enough for them to be imposed on the international market. I think that mechanical, electrical, hydraulic, mechatronic equipment and subassemblies, those for green energetics, agriculture and many more can be topics to approach in Romania in action of modernization through innovation. A final issue that creates difficulties for innovation action is the strange idea of imposing weird conditions to specialists who have ideas, often patentable, and want to participate in national programmes in the field. These conditions related to specialists and products occurred precisely because of wrong selection of decision makers in this regard. And with all these problems and probably many more I still hope that innovation will develop intensively and will influence, in our country as well, the economy and especially industrial production.

EDITORIAL

Dr. ing. **Petrin DRUMEA**
DIRECTOR
Institutul de Cercetari pentru Hidraulica si Pneumatica

## INOVAREA

Notiunea de inovare este inclusa in ultimii ani in tot felul de discursuri, de la cele tehnico-stiintifice la cele economice si de aici la cele politice. Toata lumea este de acord ca fara inovare nu se poate vorbi de iesirea din criza sau din recesiune, insa cand este timpul sa se treaca la fapte apar cateva probleme. In continuare vom discuta numai despre inovarea in tehnica si doar despre produse inovative si tehnologiile noi, inovative. Prima problema pe care o consider extrem de importanta este cine sunt cei care iau decizii sau fac pe expertii in aceasta problema de interes national. Neplacut si chiar periculos este ca numarul celor care nu au participat niciodata la crearea de produse sau tehnologii noi este mult prea mare si in general sunt extrem de agresivi in insuficienta lor. Faptul ca acesti oameni detin niste titluri obtinute in alte domenii si pentru alte activitati nu ii ajuta cu nimic sa inteleaga fenomenul inovarii si sa poate lua deciziile cele mai bune. Probabil ca ar fi necesar ca in toate comitetele si consiliile din domeniu sa fie inclusi in primul rand specialistii si abia in al doilea sau al treilea rand cei cu pretentii care nu pot fi sustinute prin realizari concrete. A doua problema este pentru cine se discuta inovarea. Este interesant ca si in aceasta situatie toata lumea este de acord ca trebuie ca beneficiarul sa fie economia romaneasca cu precadere IMM-urile, data fiind putinatatea marilor firme care au ca activitate de baza fabricarea de produse industriale. Sigur ca in grupa produselor pot fi incadrate de fapt si tehnologiile noi, concepute in tara la nivel de echipamente. Daca se admite ca inovarea se adreseaza in primul rand IMM-urilor, ar trebui analizat la modul serios ce produc, cum produc si la ce nivel tehnic si tehnologic o fac. Inovarea, din punctul meu de vedere, trebuie sa se adreseze majoritatii acestor firme si nu catorva, care lucreza in domenii de varf, domenii in care accesul nostru la inovare este limitat.

A treia problema este ce fel de produse trebuie sa fie oferite acestor firme tinand cont de tehnologiile de care dispun, de personalul pe care il au si clar si de capacitatea financiara a fiecaruia. Nu cred ca sunt de interes general, ca produse inovative, cele de mare anvergura precum avioanele, automobilele electrice, reactoarele pentru centrale nuclear-electrice, etc. intrucat de obicei nu sunt suficiente doar noutatile tehnice pentru a fi impuse pe piata internationala. Consider ca echipamentele, subansamblele mecanice, electrice, hidraulice, mecatronice, pentru enegetica verde, pentru agricultura si multe altele pot fi subiecte abordabile in Romania in actiunea de modernizare prin inovare. O ultima problema care creaza dificultati actiunii de inovare o constituie ideea ciudata a impunerii unor conditii aiurea specialistilor care au idei, de cele mai multe ori brevetabile si care doresc sa participe la programele nationale pe domeniu. Aceste conditii referitoare la specialisti si la produse au aparut tocmai ca urmare a selectarii gresite a celor care iau decizii in aceasta privinta. Si cu toate aceste probleme si probabil multe altele eu sper ca inovarea se va dezvolta intens si va influenta si in tara noastra economia si mai ales productia industriala.

UNIUNEA EUROPEANĂ

GUVERNUL ROMÂNIEI MINISTERUL MUNCII, FAMILIEI ŞI PROTECŢIEI SOCIALE AMPOSDRU

Fondul Social European POSDRU 2007-2013

Instrumente Structurale 2007-2013

GUVERNUL ROMÂNIEI MINISTERUL MUNCII, FAMILIEI ŞI PROTECŢIEI SOCIALE OIRPOSDRU S-V OLTENIA

Camera de Comerţ şi Industrie Vâlcea

# "TRAINING OF SPECIALISTS IN THE FIELDS OF MECHANICS, HYDRAULICS AND PNEUMATICS IN ORDER TO PROMOTE ADAPTABILITY AND INCREASE COMPETITIVENESS" POSDRU/81/3.2/S/47649

The project "Training of specialists in the fields of mechanics, hydraulics and pneumatics in order to promote adaptability and increase competitiveness" is co financed from The Social European Fund, by means of the Operational Sectorial Program Human Resources Development 2007-2013, Prioritary axis 3 "Increase adaptability of the workers and enterprises", Main field of intervention 3.2. "Training and support for enterprises and employees in order to promote adaptability".

***Applicant:*** Chamber of Commerce and Industry Valcea, Romania
implement the project together with partners:
***Partner 1:*** National Professional Association of Hydraulics and Pneumatics - FLUIDAS,Romania
***Partner 2:*** Technical University of Cluj-Napoca, Romania
***Partner 3:*** Technical University "Gheorghe Asachi" of Iasi, Romania
***Partner 4:*** PIA e.V. – Development and Assessment Institute in Waste Water Technology at RWTH - Aachen University – Germania

The project originated in the requirement for professional training of the workers who are charged with the maintenance and repairing of the hydraulic and pneumatic equipment. By the instrumentality of this project it is aimed to specialize 460 workers from mechanics in the fields of Fluid Power drives.

## 1.1 The objective and content of the project

The project was proposed in order to upgrade the professional level of the workers from the Fluid Power field. It has been observed lately that it occurs an undesirable situation in which the workers from the related fields are no more capable of ensuring the maintenance and repairing of the imported equipment.

The incompetence in the field is due to the inadequate professional training of the mechanical locksmit who have problems in using the measurement and control apparata, in adjusting components or in other operations related to fine mechanics but who despite this fact are appertained tasks of maintenance and repairing the hydraulic systems which in normal conditions may be solved only by experts in the field.

It is ascertained in EU that the level of professional training provided by school does not say much about a person's capability of applying into practice all or at least some of the acquired theoretical knowledge. Starting from this situation the consortium which is in charge for developing the project has structured activity on the following directions:

- Completion training courses for the mechanic field for updating the professional training of the mechanical locksmiths
- Completion training courses for hydraulics at a basic level contiguous to the 1st CETOP level
- Completion training courses for managers dealing with the management of environment

- Completion training courses for inspector specialized in labor safety and protection
- Establishment of some proficiency training centers in the field of Fluid Power which to ensure the training of workers according to the European standards
- Activities of counseling and guidance by means of which the employees to be informed about the purpose, utility, level and methodology of professional proficiency training
- Activities of presentation of the field, of the national and international achievements, of national and international exchanges on themes related to Fluid Power and the professional training in the field, which to allow the participants at the project to be connected on European level.

The project should represent a first step towards establishing on national level of a professional proficiency training at European standards.

**1.2. The basis for implementing the project**

The project is very important for the national economy and for this reason was launched only when were met the main required conditions for its proper development. The project basis are the following:

-the requisite of providing completion training for the workers in the field, due to the wide spread of hydro pneumatic drives on all mobile equipment and most of the steady ones and due to the modernization of the technical and scientific level of the equipment and systems.

-the existence of some available centers equipped with laboratories and with personnel capable of performing these training activities.

-the alignment of the professional training of the workers from Romania from the Fluid Power field at the EU standards and requirements it is mandatory.

-the undesirable situation which appeared cause of the abandonment of the medium and long term professional training for the professions involved in the project which led to a dramatic decrease of the professional level of the workers from the related field

-the real situation existent in Romania which imposes a subsequent training in the field of hydraulics after a preliminary requisite training in the field of fine mechanics.

-the possibility of using European funds for starting a long term program of professional training in the field of Fluid Power.

-a good international collaboration of the national professional association with the other professional associations from Europe.

-the inclusion in the consortium of a reliable partner from Germany which has a good acquaintanceship with the completion training and has good connections in Germany with the specialists from the field of Fluid Power.

**1.3. The requisite for improving professional training in the fields approached by the project**

In the last years it has been noticed an increasingly deficit of the specialized workers concurrent with the release on the market of upgraded hydro pneumatic equipment imported directly or of certain components of complex installations. The result is that after serial failures in maintenance and repairing caused by pseudospecialists, the owners of these equipment and installations were forced to call on foreign specialists. The expenses became that high that led to an unwanted artificial increase of the products and services provided by these equipment and installations. Practice has proven that our workers encounter difficulties not only in hydropneumatics but also in fine mechanics and mechano hydraulic assemblies.The workers and their managers have not yet understood the importance of applying ecologic principles to their activities and of certain operational and maintenance technologies which to not affect the environment by any means. All these planned activities have to be developed carefully taking into account labor safety, due to the novelty of the products and the imminent dangers involved.

Another reason which makes this project a requisite is that in Romania have never existed apprenticeship schools for the workers from this large field of activity and the so called specialists are qualified at the working place.

## 2. The general structure of the courses

The basic subject of the project consists of the completion professional training of the workers from industry who operate in the field of Fluid Power drives. In the notion of Fluid Power are included for this project the chapters referring at pneumatics and centralized lubrication. The project will provide information to the trainees about the relation between hydraulics and mechatronics for allowing them a proffesional approach of the equipment and systems presented as mechatronic.

a. The main idea is that a complete course for the trainees specializing in hydraulics comprises two parts.

- The first part is that which provides the completion training of the trainees from the mechanical field and will be finalized with a CNFPA certificate acknowledged internationally. This course which represents the first part of the entire course will have 120 hours from which 60 hours will be of practice in the workshops. Practice may be made separately or intercalated with the theory and may be made at the proficiency training center or at the headquarters of a company which is interested in training at least more than 50% of the total number of trainees from the group.

-The second part of the course provides the training of the trainees in the field of Hydraulics and will be finalized with a graduation diploma issued under the coordination of the professional association FLUIDAS which is a CETOP member. This course which represents the second part of the entire course will comprise 120 hours 60 of theory and 60 of practice. All the courses will take place at the regional training center..

The courses of labor safety and management of the environment will comprise 80 hours which will take place at the regional training center and will be finalized by CNFPA graduation certicates

For further details please do contact us at:

FLUIDAS Association
Address: 14, Cutitul de argint street, district 4, Bucharest, P.O. 040557
Tel. 021.336.39.91, Fax. 021. 337.30.40,
Tel/Fax: 0314052185

Contact person:
PhD. eng. Gabriela MATACHE,
Area Project Manager
E-mail: fluidas@fluidas.ro
Web: www.fluidas.ro

UNIUNEA EUROPEANĂ

GUVERNUL ROMÂNIEI MINISTERUL MUNCII, FAMILIEI ȘI PROTECȚIEI SOCIALE AMPOSDRU

Fondul Social European POSDRU 2007-2013

Instrumente Structurale 2007-2013

GUVERNUL ROMÂNIEI MINISTERUL MUNCII, FAMILIEI ȘI PROTECȚIEI SOCIALE OIRPOSDRU S-V OLTENIA

Camera de Comerț și Industrie Vâlcea

# "PREGATIREA SPECIALISTILOR IN DOMENIILE MECANICII, HIDRAULICII SI PNEUMATICII IN SCOPUL PROMOVARII ADAPTABILITATII SI CRESTERII COMPETITIVITATII" POSDRU/81/3.2/S/47649

Proiectul "Pregatirea specialistilor in domeniile mecanicii, hidraulicii si pneumaticii in scopul promovarii adaptabilitatii si cresterii competitivitatii" este co-finantat din Fondul Social European, prin Programul Operational Sectorial Dezvoltarea Resurselor Umane 2007-2013, Axa prioritara 3 "Cresterea adaptabilitatii lucratorilor si a intreprinderilor", Domeniul major de interventie 3.2. "Formare si sprijin pentru intreprinderi si angajati pentru promovarea adaptabilitatii".

***Solicitant:*** Camera de Comert si Industrie Valcea
implementeaza proiectul alaturi de partenerii:
***Partener 1:*** Asociatia Profesionala de Hidraulica si Pneumatica din Romania – Fluidas
***Partener 2:*** Universitatea Tehnica din Cluj-Napoca
***Partener 3:*** Universitatea Tehnica "Gheorghe Asachi" din Iasi
***Partener 4:*** PIA e.V. – Development and Assessment Institute in Waste Water Technology

Proiectul a pornit de la necesitatea pregatirii profesionale a lucratorilor care se ocupa cu intretinerea si repararea echipamentelor hidraulice si pneumatice. Prin acest proiect se doreste o specializare a 460 de lucratori din domeniul mecanic in domeniul actionarilor hidraulice si pneumatice

**1.1 Scopul si continutul proiectului**

Proiectul a fost propus pentru ridicarea nivelului profesional al lucratorilor din domeniul actionarilor hidraulice si pneumatice. S-a constatat si se constata in fiecare zi ca s-a ajuns in situatia nefavorabila in care lucratorii nostrii din domeniu nu mai pot nici macar intretine si repara echipamentele si utilajele importate. Lipsa de pregatire din domeniu se datoreaza si slabei pregatiri profesionale a lacatusilor mecanici care au probleme in utilizarea aparaturii de masura si control, in ajustarea reperelor sau in alte activitati de mecanica fina si care cu toate acestea primesc sarcini de intretinere si reparatii ale sistemelor hidraulice care in mod normal pot fi rezolvate doar de specialisti.

Se admite, la nivel european, ca nivelul de pregatire profesionala asigurat de scoala nu spune nimic despre capacitatea unei persoane de a aplica in practica toate sau macar o parte din cunostintele teoretice acumulate. Pornind de la aceasta situatie, consortiul care se ocupa de proiect a structurat activitatea pe urmatoarele directii:

-Cursuri de perfectionare in domeniul mecanic pentru aducerea la zi a pregatirii profesionale a lacatusilor mecanici

-Cursuri de perfectionare in domeniul hidraulicii la un nivel de baza apropiat nivelului 1 CETOP.

-Cursuri de perfectionare manager al sistemelor de management de mediu

-Cursuri de perfectionare pentru inspector specialitate protectia muncii

-Crearea unor centre de perfectionare in domeniu sistemelor de actionare mecanice hidraulice si pneumatice capabile sa aduca pregatirea lucratorilor la nivel european.

-Activitati de consiliere si orientare prin care personalul lucrator al firmelor sa fie informat despre scopul, utilitatea, nivelul si metodologia de perfectionare profesionala.

-Activitati de prezentare a domeniului, de prezentare a realizarilor nationale si internationale, de realizare a unor schimburi nationale si internationale pe teme legate de domeniul actionarilor hidropneumatice si al pregatirii profesionale, care sa permita participantilor la proiect sa fie conectati la nivelul european.

Proiectul trebuie sa reprezinte un prim pas in stabilirea la nivel national a unui sistem de perfectionare profesionala la nivel european.

**1.2. Bazele realizarii proiectului**

Proiectul este foarte important pentru economia nationala si ca atare a putut fi lansat abia cand au fost atinse principalele conditii pentru desfasurarea acestuia. Bazele proiectului sunt urmatoarele:

-Necesitatea perfectionarii profesionale a lucratorilor din domeniu, pornind de la larga raspandire a actionarilor hidropneumatice pe toate utilajele mobile si pe majoritatea celor fixe si de la cresterea nivelului tehnico-stiintific al echipamentelor si sistemelor.

-Existenta unor centre dotate cu personal si laboratoare capabile sa desfasoare o astfel de perfectionare.

-Obligativitatea alinierii pregatirii profesionale a lucratorilor din tara in domeniul hidraulicii la nivelul cerintelor europene

-Situatia neplacuta la care s-a ajuns prin renuntarea la pregatirea profesionala de medie sau lunga durata pentru meseriile implicate in proiect si care au condus la o scadere dramatica a nivelului profesional al lucratorilor din domeniu.

-Situatia reala existenta in tara care face ca sa nu se poata trece la o perfectionare in domeniul hidraulicii fara o perfectionare prealabila in domeniul mecanicii fine.

-Posibilitatea utilizarii unor fonduri europene pentru inceperea unui program pe lunga durata de perfectionare profesionala in domeniul hidraulicii si pneumaticii.

-Existenta unei bune colaborari internationale a asociatiei profesionale cu celelalte asociatii din Europa.

-Includerea in consortiu a unui partener din Germania care are o buna cunoastere a sistemului de perfectionare si foarte bune legaturi in Germania cu specialistii care se ocupa de hidraulica.

**1.3. Necesitatea perfectionarii pregatirii profesionale in domeniile analizate de proiect**

In ultimii ani s-a constatat o crestere a deficitului de lucratori specializati simultan cu cresterea nivelului tehnic al echipamentelor hidropneumatice importate direct sau in componenta unor utilaje complexe. Urmarea este ca de cele mai multe ori dupa incercari esuate de mentenanta si reparatii cu diversi pseudospecialisti, posesorii acestor utilaje si echipamente au fost obligati sa apeleze la specialistii straini. Cheltuielile cu acestia au devenit foarte mari conducand la o crestere artificiala si nedorita a produselor si serviciilor realizate cu aceste utilaje si echipamente. Practica a dovedit ca lucratorii nostri au probleme nu doar cu hidropneumatica ci si cu mecanica fina si cu ansamblurile mecano-hidraulice.

De asemenea lucratorii precum si conducatorii acestora inca nu au inteles importanta ecologizarii activitatii si necesitatea aplicarii unor tehnologii de lucru si de mentenanta care sa permita si sa asigure evitarea impactului negativ asupra mediului. Toate aceste actiuni preconizate trebuie sa se desfasoare cu o mare atentie privind protectia muncii, date fiind noutatile tehnice si pericolele implicate asupra personalului de intretinere.

Un alt motiv pentru care e necesar acest proiect este acela ca la nivel national nu a existat si nu exista scoli pentru lucratorii din acest mare camp de activitate si ca urmare asa zisii specialisti sunt proveniti din personal calificat in domeniu la locul de munca.

## 2 Structura generala a cursurilor din proiect

Subiectul de baza al proiectului il costituie perfectionarea pregatirii profesionale a lucratorilor din industrie care au preocupari in domeniul actionarilor hidraulice. In notiunea de hidraulica sunt incluse pentru acest proiect si capitolele de pneumatica si de ungere centralizata. De asemenea proiectul va preciza cursantilor care este relatia dintre hidraulica si mecatronica pentru a le permite o abordare profesionala a echipamentelor si sistemelor care sunt prezentate sub denumirea de mecatronice.

a. Idea de baza care trebuie retinuta este ca un **curs complet** pentru cei care se specializeaza in hidraulica este alcatuit din doua parti.

- **Prima parte este cea care face perfectionarea cursantilor in domeniul mecanic** si se va finaliza cu un certificat CNFPA recunoscut la nivel national. Acest curs care constituie prima parte a cursului complet va dura 120 de ore din care 60 de ore vor fi de practica in atelier. Practica poate fi facuta intercalat cu teoria sau total separat si poate fi facuta la sediul centrului de perfectionare sau la sediul unei firme care are la pregatire un numar de peste jumatate din membri grupei.

-A **doua parte a cursului face perfectionarea cursantilor in domeniul hidraulic** si se va finaliza cu o diploma de absolvire emisa sub coordonarea asociatiei profesionale FLUIDAS care este membra a Asociatiei europene CETOP. Acest curs care constituie a doua parte a cursului complet va dura tot 120 de ore si la fel ca si la cursul de mecanica vor fi 60 de ore de teorie si 60 de ore de practica. Toate orele se vor desfasura, in principiu, la sediul centrului de perfectionare regional.

Cursurile pentru protectia muncii si managementul de mediu vor avea o durata de 80 de ore care se vor desfasura la sediul centrului regional de perfectionare si se vor finaliza cu certificate de absolvire CNFPA

Pentru detalii suplimentare va rugam sa ne contactati la:

Asociatia FLUIDAS
Adresa: Strada Cutitul de Argint, nr. 14, sector 4, Bucuresti, 040557
Tel. 021.336.39.91, Fax. 021. 337.30.40,
Tel/Fax: 0314052185

Persoana de contact:
Dr. ing. Gabriela MATACHE,
Director Zonal Proiect
E-mail: fluidas@fluidas.ro
Web: www.fluidas.ro

UNIUNEA EUROPEANĂ

GUVERNUL ROMÂNIEI
MINISTERUL MUNCII, FAMILIEI
ŞI PROTECŢIEI SOCIALE
AMPOSDRU

Fondul Social European
POSDRU 2007-2013

Instrumente Structurale
2007-2013

PROIECTUL ESTE FINANTAT DIN FONDUL SOCIAL EUROPEAN PRIN
PROGRAMUL OPERAŢIONAL SECTORIAL PENTRU DEZVOLTAREA RESURSELOR UMANE 2007 – 2013
“CREŞTEREA ADAPTABILITĂŢII LUCRĂTORILOR ŞI A INTREPRINDERILOR”

# PREGATIREA SPECIALISTILOR IN DOMENIILE MECANICII, HIDRAULICII SI PNEUMATICII IN SCOPUL PROMOVARII ADAPTABILITATII SI CRESTERII COMPETITIVITATII

Camera de Comert si Industrie Valcea
Romania

Asociatia Profesionala de Hidraulica si Pneumatica
Romania

Development and Assessment Institute in Waste Water Technology
Aachen - Germania

Universitatea Tehnica Cluj-Napoca
Romania

Universitatea Tehnica “Gheorghe Asachi”- Iasi
Romania

# NOISE DETECTION IN A MECHANICAL SYSTEM

*Alexandru DVORNIC ,*Sorin CANANAU,**Thami ZEGHLOUL

*Universitatea “POLITEHNICA” Bucureşti, Romania
** PPRIMME - Pôle Poitevin de Recherche pour l'Ingénieur en Mécanique, Matériaux et Energétique - Département Génie Mécanique et Systèmes Complexes -France

**ABSTRACT**

The current paper presents a general image of acoustics within the car building industry. The thesis is focusing on the ˝booming˝ noise (noise produced by the internal combustion engine). In the urban areas, at low and medium speeds this represents the main type of noise produced by an automobile. Methods of measurement have been presented and the results have been compared with those existing in the specifications given in the initial state of design. Furthermore methods of controlling and improving the noise level in a transmission chain have been illustrated

## 1. INTRODUCTION

The increase in power and speed in automotive industry has brought an increase in the noise level generated by cars.
The problem of studying and controlling the noise generated by an automobile is complex and has different aspects:
(a) The physical study of noise appearance, detection of noise sources and analysis of noise spectrum; (b) the study of noise level quality of different car pats in different speed and load regimes; (c) the determination of a series of solutions for controlling the noise and stopping its propagation; (d) the designing and manufacturing of a series of efficient noise attenuators; (e) the determination of unique measurements of automotive noise and the determination of acceptable acoustical levels.
There is a high number of sources of noise in a mechanical system chain. For example, in a mechanical system of an automotive product we can identify:
1. Internal sources: (a) main source, due to the engine propelling group(engine and gearbox )
(b) secondary sources, due to ventilation system etal.
2. External sources like wheels noise, due to the contact between the wheels and the pavement, aerodynamic noise etal.

Let us consider that the engine noise is predominantly at idle and in situations where the engine is running in high speed (when trying to overcome a slope) and the aerodynamic noise can be neglected at speeds of lower than 2,5 m/s. At high speeds this is the main source of noise.
The rolling noise(produced by the wheels) increases with speed but also in smaller proportion than the aerodynamic noise, its level is mainly influenced by the road condition. The major sources of noise and vibration that occurs during the movement of a vehicle are the engine, gear bodies, air resistance and the wheels.
The noise share can be systematized as in the figure below [1]:

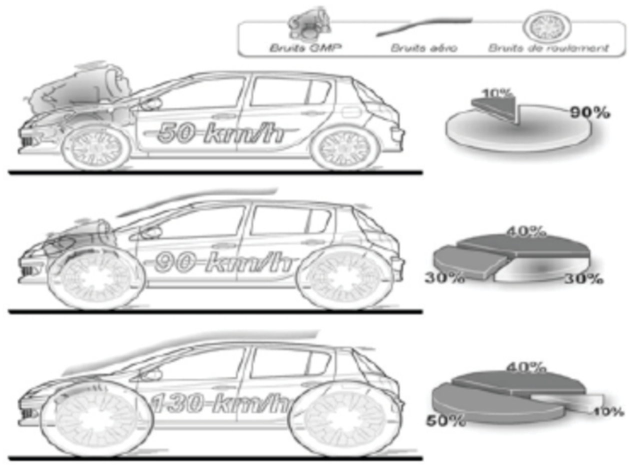



Fig.1. Noise sources share in a vehicle

## 2. POWER ENGINE NOISE GENERATOR: THE BOOMING NOISE

It is a pure sound with a frequency equal to the explosion in different cylinders. in the case of a 4 stroke gasoline engine with four cylinders there are two explosions for one rotation of the crankshaft (on explosion per cylinder for every two revolutions of the crankshaft).

A regular engine has a speed regime between 900 rpm and 6000 rpm that corresponds to a frequency between 30 Hz and 200 Hz (30 Hz to 900 rpm, 100 Hz to 3000 rpm, 200 Hz to 6000 rpm). Inside the cockpit these levels can be important, especially in some particular speed regimes where they can cause major discomforts.

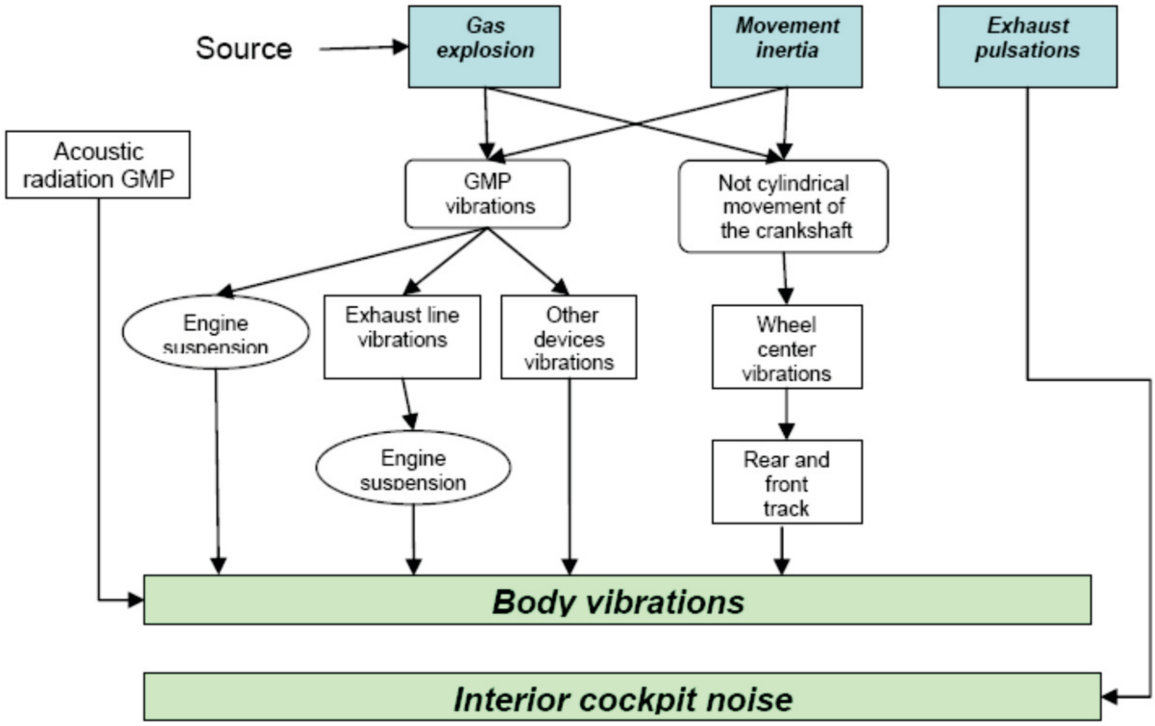



Fig.2. Booming noise sources

Consider the following system: the engine block which contains the piston and crank shaft which are connected by a rod. Let us consider the geometrical data as are given in the Fig.3.

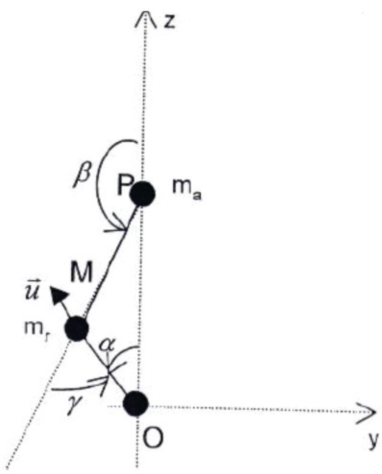



Fig.3. The engine block as mechanism system

With these notations we compute the position of P with respect to time in a dynamic system.

We have :

$$z_p = R \cos \alpha - L \cos \beta \qquad (1)$$

Also:

$$z_p = R\left(\cos \alpha + \lambda \sqrt{1 - \frac{\sin^2}{\lambda^2}}\right) \qquad (2)$$

Neglecting the rotational speed of the engine changes we can write α = ωt with ω being constant, in practice λ is greater than 3. We can then calculate $\dot{z}_p$ and $\ddot{z}_p$ hence (retaining only the dominant terms in $1/_{\lambda}$)

$$\dot{z}_p = -R\cdot\omega\left[\sin\ \alpha + \frac{1}{2\lambda}\cdot\sin\ 2\alpha - \frac{1}{16\ \lambda^3}\cdot\sin\ 4\lambda + ...\right] \quad (3)$$

$$\ddot{z}_p = -R\cdot\omega^2\left[\cos\ \alpha + \frac{1}{\lambda}\cdot\cos\ 2\alpha - \frac{1}{4\lambda^3}\cdot\cos\ 4\lambda + ...\right] \quad (4)$$

The forces exerted on the piston are $\vec{F}_g$, $\vec{F}_c$ and $\vec{F}_b$ the latter being exerted by rod.
Thus neglecting piston movements we have:

$$m_a\cdot\ddot{\vec{z}}_p = -\vec{F}_g - \vec{F}_c + \vec{F}_b \quad (5)$$

Thus we can write :

$$\vec{F} = -\vec{F}_b - \vec{F}_t - m_a\cdot\ddot{\vec{z}}_p \quad (6)$$

We conclude that the final force exerted on the engine block is:

$$\vec{F} = m_r\omega^2 R\vec{u} - m_a z_p \quad (7)$$

## 3. PRACTICAL CASE STUDY

Inside the testing system that we measure microphones are installed, at sensible points of noise detection, followed by running in the gear in which we want to obtain the data.
We use a digital analog converter which records the movement of microphone's membrane transforming it into electrical impulses. The next step is the post processing of data obtained during the measurement, which includes a detailed analysis of this spectrum to identify each type of noise, of causes and possible solutions to combat. Processing the measurements using the TestLab application. For a better identification of the noise, the spectrum obtained is later transformed into a 3D spectrum(rpm / db (A) / Hz), where the identification of the main types of noise is composed. The results for the noise spectrum are shown in Fig.4.
The second Harmonic of the engine noise from engine source is captured and is shown in Fig.5

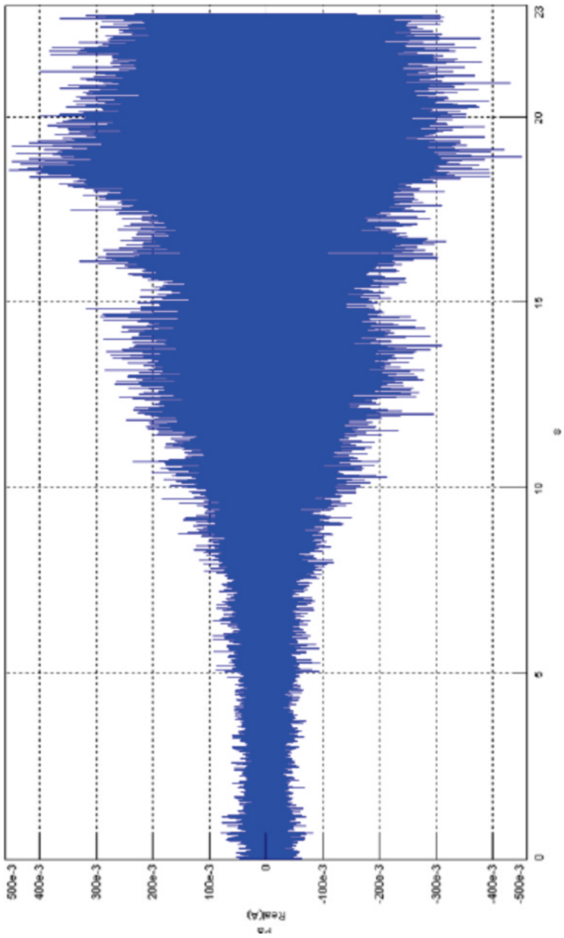


Fig.4. Noise spectrum

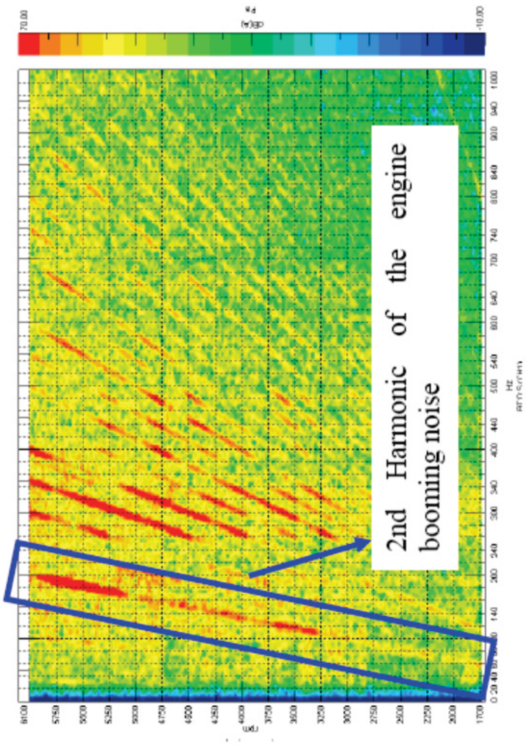



Fig.5. 3D noise spectrum separation

Following measurements on this mechanical system, in the third speed, for the gear system the following results have been obtained (compared with the optimal set at the design stage):

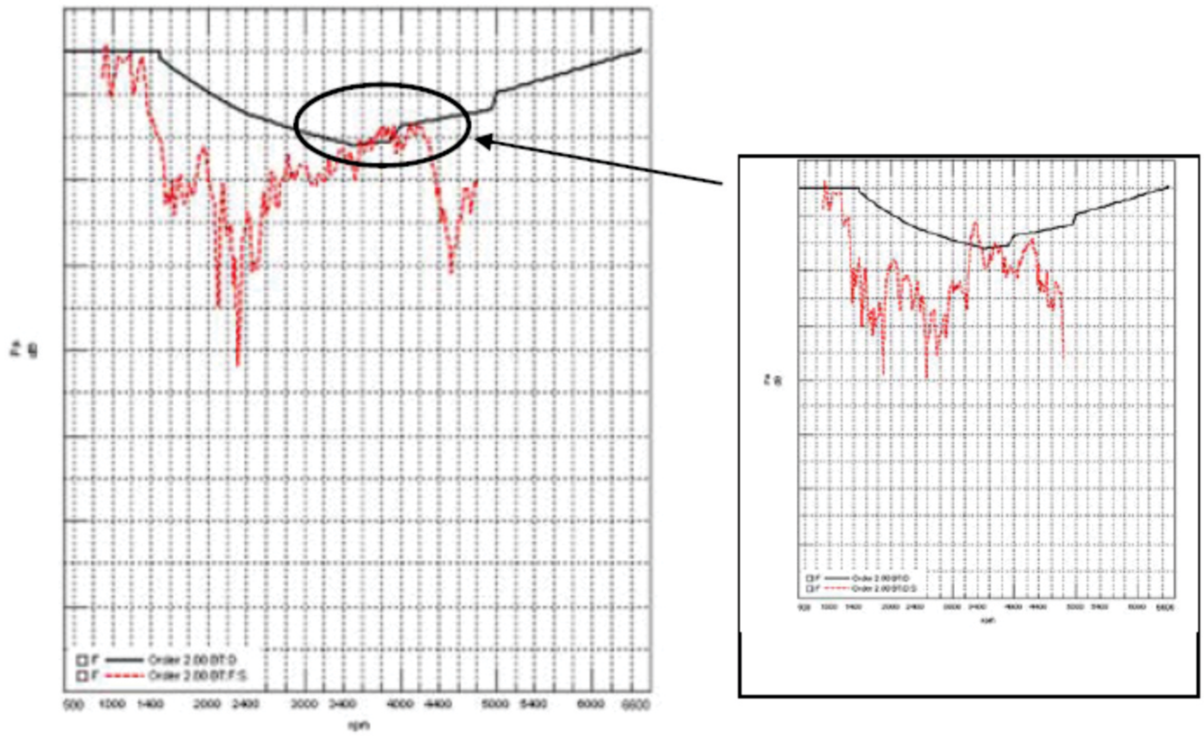


Fig.6. Comparison between measurement and technical specification

## 4. CONCLUSIONS

For the current stage of development of the automotive industry and for perspective, it emerges clearly the attempt of designing vehicles having quality indicators (reliability, ergonomics, etc.) as well as high performance (power, torque).
The examples are endless, yet experience shows that the operation of such vehicles in line with the trend set out above is likely to generate considerable vibration and noise. This is a very negative aspect when purchasing a vehicle.
On the other hand the negative impact of noise and vibration on human health is well known. Noise and vibration also influence the design and performance of vehicles having obvious consequences in the quality of the automotive products.

Therefore there are well justified reasons for the development of measures to prevent and control noise and vibration both from social and economical point of view.
From an economical perspective, reducing noise and vibration is one of the requirements of modern car buyer. As we know, there are many cases in which quality is assessed from the level of noise and vibration point of view.
When combating noise we always have to consider the 'source-propagation path-receiver' system, system that can have a large diversity.
Combating the noise can be achieved at source, propagation path or receiver level.

## REFERENCES

[1] Parizet, E. : Acoustique Automobile – Doc. Insa Lyon Laboratoires de Vibration et d'Acoustique, mars 2002

# 8. IFK call for papers

www.ifk2012.com

**International Conference on Fluid Power 26 - 28 March 2012, Dresden**

The 8th International Conference on Fluid Power (8th IFK) will be held in Dresden from 26 to 28 March 2012. The IFK is one of the world's most significant scientific conferences on fluid power control technology and systems. It offers a common platform for the presentation and discussion of trends and innovations to manufacturers, users and scientists. The symposium on the first day is dedicated to presentations focused on methodology and fundamental research. The two following conference days offer a wide variety of application and technology orientated papers about the latest state of the art in fluid power. It is this combination that makes the IFK an unique and excellent forum for the exchange of academic research and industrial application experience. An exhibition taking place simultaneously offers the possibility to get product information and to have individual talks with manufacturers. The conference is followed by two days of excursions to regional companies and technical sights. The relevance of fluid power in machinery and plant engineering is immense. Driven by the challenges of rising commodity prices, a responsible use of resources, the competition with electric drives and not least the legal framework, the manufactures and users take the next steps of development offensively. The consequent improvement of technology in terms of energy efficiency, energy recovery and environmental compatibility plays a particular role. With innovative approaches the fluid power industry emerges confidently from the crisis and manifests itself again as motor of the drive and control branch. This is reflected in the variety of conference topics, which are themed **"Fluid Power Drives!".** Authors interested in contributing a paper to the conference are requested to forward an abstract of the article to the e-mail address papers@ifk2012.com at the latest by 31 May 2011 and full paper at the latest by 15 December 2011.

**Please use the template available at www.ifk2012.com.**

The IFK addresses to application-oriented topics in the field of industrial hydraulics, mobile hydraulics and pneumatics as well as innovations in fluid power components. Moreover, articles about specific applications, renewable energies, fundamentals, tribology, materials and machine availability and safety are welcome. The important aspect of education and further training in the fluid power branch is also addressed for the first time to the 8th IFK. The conference topics are suggestions for the papers. Unconventional contributions are explicitly welcome! Further information is available under www.ifk2012.com. The IFK team led by Prof. Jürgen Weber looks forward to receiving a lot of interesting papers for the 8th IFK 2012 in Dresden.

**Contacts:**

***General:*** Markus Schneider, Tel.: +49 351/463-38609, E-Mail: general@ifk2012.com

***Authors:*** Martin Petzold, Tel.: +49 351/463-33701, E-Mail: papers@ifk2012.com

***Exhibition:*** André Sitte, Tel.: +49 351/463-33707, E-Mail: exhibition@ifk2012.com

# THEORETICAL AND EXPERIMENTAL RESEARCH REGARDING THE DYNAMIC BEHAVIOUR OF LINEAR HYDRAULIC MOTORS

**Ph. D. Eng. Corneliu CRISTESCU*, Ph. D. Eng. Petrin DRUMEA*,**
**Ph. D. Eng. Dragos ION GUTA*, Ph. D. St. Catalin DUMITRESCU*, Eng. Petrica KREVEY***

*INOE 2000 - IHP

**Abstract**
*The paper presents the results of extensive research on the dynamic behaviour of linear hydraulic motors, carried out in INOE 2000-IHP, in the framework of the NUCLEU Programme. The research has been conducted both theoretically and experimentally. Theoretical research has taken place with the modern means of mathematical modeling and numerical simulation, while experimental investigations were conducted on an experimental stand with data acquisition and computer processing. The article presents some theoretical and experimental results obtained in the research, results that are of real scientific interest, but also have practical value through their use in the design of fluid power components and equipment.*

**Keywords:** linear hydraulic motors, dynamic behaviour, mathematical modeling and numerical simulation, theoretical research, experimental research, test stand, sensors and transducers, data acquisition.

## 1. INTRODUCTION

In the structure of hydraulic drive systems, in addition to the equipment for generating, conditioning, control and distribution of the working fluid, there are hydraulic operative elements (motors) that make transformation / conversion of hydraulic energy into mechanical energy and perform mechanical work required by the drive system. Therefore, knowing the dynamic behaviour of hydraulic operative elements, as *operative parts of the working machines*, is of particular interest to ensure performance of hydraulic drive equipment and systems. The main operative elements used within hydraulic control and actuation systems are classified in two categories namely: *linear hydraulic motors and rotary hydraulic motors*.[1].

**Linear hydraulic motors,** which are the subject of research presented, are operative hydraulic elements that perform a linear motion at the working mechanism of equipment and machinery. These operative elements have as their characteristic the **rectilinear motion** and they are currently known as *hydraulic cylinders*, or servo cylinders, or hydraulic actuators, Figure 1.

Knowing the dynamic behaviour of linear hydraulic motors (MHL), early since the design stage, involves conducting theoretical research based on mathematical modeling of mechanical and hydraulic processes, as well as computer numerical simulation. To develop mathematical models of dynamic behaviour specific equations from theoretical mechanics have been used, as well as the specific ones from fluid mechanics and fluid power. For numerical simulation of dynamic processes there has been used a simulation environment [5] (MATLAB with Simulink), which allowed the development of appropriate simulation software.

In addition to theoretical research of dynamic behaviour, a substantial role is played by experimental research, designed to confirm, based on *experimental measurements*, *the actual performance* of the dynamic behaviour of hydraulic operative elements. To experimentally test the linear hydraulic motors in order to investigate the factors that influence the dynamic behaviour of linear hydraulic motors (MHL), there has been necessary to design and construct an experimental test stand, which allowed conducting experimental research in good condition.

**Figure 1:** *REXROTH Linear Hydraulic Motor (MHL)*

The test stand was developed inside of the *Laboratory of Servo-Control Equipments* from INOE 2000-IHP.

At the begining, having at disposal modern methodologies of research for the dynamic behaviour [2], [3], [5], [6], based on using some performant simulation softwares for the dynamic systems, it has started a theoretical research of the transitory and stationary working regimes, in order to find the dynamic responses of this modern servo-systems.

## 2. THEORETICAL RESEARCH OF THE DYNAMIC BEHAVIOUR OF A *MHL*

To conduct theoretical research on dynamic behaviour of linear hydraulic elements, over time range, there was chosen as research object an electro-hydro-mechanical ***servo system*** comprising a real **linear hydraulic motor** MHL, with bilateral shaft, controlled by an **electro hydraulic servo valve SV**, manufactured by MOOG, as is shown in Figure 2.

The linear hydraulic motor MHL **is positioned vertically** and is loaded, successively, with different inertial masses *M*, of weight *G*, in accordance with those shown in Figure 2. Actuation of the linear hydraulic motor MHL is performed using the servo valve SV, based on pressurized oil supplied by the generator-hydrostatic pump with variable flow, assisted by a pressure limiting valve SLP.

To see what is the dynamic response of the system there has been designed a mathematical model for each system component, and based on interacting between them, there has been made the mathematical model for the entire system under investigation.

The theoretical research was performed on the groundwork of the modern method of analysis and synthesis, for each component element and for the entire system, by computerized modeling and simulation of the dynamic behaviour. Modeling of elements and systems was performed, in more stages, as listed below: physical modeling of the electro hydraulic servo-system; systemic modeling of the servo-system; mathematical modeling for each element and, then, for the entire servo-system. Finally, there was elaborated the simulation program which allows to obtain the theoretical research regarding the dynamic behaviour of the linear hydraulic motor.

### 2.1. The physical model of the servo system

The physical model of the servo-system, presented in figure 2, is a principial hydro mechanic schematic diagram, necessary to indicate all the physical parameters involved in the process. The physical model comprises a linear hydraulic motor MHL, with its servo-valves and hydraulic circuits considered as concentrated parameters of resistances and hydraulic capacity. The main component elements are the following: the electric drive motor ME, the hydraulic generator GH, the pressure limiting valve SLP, the distribution servo valve SV, the linear hydraulic motor MHL and, also, the oil tank RU. The charge is realized by means of some masses *M* with *G* weight. Also, the hydro-mechanic model comprises two other hydraulic circuits: the hydraulic drive circuit (GH-MHL), from pump HG to the hydraulic motor MHL and the discharge hydraulic circuit (MHL -RU), from motor to the oil tank RU.

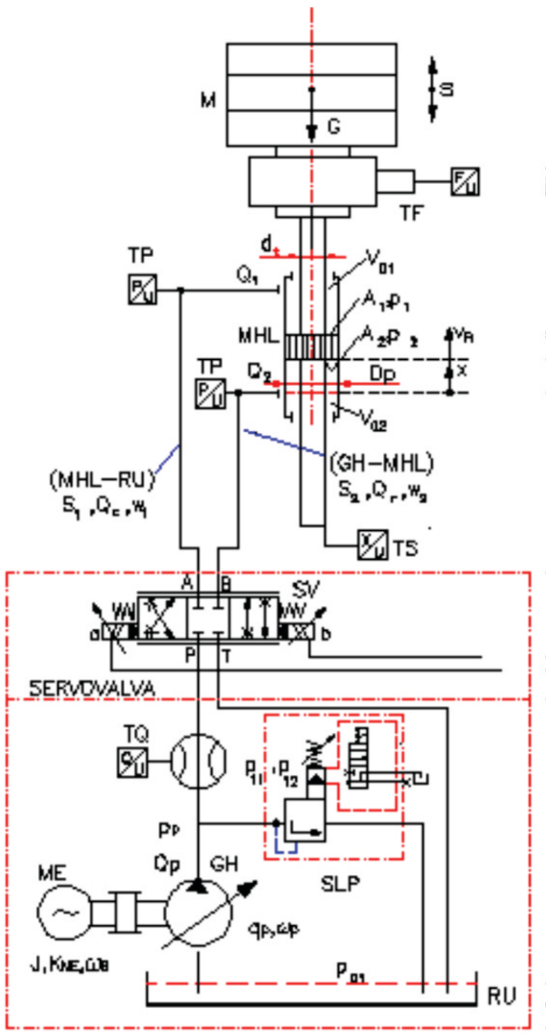



**Fig. 2** *The physical model of the servo-system*

## 2.2 The systemic model of the servo-system

Because the dynamic phenomena which take place during the operational modes of the hydro mechanic drives [8] represent a result of the interaction between the electro-mechano-hydraulic servo-system and, also, the effective operational process, it was approached the ***theory of systems*** as theoretical groundwork for the research development.

Starting from the physical model shown in Figure 2, there was elaborated on the base of the theory of automatic systems, a ***systemic model,*** which analyzes the specific systemic elements, the afferent inputs and outputs and the interrelations between them. The systemic model elaborated is shown in Figure 3 and it comprises ***7 systemic elements***:

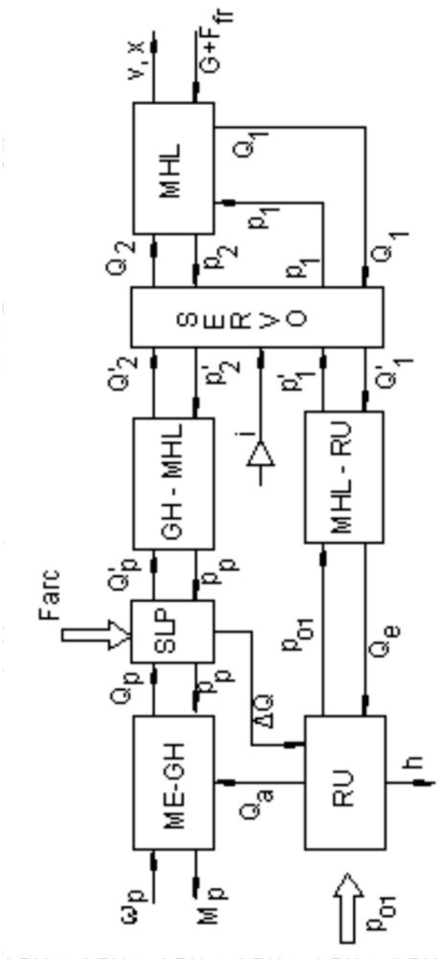



**Fig. 3** *The systemic model*

- **hydraulic generator GH,** rigidly coupled with the electric motor ME, as a ***quadrupolar element***, ME - GH, characterized by the mechanical parameters $\boldsymbol{\omega}_p$ and $M_p$ and the hydraulic parameters $Q_p$ and $p_p$. The input parameters are $\boldsymbol{\omega}_p$ and $p_p$ (pressure at pump and the output parameters are the flow $Q_p$ and the necessary moment $M_p$. In this hypothesis the moment of inertia **J** is cumulated for the pump and electric motor, which allows approaching them as a single systemic element and this is associated with the hydraulic generator element GH;

- **the pressure limiting valve SLP**, as ***bipolar element***, where the input parameter is the pump pressure $p_p$ , and the output parameter is the flow leakage $\Delta Q$.

- **the hydraulic drive circuit GH-MHL,** as a ***quadrupolar element***, characterized only by hydraulic parameters, $Q_p$ and $p_2$ being input parameters, and $Q_2$ and $p_p$ being output parameters;

- **the servo-valve SV,** as a ***multipolar and multivariable element,*** which commands the linear hydraulic motor LHM, in order to obtain a proportional movement at the MHL;

- **the linear hydraulic motor MHL, as *a hexapollar element***, transforms the hydraulic parameters $Q_p$ and $p_{2,}$ into mechanic parameters: elevation speed $v_r$ and elevation force $F_R = G_{Mv} + F_{fr}$. The linear hydraulic motor is characterized by the flow input values $Q_2$ and the resistant forces $F_R$ (represented by the weight of the mobile masses $G_{Mv}$ and friction forces $F_{fr}$) and the output values $v_R$ and pressure from the elevation circuit $p_2$;

- **the discharge hydraulic circuit MHL-RU,** as a ***cvadripolar element*** with the input values flow $Q_1$ and $p_{01}$, and the output values $Q_e$ and pressure $p_1$;

- **the oil tank RU,** as a ***bipolar element***, with input parameter the flow $Q_e$ and the output the variation of the fluid level *h*.

### 2.3 The mathematical model of the servo-system

The analysis and synthesis imply a description of the dynamic behaviour of the hydro-mechanic elements by mathematical modeling in the form of some differential equations which to express the constructive and operational characteristics; then according to an adequate mathematical formalization on the base of interconditionings there is made their synthesis [1],[2],[3],[6].

On the base of the physical model from Figure 2 and of the systemic model from figure 3, was conceived a mathematical model, which depicts the dynamic behavior of the transfer manipulator for the elevation phase, on the base of the following symbols: $D_P$ - piston diameter, $d_t$ -rod diameter; $G$ - the weight of the mobile parts; $M$ - reduced mass; $p_{01}$ - the fluid pressure from RU; $_1$ - coefficient equivalent of the pressure losses on the discharge circuit; $x$ - piston stroke; $S_1$ - section of the discharge pipe; $w_1$ - flow speed; $m_1$ - reduced volume of the fluid from the discharge pipe; $V_{01}$ - the volume of fluid from the passive chamber and the discharge pipe; $x.A_1$ - the volume of fluid discharged by piston; $A_2$ - the volume created by the elevation of the piston on the stroke $x$; $V_{02}$ - the volume of fluid on the active chamber and the elevation pipe; $p_1$ - effective pressure on the passive surface of the piston $A_1$; $p_2$ - the effective pressure on the active surface of the piston $A_2$; $S_2$ - the section of the pipe of elevation of the piston; $w_2$ - flow speed; $m_2$ - reduced volume of fluid from the elevation pipe; $_2$ - coefficient equivalent to the pressure losses; $p_{01}$ - fluid pressure from the aspiration recipient; $Q_1/Q_2$ – discharge/elevation flow, at exiting entering in the motor; $Q_e$ - the flow on the discharge pipe; $p_e$ - fluid pressure at the exit from the discharge pipe; $Q'_p$ - momentary flow discharged by pump; $q_p$ - the pump volume; $_p$ - momentary angular speed of the pump; $_s$ - angular speed of synchronization of the electric motor; $F_{fg}$ - friction force in the slider; $_0$ - dry friction coefficient; $b_M$ - linear coefficient of the force losses proportional with speed; $a_M$ - linear coefficient of flow losses at piston MHL, proportional with pressure; $h$ - height of fluid in the the oil aspiration reservoir; $A_{RU}$ - aria section of the aspiration reservoir; $vh$ - level variation speed; $E_1/E_2$ equivalent elasticity modules of the pipes; $a_P$ - linear coefficient of the flow losses at pump; $b_P$ - linear coefficient of the couple losses; $c_{fP}$ - dry friction coefficient at pump; $K_{ME}$ - slope specific for the electric motor. Taking into account the motion direction of the slider of the manipulator, which is from downwards to upwards, are written the expressions of the fluid volumes from the bottom and top chambers of the linear hydraulic motor, which are given by the sum, respectively, the difference between the initial volumes $V_{02}$, $V_{01}$, and the volumes generated by piston, $x.A_2$, respectively, $x.A_1$, which are compressed. With the symbols and hypothesis from above, in accordance with the professional specialized literature, [2] and [3], were written the corresponding equations for each of the ***7 systemic elements*** and, finally, was obtained the mathematical model for the servo system in the elevation of the masses, composed of the next equations:

$$Q_p^{\cdot} = \frac{q_p}{2\pi}\omega_p - a_p p_p - \frac{q_p}{2E}\frac{dp}{dt} \quad (1)$$

$$J\frac{d\omega_p}{dt} + b_p\omega_p + \frac{q_p}{2\pi}p_p + c_{fp}\frac{q_p}{2\pi}p_p = K_{ME}(\omega_s - \omega_p) \quad (2)$$

$$m_2\frac{dw_2}{dt} = S_2[(p_p - p_2) - 0{,}5\frac{\gamma}{g}\xi_2 w_2^2] \quad (3)$$

$$\frac{dp_2}{dt} = \frac{E}{S_2 l_{cond}2}[Q_p - Q_2] \quad (4)$$

$$Q_2 = S_2 \cdot w_2 \quad (5)$$

$$M\frac{d^2x}{dt^2} + b_M\frac{dx}{dt} + \mu_0\frac{\dot{x}}{|\dot{x}|}(A_2 p_2 - A_1 p_1) + F_{fg} = (A_2 p_2 - A_1 p_1) - G \quad (6)$$

$$Q_2 = A_2\frac{dx}{dt} + a_M(p_2 - p_1) + \frac{V_{02} + x A_2}{E}\frac{dp_2}{dt} \quad (7)$$

$$Q_1 = A_1\frac{dx}{dt} + a_M(p_2 - p_1) - \frac{V_{01} - x A_1}{E}\frac{dp_1}{dt} \quad (8)$$

$$Q_e = S_1 \cdot w_1 \quad (9)$$

$$\frac{dp_1}{dt} = \frac{E}{S_1 l_{cond}1}[Q_1 - Q_e] \quad (10)$$

$$m_1\frac{dw_1}{dt} = S_1[(p_1 - p_{01}) - 0{,}5\frac{\gamma}{g}\xi_1 w_1^2] \quad (11)$$

$$Q_a - Q_e = -A_{RU}\frac{dh}{dt} \quad (12)$$

$$Q_a = Q_p \quad (13)$$

$$\begin{cases} \frac{1}{\omega_N^2}\cdot\frac{d^2y}{dt^2} + \frac{2\xi}{\omega_N}\cdot\frac{dy}{dt} + y = Ki \\ y(0) = \dot{y}(0) = 0 \end{cases} \quad (14)$$

$$Q_{SV} = K.i \quad (15)$$

$$p_{SV} = (p_2 + \Delta p_i) + (p_1 + \Delta p_e) \quad (16)$$

$$U = K_N \cdot x \quad (17)$$

$$i = K_R V \quad (18)$$

The equations (14-16) represent a simplified mathematical model of the servo valve, where y is the stroke of its plunger and Nw- natural pulsation; x- amortizing factor; K- amplification factor. The relation (17) is the ST stroke transducer equation and the relation (18) is the electronic regulator equation. This set of equations represents the mathematical model for the studied servo-system.

**2.4. The simulation model of the dynamic behavior of the MHL**

For making the simulation experiments on computer upon the dynamic behavior of the MHL, was necessary to elaborate, on the base of the mathematical model presented above, a specialized calculation program. For this, was used a modern simulation software, tool *MATLAB with Simulink* [5]. The elaborated program realizes a simulation model of the dynamic behavior of the servo-system, in the phase of elevation, on the base of the specific block from the Simulink menu and, also, on the special created blocks. The simulation model is presented in figure 4:

Using this special created simulation model, it was possible to obtain the graphical variation of the main dynamic parameters, which are presented in the next figure.

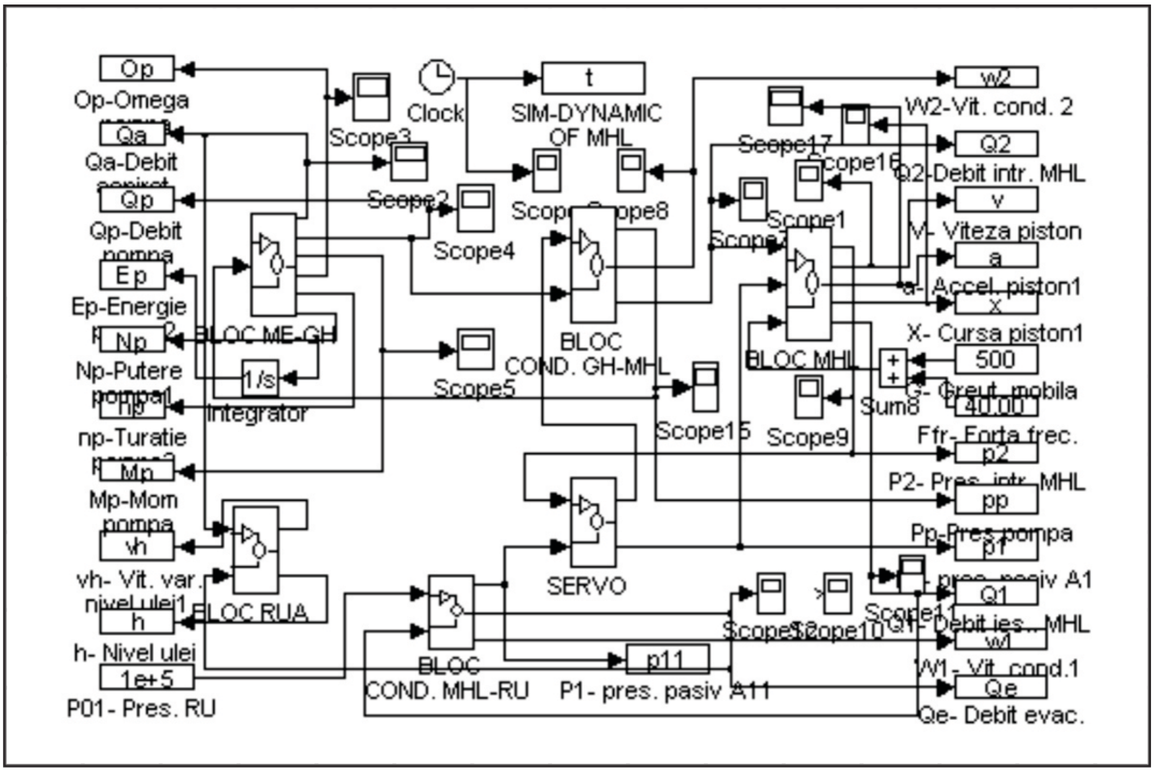



**Fig. 4.** *The simulating model of the dynamic behavior of the MHL*

## 2.5 The computer simulation experiments

The simulation experiments were performed using as input data constructive functional parameters of the servo-system from which are mentioned:

· Nominal loads ................. 50-100-170 N;
· Max mobile weight ...................... 500 N;
· Max stroke .............................. 0,160 m;
· Max elevations velocity..........0,100 m/s;
· Piston/rod diameter of MHL.....0,1054/0,050 m;
· Pump displacement ... .0,018 x $10^{-3}$ $m^3$/rot;
· Power/revolution of ME ...7 kW/1450 rot/min;
· Nominal diameters of pipes .........12 mm.

By using the simulation programs, was acquired the variation of all parameters of interest, in graphical form, their evolution being in accordance with the known data.
In figure 5, is shown the variation of the elevation stroke, which has a lower evolution in the first part, motivated by the presence of inertial forces in the system, and, then, an almost constant variation. From the diagram may be found that the maximum elevation stroke of 0,160 m is made in about 2 second.
The elevation speed has an evolution characterized by a rapid increase, then stabilization at a value of aprox 0,085 m/s, according to figure 6, which is in the range of maximum theoretical speed values of 0,100 m/s.
In the first part of the elevation stroke, the acceleration varies very abrupt, then, according to figure 7, when the speed is stabilized, the acceleration is annulled. By knowing this evolution may be appreciated the values of the inertial forces from the system at the start of the elevation stroke being possible to take the necessary measures for eliminating their effects.
The variation of the rotation speed of the pump shaft is presented in the Figure 8, where it can see o normally evolution to the synchronizing value of 1500 rot/min, when the pump pressure varies like in the Figure 9, characterized by some oscillations which appear at the beginning, but the stabilized pressure is about 85 bar, which is in concordance with servo-valve SV.

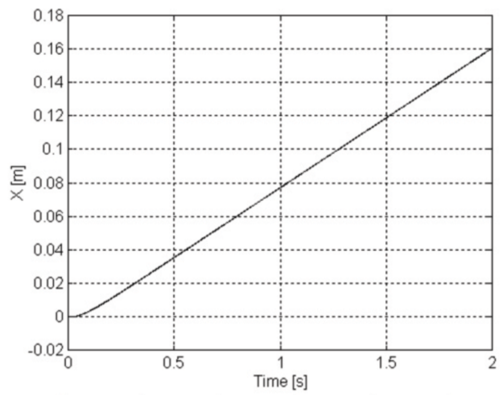



**Fig. 5** *The variation of the elevation stroke*

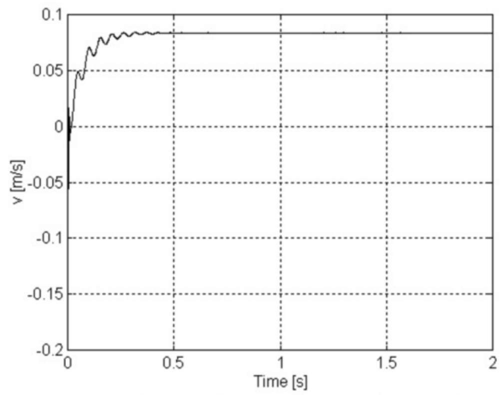



**Fig. 6** *The variation of the elevation velocity*

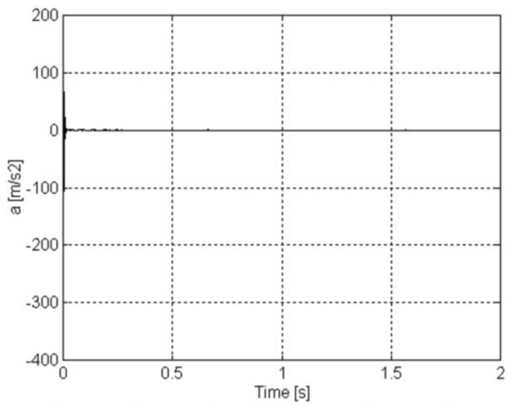



**Fig. 7** *The variation of the acceleration*

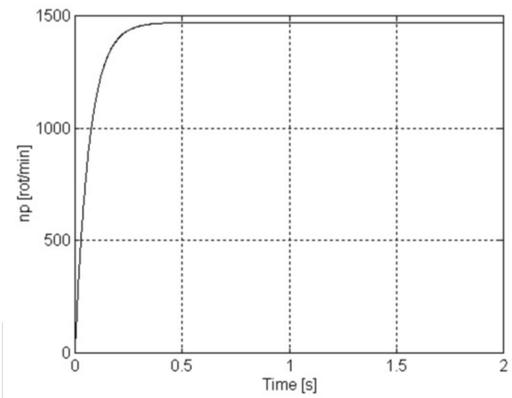



**Fig. 8** *The variation of the rotation speed*

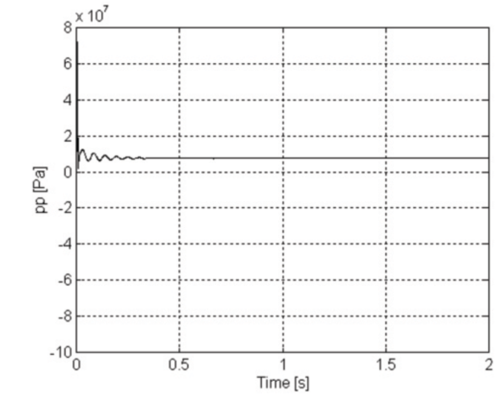



**Fig. 9** *The pump pressure variation*

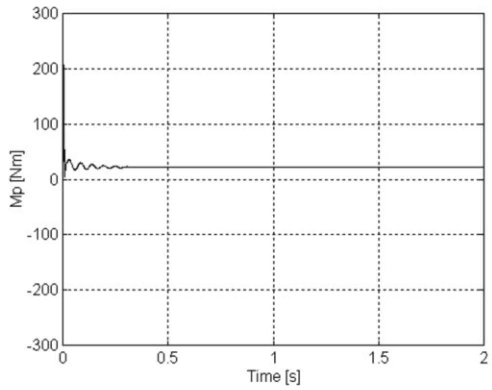



**Fig. 10** *The pump shaft moment variation*

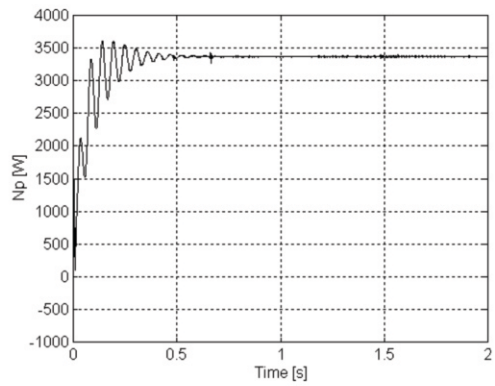



**Fig. 11** *The pump shaft power variation*

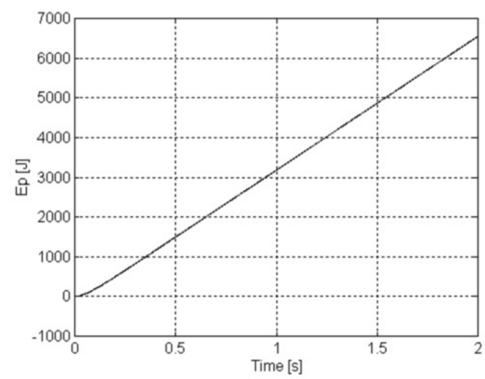



**Fig. 12** *The pump shaft energy variation*

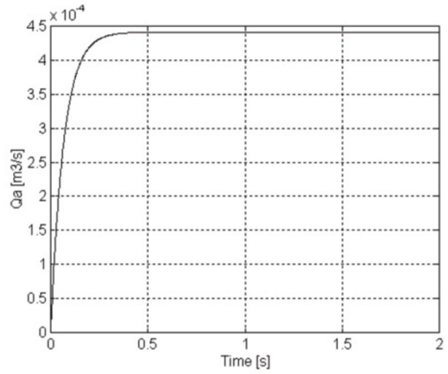



**Fig. 13** *The pump theoretical flow variation*

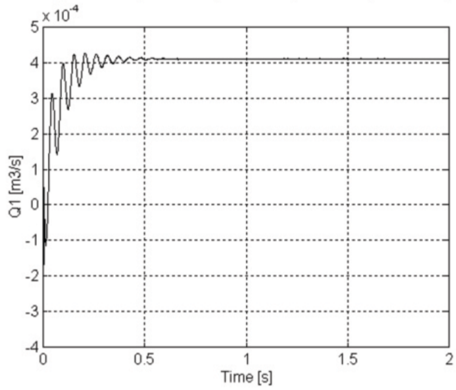



**Fig. 14** *The output flow variation from MHL*

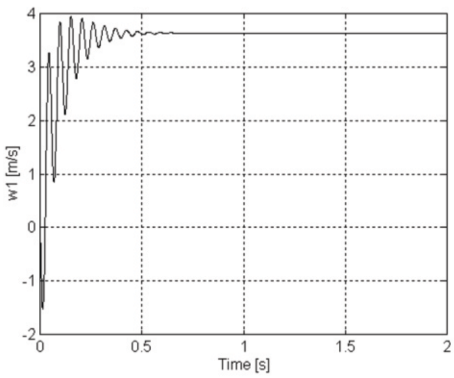



**Fig. 15** *The output flow velocity variation*

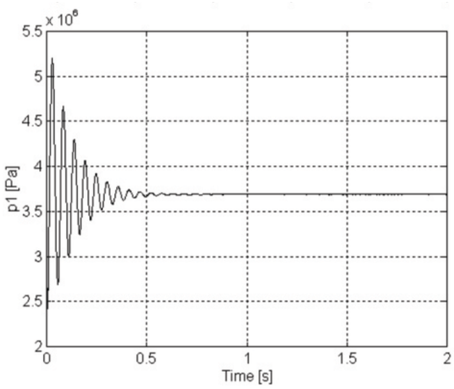



**Fig. 16** *The output flow pressure variation*

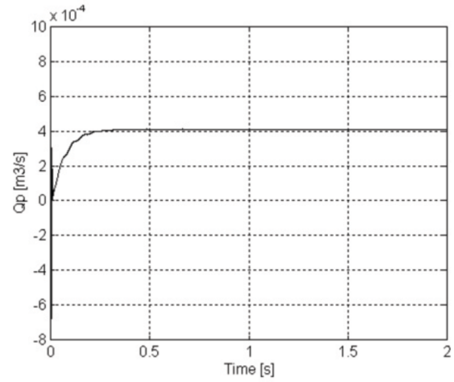



**Fig. 17** *The  pump real flow variation*

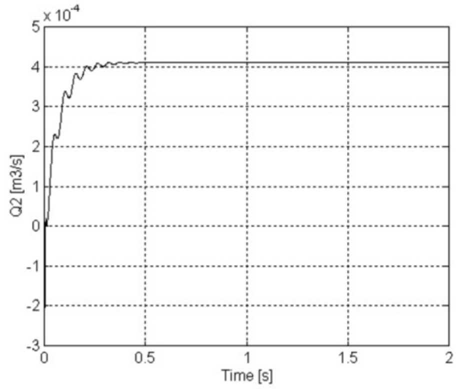



**Fig. 18** *The input flow variation into MHL*

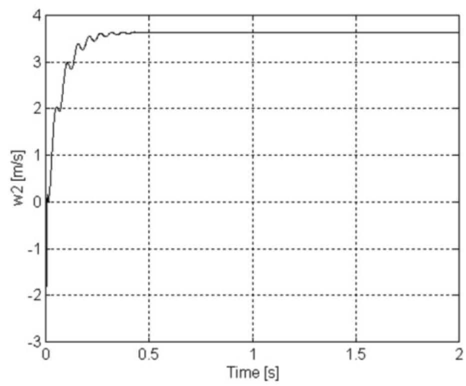



**Fig. 19** *The input flow velocity variation*

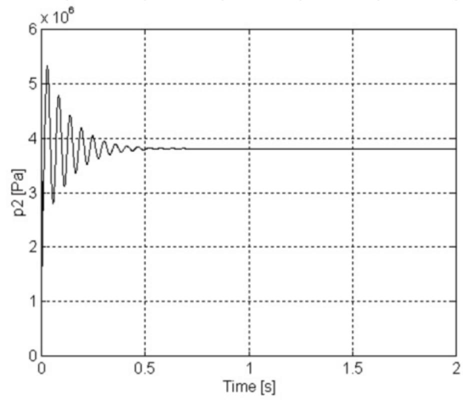



**Fig. 20** *The input flow pressure variation*

The diagram from Figure 10 shows the evolution of the necessary moment at the pump shaft, and the diagram from Figure 11 shows the power variation at the of pump shaft. On the base of these diagrams of the operational cycle, may be chosen precisely the suitable electric motor.

Figure 12 shows the variation of energy consumption in elevation stroke, and allows evaluating of the energy efficiency of the studied elevation servo-system. The Figure 13 shows the variation of the theoretical flow of the pump, which is about 4.4 $10^{-4}$ $m^3$ /s, very close with the calculated values. The evacuated flow from the MHL $Q_1$ is presented in the Figure 14, where it can see some oscillations at the beginning of the elevation stroke. The evacuated flow velocity, from MHL, is shown in the Figure 15 and the pressure in the Figure 16. The real pump flow variation is illustrated in the Figure 17, where it can see that the real floe is about the 4.0 $10^{-4}$ $m^3$ /s, smaller than the theoretical flow, presented in the Figure 13, with 10 %.
The variation of the fluid flow $Q_2$, which enters the linear hydraulic motor MHL, shown in Figure 18, is affected by the evolution of the elevation speed and corresponds with the theoretical flow of the pump. The flow velocity $w_2$ is under 4 m/s, as it can see in Figure 19. The Figure 20 highlights an elevation pressure, at the entrance of MHL, with the value to 40 bar, which is greater than the pressure value of the exit from MHL, bat a little than 35 bar, specific of the servo-valve.
The pressure required for elevation, at the enter to linear hydraulic motor MHL, varies like in Figure 20, where is noticed that although for operation in stabilized mode, is required only a pressure of 10-15 bar, the servo-valve do to appear a pressure exceeding values of over 70 bar, and the pressure oscillations are powerful enough, but are damped in the end. This oscillations highlights that it is required to be taken certain measures for alleviating pressure oscillations, which may lead to strong vibrations in the transfer system.
From the Figure 11, may be understood that the max power required at the electric drive motor may be less, depending on the real operational mode.

Also, on the base of this diagram from Figure 11, may be drawn the conclusion that the power of the electric motor of 7 kW, specified in the project, is too high, and that it should be provided a motor with less electric power, about. 4.0 kW.
This scientific study was finalized with interesting results which may constitute the groundwork for future research projects regarding the transfer systems.

## 3. EXPERIMENTAL RESEARCH OF THE DYNAMIC BEHAVIOUR OF A LHM

### 3.1. Introduction

To test the linear hydraulic motor by experiments, in order to investigate the factors that influence the dynamic behavior of these hydraulic operative elements, there has been necessary to design and construct an experimental test stand, allowing conducting of such experimental research. These experimental stands are a functional group of components, equipment, devices and proper instrumentation, aiming at allowing the necessary experimental research to be conducted, by which to highlight and quantify the factors influencing the dynamic behaviour of linear hydraulic motors.
In design and implementation of test stand for dynamic behaviour linear hydraulic motors the intention was to maximally use the existing facilities, to which there have been added other equipment, instruments, components and devices specially purchased, to minimize the costs of such an action. In this respect, there was made extensive use of the facilities existing in the institute.

### 3.2. Design of dynamic test stand

In design of the experimental test stand for dynamic behaviour of linear hydraulic motors, there has been considered the equipment existing in the *Laboratory of Servo-Control Equipments,* in INOE 2000-IHP, taking as a base the structure of an existing stand for testing seals of hydraulic cylinders. It is equipped with a hydraulic cylinder with bilateral rod, which is actually a linear hydraulic motor, mounted vertically, which may be required to lift various weights. Supply of the linear hydraulic motor is made through a servo valve, from a hydraulic pressure group, with manually adjustable flow and possibility for flow measurement.

The hydro-mechanical diagram of test stand for dynamic behaviour of a linear hydraulic motor is shown in Figure 21.

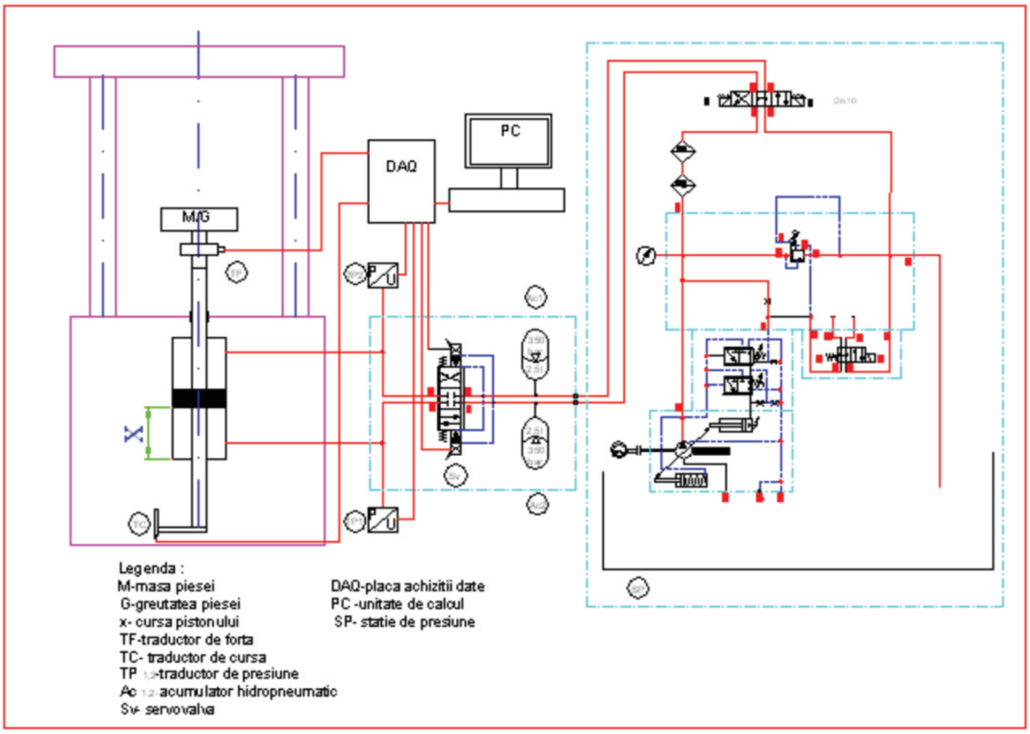



**Fig. 21** *Hydro-mechanical diagram of dynamic test stand*

The main components of the dynamic test stand, according to hydro-mechanical diagram shown in Figure 21, are the next ones:

- linear hydraulic motor, MHL;
- stroke transducer, TS;
- mass /weight to be lifted, M / G;
- force transducer, TF,
- pressure transducers, TP1 and TP2;
- servo valve SV;
- pressure accumulators, AC1 and AC2;
- pressure unit RU, comprising typical parts, mounted on a oil tank Rz;
- data acquisition board, DAQ;
- computer PC.

In principle, as shown in Figure 21, the hydro-mechanical diagram of the test stand for dynamic behaviour of linear hydraulic motor includes three major subassemblies, namely: the pressure unit, the hydro-mechanical system, which contains the linear hydraulic motor being tested, and the data acquisition system with computer, sensors and transducers [4].

**The pressure unit,** RU, provides adjustable oil flow under pressure, and it has all the elements specific to usual pressure blocks.

**The hydraulic operative system with linear motion** consists of *linear hydraulic motor* MHL and servo valve SV. The system is equipped with transducers needed to capture the evolution of parameters of interest: built-in stroke transducer TS, force transducer TF and pressure transducers PT1 and PT2.

The hydraulic operative system with linear motion, consisting of linear hydraulic motor and servo valve, is actually **the subject tested**, for the purpose of ascertaining the dynamic behaviour of linear hydraulic operative elements.

**The data acquisition system** consists mainly of the data acquisition board DAQ, computer PC and stroke transducer TS, force transducer TF and pressure transducers PT1 and PT2, and it works on the basis of data acquisition and processing software.

Charging the operative system is performed by placing on the motor shaft, over the force transducer, some parts with masses with different, but known, weights, M / G.

To conduct experimental research on dynamic behaviour of linear hydraulic elements MHL, there was chosen, as research object, an ***electro-hydro-mechanical servo system*** comprising a real **linear hydraulic motor MHL** with bilateral shaft, Figure 22, controlled by an **electro hydraulic servo valve SV**, manufactured by **MOOG Company,** Figure 23.

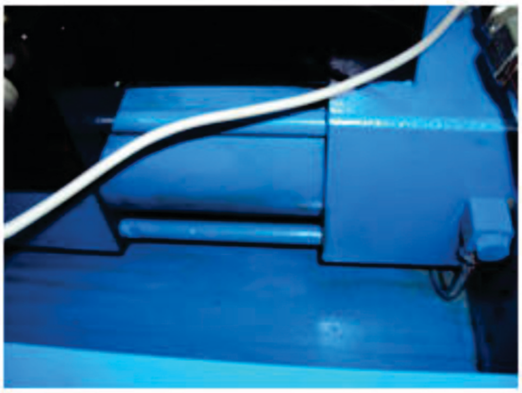


**Fig. 2** *The Liner Hydraulic Motor (MHL)*

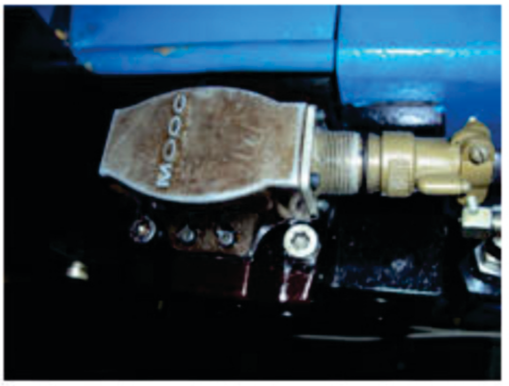


**Fig. 3** *The MOOG Servo-Valve (SV)*

**3.3 Physical implementation of dynamic test stand**

After design, the test stand has been physically implemented by mounting its components according to the hydro-mechanical diagram in Figure 21 and located in the *Laboratory of Servo-Control Equipments*, as it can be seen in Figure 22.

**Fig. 22** *Dynamic test stand for linear hydraulic motors*

**Location and mounting of transducers**

Of particular interest is how are located and mounted the transducers required to absorb variations of the main dynamic parameters, on which depends, ultimately, the acquisition accuracy of evolution of these quantities. The control servo-valve of the hydraulic linear motor is shown in Figure 23 and in the Figure 24 is shown the flow transducer mounted on the pump discharge circuit.

Figure 25 shows the force transducer which is located under the weights that are the load charging items of MHL and Figure 26 shows the pressure transducer mounted on the top of the linear hydraulic motor. In Figure 27 there can be seen the pressure transducer with local display and the gauge mounted on the discharge circuit of the MHL. and in Figure 28 is shown the stroke transducer partially incorporated in the structure of the hydraulic motor MHL [4].

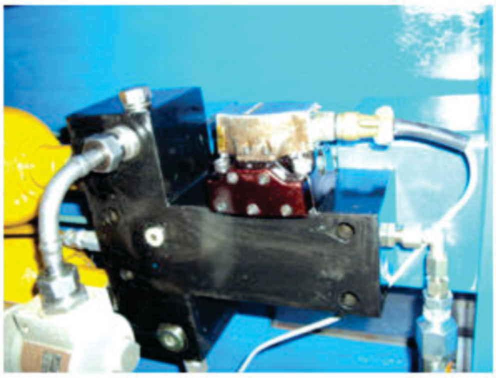


**Fig. 23** *Servo Valve*

**Fig. 26** *Pressure transducer*

**Fig. 24** *Flow transducer*

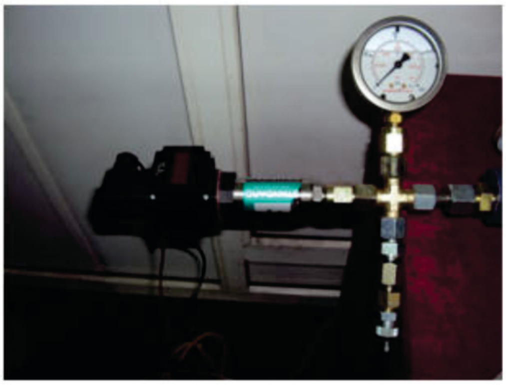


**Fig. 27** *Pressure transducer and gauge*

**Fig. 25** Force transducer

**Fig. 28** *Stroke transducer*

## 4. CONDUCTING EXPERIMENTS ON THE DYNAMIC BEHAVIOUR OF LHM

Conducting experiments on the dynamic behaviour of linear hydraulic motors was based on the testing software outlined at the beginning of the tests.

**Working mode**

For research of dynamic behaviour, there must be known variations over time of dynamic parameters of interest namely: variation of stroke, speed and acceleration, pressure variation in the two circuits of the linear hydraulic motor and variation of inertia force.

Therefore, after preparation and implementation of all technical conditions necessary for the operation of this experimental stand, it proceeds as follows:

- there are placed, successively, different known **masses** M, of weight G, on the shaft of the linear hydraulic motor, which represents its load / charge;
- there is performed actuation of the linear hydraulic motor for one, two or three ***up and down***, consecutive cycles;
- there is measured the variation of parameters of interest by acquiring and registering their evolution over time;
- there are analyzed the values and **graphical evolution** of parameters of interest.

## Technical data of the hydraulic operative system with linear motion

The hydraulic operative system with linear motion is an assembly of the linear hydraulic motor and its control servo valve.

Technical data regarding the **linear hydraulic motor MHL:**

- type: bilateral shaft cylinder;
- diameter of the cylinder: 105.4 mm;
- diameter of the rod: 70 mm;
- working stroke: 160,
- working pressure: 250 bar.

Technical data regarding the **servo valve SV:**

- type : MOOG, series: 760;
- pressure: 70 bar;
- rated diameter: 6 mm:
- maximum flow: 40 l / min.

## Technical documentation which formed the basis of testing

Technical documentation which formed the basis of testing was represented by the testing diagram, show in Figure 21 and the Testing Program, as well as documentation for execution of the linear hydraulic motor and technical leaflets for the type MOOG servo valve. Tests were conducted in accordance with the Testing Program, which included the following tests:

- testing the dynamic behaviour for **2 steps of flow**: 12.5 l/min and 27 l/min;
- testing the dynamic behaviour for **3 inertial masses**: 5 kg-Figure 29, 12 kg-Figure 30 and 17 kg-Figure 31.

**Fig. 29** *Loading with mass of 5 kg*

**Fig. 30** *Loading with mass of 12 kg*

**Fig. 31** *Loading with mass of 17 kg*

## 5. EXPERIMENTAL RESULTS

After conducting experiments on the dynamic behaviour of linear operative hydraulic components and systems, along time range, in accordance with the testing program, there were obtained a series of graphical and tabular results regarding variation over time of the main dynamic parameters of the system tested namely:

- variation of inertia force on the rod of MHL, depending on time, Fi = Fi (t)
- variation of stroke of the servo system rod, x=x(t);
- variation of velocity of the servo system rod, v=v(t);
- variation of pressure p1, at the input of MHL, depending on time, p1=p1(t);
- variation of pressure p2, at the output of MHL, depending on time, p2=p2(t);
- variation of temperature at the input of MHL, T = T (t).

In the following part, there be presented some of the obtained results.

**Variation of dynamic parameters at MHL loaded with a mass M = 17 kg**

These variations were obtained at a flow of 27 l /min at the supply pump, i.e. a theoretical velocity of 90 mm/s, over a complete cycle ***down-up-down***, along the MHL maximum stroke of 160 mm. Figure 32 presents graphical variations of dynamic parameters, and Table 1 presents selections from the numerical values of parameters of interest.

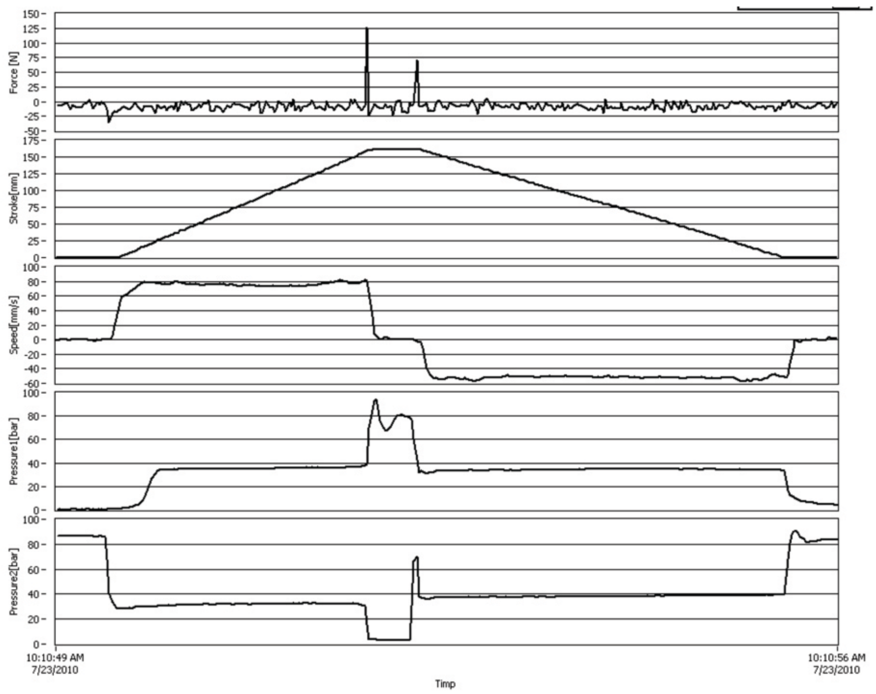



**Fig. 32** *Graphical variations of parameters of interest*

All the experimental results are in concordance with numerical calculus and, also, with the theoretical results obtained by mathematical modeling and computer simulation.

**Table 1**- Selective numerical values

| t[s] | F[N] | P1[bar] | z[mm] | v[mm/s] | P2[bar] | t[$C^0$] |
|---|---|---|---|---|---|---|
| **Start of the lifting stroke** | | | | | | |
| 0.340000 | -5.034031 | 0.355467 | 0.000000 | 0.000000 | 86.076427 | 29.294225 |
| 0.360000 | -13.481552 | 0.428473 | 0.000000 | 0.000000 | 67.580210 | 29.359033 |
| 0.380000 | -34.954094 | 0.474335 | **0.000000** | 0.8 00000 | 41.344455 | 29.296111 |
| 0.400000 | -20.329516 | 0.602873 | 0.000000 | 0.800000 | 33.421124 | 29.280250 |
| 0.420000 | -16.280332 | 0.752547 | 0.000000 | 5.600000 | 31.022063 | 29.291585 |
| **Lifting stroke** | | | | | | |
| 1.640000 | -18.466706 | 35.362931 | 98.464000 | 74.400000 | 31.935266 | 29.447580 |
| 1.660000 | -12.530954 | 35.328498 | 99.856000 | 72.800000 | 31.824621 | 29.381243 |
| 1.680000 | 1.404442 | 35.401431 | **101.360000** | 73.600000 | 32.067160 | 29.397042 |
| 1.700000 | -5.348459 | 35.438831 | 102.832000 | 72.800000 | 32.098029 | 29.397838 |
| 1.720000 | -6.280773 | 35.398317 | 104.288000 | 73.600000 | 32.043370 | 29.429540 |
| **End of the lifting stroke** | | | | | | |
| 2.360000 | -15.600285 | 37.388078 | 155.200000 | 79.200000 | 31.238595 | 29.386209 |
| 2.380000 | -7.856575 | 37.944064 | 156.832000 | 80.000000 | 30.634632 | 29.424930 |
| 2.400000 | 125.998555 | 39.070285 | **158.512000** | 81.600000 | 15.647357 | 29.428723 |
| 2.420000 | -24.261699 | 70.179232 | **160.080000** | 80.800000 | 3.488398 | 29.529842 |
| 2.440000 | -4.489268 | 93.178331 | 160.512000 | 32.800000 | 3.478626 | 29.557835 |
| 2.480000 | -11.710154 | 76.583914 | 160.528000 | 0.000000 | 3.110234 | 29.459419 |
| **Descending stroke** | | | | | | |
| 3.940000 | -11.887470 | 34.214768 | 102.352000 | -50.400000 | 37.520861 | 29.410242 |
| 3.960000 | -15.622227 | 34.208431 | 101.312000 | -51.200000 | 38.066567 | 29.349709 |
| 3.980000 | -6.423368 | 34.317335 | **100.320000** | -51.200000 | 37.897563 | 29.431761 |
| 4.000000 | -9.684647 | 34.377044 | 99.296000 | -50.400000 | 38.017337 | 29.393019 |
| 4.020000 | -3.169395 | 34.408986 | 98.272000 | -50.400000 | 37.984599 | 29.400625 |
| **End of the descending stroke** | | | | | | |
| 5.860000 | -12.796027 | 33.790251 | 2.752000 | -49.600000 | 39.420944 | 29.382228 |
| 5.880000 | -2.957341 | 33.587682 | 1.712000 | -50.400000 | 39.724459 | 29.416214 |
| 5.900000 | -7.050394 | 33.060744 | 0.640000 | -50.400000 | 40.394091 | 29.362616 |
| 5.920000 | 1.270988 | 16.941921 | **0.000000** | -52.800000 | 67.962844 | 29.307363 |

## 5. CONCLUSIONS

The research of the dynamic behavior of the modern servo-systems, based on linear hydraulic motor, was made by using the modern method of analysis and synthesis, by mathematical modeling and computerized simulation. The theoretical results are of high scientific interest and with wide possibilities of capitalization in the practical activity of research and design.

Although the theoretical results were validated logically and dimensionally, on the base of the known data, it has been necessary to develop, also, an experimental research which offered the comparative data, in order to obtain a complete validation of the mathematical model and for improving the capability of description of the mathematical models.

The comparative analysis of the theoretical and experimental results highlights the very close values, which leads to validating of the elaborated mathematical modeling.

Tests have validated the constructive solutions chosen and the proposed test method for determining the main factors influencing the dynamic behaviour of linear hydraulic motors.

By interpreting the graphical evolutions of the parameters of interest, there may be acquired significant knowledge in what regards the dynamic behavior of the linear hydraulic motor, which cannot be revealed by traditional methods. The simulation methods achieved offer the possibility of optimizing the dynamic regimes of the servo-systems, on the base of detecting the parameters with behavioral sensitivity, for obtaining prompter and more accurate responses, in complete accordance with the drive requirements related to a certain technological model of manufacture, in the conditions of minimum energetic consumption, precisely quantified and controlled in their development.

## REFERENCES

[1] D. Prodan, C. Cristescu, D. Popescu, *Masini-unelte* şi prese. Elemente hidrostatice (Machine tools and presses. Hydrostatic elements). PRINTECH Printing House, Bucharest, 2001.

[2] Marin V., Marin Alex. *Sisteme hidraulice automate–constructie reglare exploatare* (Automatic hydraulic systems-building control operation)–, Technical Printing House, Bucharest, 1987.

[3] Oprean, A., Ispas C., Ciobanu E., Dorin Al., Medar S., Olaru A, Prodan D., Acționări şi automatizări hidraulice, Modelare, simulare, încercare *(Hydraulic drive and automation. Modeling, simulation, testing)*, Technical Printing House, Bucharest, 1989.

[4]. *C. Calinoiu, Senzori si traductoare (*Sensors and transducers), volume I, Technical Printing House, Bucharest, 2009.

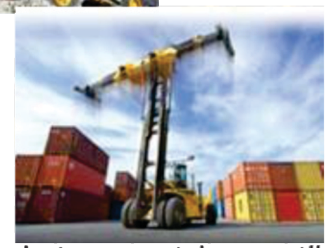
[5] xxx. Simulink, A Program for Simulink Dynamic Systems, User's Guide, The MATH WORKS Inc., Masschusets, 1993.

[6] C. Velescu, Aparate si echipamente hidrostatice proportionale. MIRTON Printing House, Timisoara, 2003, ISBN 973-661-176-0.


SMC
supporting automation
11.000 produse de bază - 610.000 variante constructive
Grupul SMC Corporation (Japonia), prin SMC Romania oferă echipamente de cea mai înaltă calitate pentru automatizări industriale. Gama de fabricație este completată și diversificată continuu prin dezvoltarea de echipamente hidraulice și electrice destinate automatizării proceselor tehnologice din industrie.
CILINDRI HIDRAULICI
Cilindri standard (cu tiranți)
Diametre: 32 ÷ 160
Curse: 25 ÷ 1800
Presiuni nominale: 35 ÷ 140 bar
Cilindri rotunzi
Diametre: 20 ÷ 40
Curse: 25 ÷ 800
Presiuni nominale: 35 ÷ 70 bar
Cilindri compacți
Diametre: 20 ÷ 100
Curse: 5 ÷ 100
Presiuni nominale: 35 ÷ 160 bar
ECHIPAMENTE HIDRAULICE
Finețe de filtrare (μm)
5; 10; 20
Filtru de retur vertical
Filtru de ulei
Filtru de retur
Separator magnetic
Finețe de filtrare (μm)
74; 105; 149
Filtru de aspirație vertical
Filtru de aspirație tip casetă
Filtru de linie
Transformă presiunea aerului în presiune hidraulică
Cuprinde aparatura de reglare
Unități pneumo-hidraulice
Răcitoare de ulei cu apă
SMC Romania S.r.l.
Str. Frunzei 29, sector 2, Bucuresti Tel: 021-3205111; 3261486/87/88; Fax: 021-326.14.89
smcromania@smcromania.ro http://www.smcromania.ro

# Hidraulica mobila Parker

***Liderul mondial in tehnologia miscarii***



Dorim sa va supunem atentiei ***gama completa de solutii Parker*** pentru aplicatiile dumneavoastra din ***hidraulica echipamentelor mobile***, si nu numai:

- **Proiectare** sisteme si echipamente hidraulice si pneumatice
- **HTC – Hydraulic Technology Center –** Centru de expertiza hidraulica (proiectare, montaj, reparatii, executie)
- **Mentenanta – diagnosticarea** echipamentelor hidraulice si pneumatice
- **Garantie si post-garantie**
- **ParkerStore®** retea nationala de magazine cu numeroase repere din gama Parker mereu in stoc (conectori, filtre, furtunuri, valve, si multe altele)
- **In curand - HOSE DOCTOR®** serviciul ***mobil*** Parker de reparatie si inlocuire de furtunuri

Pentru echipamentele dumneavoastra cu elemente de hidraulica mobila va putem oferii:

- **Pompe**
- **Motoare**
- **Cilindrii**
- **Valve**
- **Filtre**
- **Furtunuri hidraulice**
- **Senso Control®** -echipamente de diagnoza a sistemelor hidraulice
- **IQAN®** - sisteme electronice de control la distanta

**Reteaua noastra de distributie acopera intreg teritoriul tarii**, astfel incat lucrand cu noi, veti avea mereu certitudinea ca o echipa inalt calificata de ingineri si tehnicieni Parker va fi langa dumneavoastra 24 de ore din 24 sa va sustina in proiectele pe care le desfasurati.

Pentru orice detalii suplimentare va stam la dispozitie:


**Reprezentanta Parker Hannifin Romania**

Calea Mosilor , nr 260, bl 6, sc A, ap 4, et 1, sector 2, Bucuresti.

Tel/Fax +4 021 252 13 82

office@parker.ro
www.parker.ro
www.parker.com

# CALCULUS OF THE FLOW RATE AND METHOD TO UNCLOGGING LUBRICATION PIPES FROM THE HIDROMECHANICAL EQUIPMENTS ATTACHED TO HYDROELECTRIC POWER PLANTS

**George Cătălin TILEA* Marius GRECU** Cătălin POPESCU*****

(*) Exploitation Manager - Phd. Eng. in Hydro Power Engineering at S.C. Hidroelectrica S.A, Tg-Jiu Subsidiary, 1 Prelungirea Vasile Alecsandri Street, Tg-Jiu, Gorj County, Romania
Catalin.Tilea@hidroelectrica.ro; Mobile +00 40 0731760808
(**) Chief of Motru Hydropower Plant - Phd. Student - Eng. in Hydro Power Engineering at S.C. Hidroelectrica S.A, Tg-Jiu Subsidiary, 1 Prelungirea Vasile Alecsandri Street, Tg-Jiu, Gorj County, Romania ; Marius.Grecu@hidroelectrica.ro; Mobile +00 40 0731760929
(***) Chief of Technical Department - PhD. Student - Eng. in Power Engineering at S.C. Hidroelectrica S.A Bucharest, Constantin Nacu Street, No. 3, Bucharest, Romania
Catalin.Popescu@hidroelectrica.ro; Mobile +00 40 0720727077

## ABSTRACT
In engineering practice, particularly in the exploitation of hydropower plants, for a lot of the hydraulic and mechanical equipments appears the phenomenon of clogging and sedimentation of the underpressure lubrication pipes, in time, flow rate of lubrication pipes changes, coming after a long period of operation a clogging up of 50% or even entirely. In time and due to the corrosive effects, the roughness of the lubrication pipes increases. Such phenomena have been observed at the hidromechanical equipments of the hydroelectric power plants in Tg-Jiu subsidiary with more than 30 years of operation. In the case of quadratic flow losses, calculations of the main parameters of the hydraulic lubrication pipes (roughness and flow rate) have been made. These have confirmed that the flow rate in a steel pipe with a diameter of 15 mm declined considerably. In two years of operation, there has been observed a decrease of the flow rate to one third of its initial value. All the calculations show that the roughness increased from 0.10 mm - at the commissioning date - to 15 mm after 25 years of operation (when the pipe is completely clogged). The idea of making "CLEAN PIPES - facility for cleaning and rehabilitation of lubrication pipes transportation capacity" appeared as a current necessity for proper functioning and safe operation of the hidromechanical equipments attached to hydroelectric power plants (for example: plane gates, ball valves, butterfly valves, etc.). "CLEAN PIPES” installation is used to clean pipes and it's made of: a single-piston displacement volumetric pump (VOLUMETRIC PUMP), an air compressor (LPK), directional valves (DV1, DV2, DV3 and DV4), filters, valves (V1, V2, V3 and V4) connectors (QUICK MUFF), a manometer (M), an oil tank (OT) and a vaseline tank (VT). "CLEAN PIPES" uses a pneumatic adjustable working pressure of the oil up to 400 bars. The compressor has a pressure regulator which enables the control of the pressure for the volumetric pump with a single piston (between 1-10 bars). After preparation "CLEAN PIPES" and making all the connections to the hydraulic circuit, the compressor (LPK) is set for a low working pressure and the cleaning of the pipes begins by increasing lubrication oil pressure. The unclogging of the pipes is complete when a full transit of the oil through the whole circuit is made and when the R4 valve (open position/pump off) returns water from the lake. The used oil can not be recovered from the circuit but it's retrieved downstream of the dam by using absorbent dams. "CLEAN PIPES" is a very efficient method to unclogging of the lubrication pipes and at the same time can be used for lubrication of many hidromechanical equipments. (VT vaseline tank is used for this case).
Compared with other methods used in the filed, the method to unclogging it's an original one, very efficient and it uses a small budget - of maximum 1000 EURO.

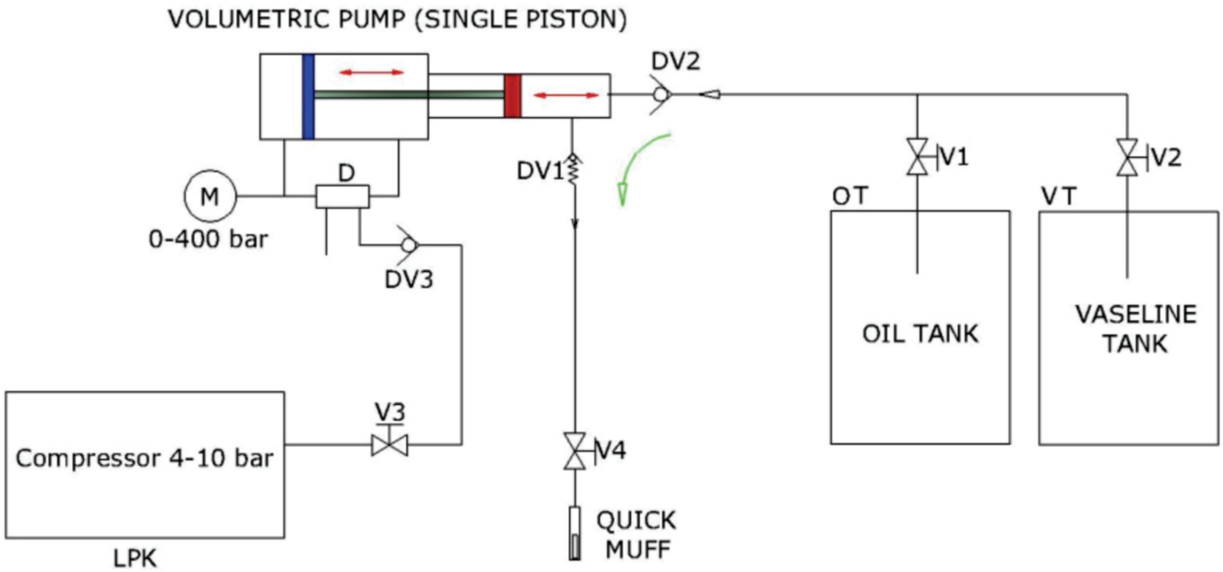
***Keywords:*** hydraulic pressure, roughness, flow rate, quadratic flow losses, volumetric pump, operation period, hidromechanical equipment.

## INTRODUCTION

In engineering practice, particularly in the exploitation of hydropower plants, for a lot of the hydraulic and mechanical equipments appears the phenomenon of clogging and sedimentation of the underpressure lubrication pipes, in time, flow rate of lubrication pipes changes, coming after a long period of operation a clogging up of 50% or even entirely. In time and due to the corrosive effects, the roughness of the lubrication pipes increases. Such phenomena have been observed at the hidromechanical equipments of the hydroelectric power plants in Tg-Jiu subsidiary with more than 30 years of operation.

The main technical parameters [3] of the installation are:

- Compression rate of the volumetric pump (40:1);
- Minimum working pressure of air control: 4 bar; Maximum working air pressure of control: 10 bar;
- H9 hydraulic oil /LiC2 grease;
- Degree of filtration filters is 40-150 m µm;




Scheme of "**CLEAN PIPES**" installation

## DESCRIPTION OF THE "CLEAN PIPES" INSTALLATION AND FUNCTIONING

"CLEAN PIPES" installation is used to clean pipes and it's made of: a single-piston displacement volumetric pump (VOLUMETRIC PUMP), an air compressor (LPK), directional valves (DV1, DV2, DV3 and DV4), filters, valves (V1, V2, V3 and V4) connectors (QUICK MUFF), a manometer (M), an oil tank - H9 type (OT) and a vaseline tank - LiC2 type (VT).

The rapid connection valves are connected to the lubrication point then the hydraulic oil tank is selected and the compressor is started with a minimum operating pressure of 4 bar – fitted for a working pressure of 40 bars for hydraulic oil. With a gradually increase of the pressure of 50 to 50 bar every 2-3 minutes, the pressure remained constant at each growth stage [3].

In the clogged hydraulic circuit pulses are produced that lead to the cleaning the flow circuit. The unclogging of the pipes is complete when: we have a full transit of oil through the whole circuit and when valve R4 (on open position/pump off) returns water from the lake.

After the unclogging of the lubrication the installation is switched on the tank of grease and grease is inserted in the whole circuit. Used oil can't be recovered from the circuit, but this is collected downstream of the dam by using absorbent dams. "CLEAN PIPES" is a very effective method is used to grease the unclogging of the lubrication pipes and at the same time can also be used for lubrication of various hidromechanical equipment (in this case the grease tank is used) [3].

## CALCULATION OF TRANSPORT CAPACITY OF THE LUBRICATED PRESSURE PIPES

While in operation, carrying capacity of pipes is modified, reducing in some cases (such as pipes that use lubricating grease lubricant) to 50% or more of its initial value. Due to the corrosive effects, in time the lubrication pipe roughness increases.[2].

Absolute roughness, after **t** years of operation, can be expressed by the mathematical formula:

$$k_t = k_0 + \alpha\, \mathrm{t} \qquad (1)$$

where: $k_t$ is the absolute roughness in **mm**, after **t** years of operation, $k_0$ is the absolute roughness in **mm**, for the commissioning of pipelines, α is a factor that characterizes the roughness growth rate in **mm/year**, and **t** is the operation period expressed in years.

The value of a depends on pipe material and physical-chemical agent movement through the pipeline.

In the calculation presented in the article lubrication pipes are made of OL37k with an equivalent absolute roughness of 0.1 mm. The pipes are designed to carry greasing agent (Vaseline) on the construction of various hidromechanical equipments: planes valves guiding, butterfly valves or ball valves. Grease is highly mineralized, corrosive, and in time can favor deposits in pipes, especially because the lubrication process is a continuous phenomenon.

Rate flow of the pipeline lubrication is 0.6 l/ s. The nominal diameter of the pipe is considered d=15 mm.

Next, the **flow rate** after 2 years of operation, 4 years of operation, 10 years of operation, 20 years of operation and 25 years of operation is determined.

For grease will be considered the vaseline (choose spreadsheet after [1]) α **= 0,6 -0,7 mm,** also $\mathbf{k_0}$**= 0,1 mm.**

According to equation (1) we get:

- After **t=2** years of operation:

$$k_2 = k_0 + \alpha\, \mathrm{t} = 0{,}1 + 0{,}6 \cdot 2 = 1{,}3 \text{ mm}$$

- After **t=4** years of operation:

$$k_4 = k_0 + \alpha\, \mathrm{t} = 0{,}1 + 0{,}6 \cdot 4 = 2{,}5 \text{ mm}$$

- After **t=10** years of operation:

$$k_{10} = k_0 + \alpha\, \mathrm{t} = 0{,}1 + 0{,}6 \cdot 10 = 6{,}1 \text{ mm}$$

- After **t=20** years of operation:

$$k_{20} = k_0 + \alpha\, \mathrm{t} = 0{,}1 + 0{,}6 \cdot 20 = 12{,}1 \text{ mm}$$

- After **t=25** years of operation:

$$k_{25} = k_0 + \alpha\, \mathrm{t} = 0{,}1 + 0{,}6 \cdot 25 = 15{,}1 \text{ mm}$$

The previous calculations show a complete clogging of the pipes after 25 years of operation. In the case of quadratic miscarriages we use relation (2) to determinate the transport flow of the pipeline:

$$\frac{\lambda_2}{\lambda_0} = \left(\frac{k_2}{k_0}\right)^{0.25} \qquad (2)$$

After **2 years of operation** obtained

$$\lambda_2 = \lambda_0 \cdot \left(\frac{k_2}{k_0}\right)^{0.25} = \lambda_0 \cdot \left(\frac{1{,}3}{0{,}1}\right)^{0.25} = 1{,}9\lambda_0 \rightarrow$$

$$\frac{Q_2}{Q_0} = \frac{C_2 \cdot \sqrt{Ri}}{C_0 \cdot \sqrt{Ri}} = \sqrt{\frac{\lambda_0}{\lambda_2}} = \sqrt{\frac{\lambda_0}{1{,}9\lambda_0}} = \sqrt{\frac{1}{1{,}9}} = 0{,}725$$

than:

$$Q_2 = 0{,}725 Q_0 = 0{,}725 \cdot 0{,}6 = 0{,}435 \text{ l/s}$$

So carrying capacity of lubrication pipes decreased with

$$\frac{0,6-0,435}{0,6}\cdot 100 = 27,5\%$$

after only two years of operation.
After **4 years of operation** obtained:

$$\lambda_4 = \lambda_0 \cdot \left(\frac{k_4}{k_0}\right)^{0.25} = \lambda_0 \cdot \left(\frac{2,5}{0,1}\right)^{0.25} = 2,24\lambda_0 \rightarrow$$

$$\frac{Q_4}{Q_0} = \frac{C_4 \cdot \sqrt{Ri}}{C_0 \cdot \sqrt{Ri}} = \sqrt{\frac{\lambda_0}{\lambda_4}} = \sqrt{\frac{\lambda_0}{2,24\lambda_0}} = \sqrt{\frac{1}{2,24}} = 0,67$$

So carrying capacity of lubrication pipes decreased with

$$\frac{0,6-0,402}{0,6}\cdot 100 = 33\%$$

after **four years of operation**.
After **10 years of operation** obtained:

$$\lambda_{10} = \lambda_0 \cdot \left(\frac{k_{10}}{k_0}\right)^{0.25} = \lambda_0 \cdot \left(\frac{6,1}{0,1}\right)^{0.25} = 2,8\lambda_0 \rightarrow$$

$$\frac{Q_{10}}{Q_0} = \frac{C_{10} \cdot \sqrt{Ri}}{C_0 \cdot \sqrt{Ri}} = \sqrt{\frac{\lambda_0}{\lambda_{10}}} = \sqrt{\frac{\lambda_0}{2,8\lambda_0}} = \sqrt{\frac{1}{2,8}} = 0,6$$

then

$$Q_{10} = 0,6Q_0 = 0,6 \cdot 0,6 = 0,36\,\mathrm{l/s}.$$

So therefore that is carrying capacity of lubrication pipes decreased with

$$\frac{0,6-0,36}{0,6}\cdot 100 = 40\%$$

after **10 years of operation**.
After **20 years of operation** obtained:

$$\lambda_{20} = \lambda_0 \cdot \left(\frac{k_{20}}{k_0}\right)^{0.25} = \lambda_0 \cdot \left(\frac{12,1}{0,1}\right)^{0.25} = 3,32\lambda_0 \rightarrow$$

$$\frac{Q_{20}}{Q_0} = \frac{C_{20} \cdot \sqrt{Ri}}{C_0 \cdot \sqrt{Ri}} = \sqrt{\frac{\lambda_0}{\lambda_{20}}} = \sqrt{\frac{\lambda_0}{3,32\lambda_0}} = \sqrt{\frac{1}{3,32}} = 0,55$$

then

$$Q_{20} = 0,55Q_0 = 0,55 \cdot 0,6 = 0,33\,\mathrm{l/s}.$$

So carrying capacity of lubrication pipes decreased with

$$\frac{0,6-0,33}{0,6}\cdot 100 = 45\%$$

after **20 years of operation**.
After **25 years of operation** obtained:

$$\lambda_{25} = \lambda_0 \cdot \left(\frac{k_{25}}{k_0}\right)^{0.25} = \lambda_0 \cdot \left(\frac{15,1}{0,1}\right)^{0.25} = 3,51\lambda_0 \rightarrow$$

$$\frac{Q_{25}}{Q_0} = \frac{C_{25} \cdot \sqrt{Ri}}{C_0 \cdot \sqrt{Ri}} = \sqrt{\frac{\lambda_0}{\lambda_{25}}} = \sqrt{\frac{\lambda_0}{3,51\lambda_0}} = \sqrt{\frac{1}{3,51}} = 0,53$$

So carrying capacity of lubrication pipes decreased with

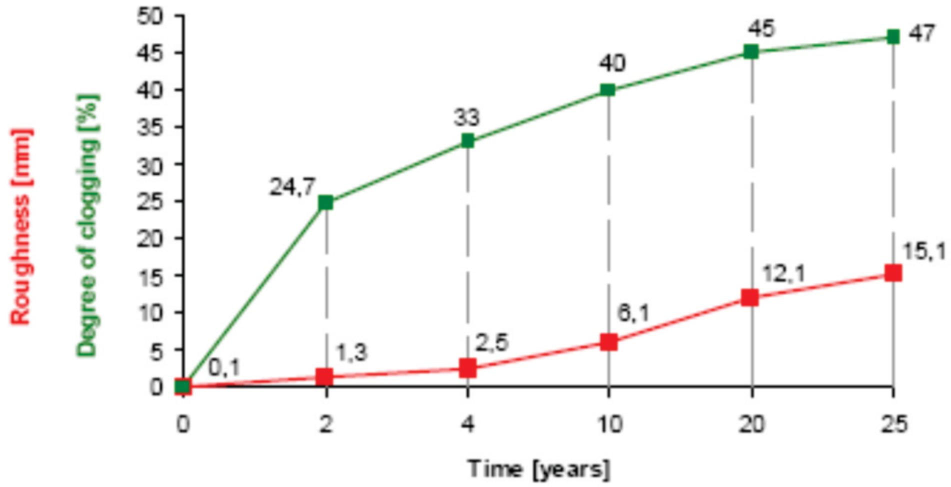
$$\frac{0,6-0,32}{0,6}\cdot 100 = 47\%$$

after **25 years of operation**.

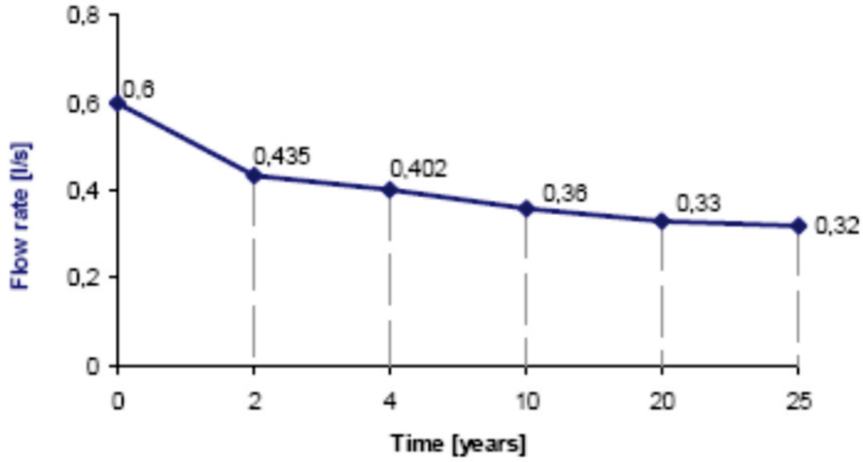
The results are centralized in **Table No. 1** and presented in **Graph No. 1** and **Graph No. 2.**

| Time | roughness | flow | degree of clogging |
|---|---|---|---|
| *years* | *mm* | *l/s* | *%* |
| *0* | *0,1* | *0,6* | *0* |
| 2 | 1,3 | 0,435 | 24,7 |
| 4 | 2,5 | 0,402 | 33 |
| 10 | 6,1 | 0,36 | 40 |
| 20 | 12,1 | 0,33 | 45 |
| 25 | 15,1 | 0,32 | 47 |

**Table No.1 Hydraulic parameters calculated**




**Graph No. 1 Variation of the roughness and the degree of clogging for lubrication pipes during operation period**




**Graph No. 2 Flow rate variation during of the operation period**

## RESULTS AND CONCLUSIONS

From calculus, the assumptions considered here, show a decrease of the transport capacity with 40% after 10 years of operation. After 25 years of operation the pipes could be completely clogged. It is to be observed that after the first two years of operation the flow rate decreases by a third of its original value.

As recommendations we mentioned:

a) for proper functioning and safe operation of installations of the hidromechanical equipment, the unclogging of the pipes should be done regularly, **annually** is recommended.

b) "**CLEAN PIPES**" it's very effective for the unclogging lubrication pipes and can be used for lubrication in various hidromechanical installations.

c) Using a PLC, "CLEAN PIPES" can be integrated into a scheme of automatic continuous lubrication, at defaults intervals by the human operator;

d) "CLEAN PIPES" it's the optimal solution for the lubrication pipes embedded in concrete and for the unreachable ones of some complex equipment.

e) The benefits of "CLEAN PIPES are obvious and clear, to be mentioned: high reliability at a budget of up to 1000 Euros.

## BIBLIOGRAPHY

[1] ALTSCHULL, A. D., *Ghidravliceskiepoteri na trenie v truboprovodah,M., Gosenergoizdat,* 1964.

[2] KISELEV *Hydraulic Calculus*, Ed. Technical, Bucharest, 1988; pp 101–102.

[3] S.C. HIDROELCTRICA S.A. – Norme interne de exploatare a instalatiilor hidromecanice afernte constructiilor hidroenergetice, Tg-Jiu, 2010.

Str. Dr. C-tin Istrati nr. 1, Sector 4, Cod 040542, BUCURESTI
Tel: 021.336.23.52; 021.335.09.85; 021.335.10.50; 0722.562.899;
Fax 021.337.24.60; Email: hesper@hesper.ro http:\\www.hesper.ro

RO 385527
J40 / 7640 / 1991
**CERTIFICAT ISO 9001: 2008**

QMS Certification
No of Certificate 325-3

Ne face placere sa va readucem in atentie HESPER SA Bucuresti, unul dintre principalii producatori romani de echipament, aparatura si instalatii hidraulice din Romania. Fontata in 1877 firma are o indelunga traditie in productia industiala, de-a lungul timpului si in prezent atat produsele cat si beneficiarii fiind o garantie a calitatii si competitivitatii pe orice piata.

HESPER lucreaza in conformitate cu normele de asigurare si control al calitatii ISO 9001:20080 si cu normele de securitate a muncii OHSAS 18001, fiind certificata de organele abilitate interne si internationale.

Principalele produse oferite de HESPER clientilor sunt:

ELEMENTE DE HIDRAULICA

- Pompe si motoare cu roti dintate,
- Divizoare de debit,
- Servodirectii si motoare hidraulice orbitale
- Pompe si electropompe de combustibil
- Instalatii hidraulice simple sau complexe
- Cuplaje si repere mecanice

**Lista de clienti, confirma faptul ca produsele HESPER sunt solutii bune pentru**
utilizatorii de hidraulica din intreaga lume, firme din industria constructiilor de masini, producatori de masini agricole si forestiere, utilaje grele, masini-unelte, sisteme elevatoare, industria extractiva, industria siderurgica .

Date suplimentare despre firma si produsele noastre se pot gasi pe site-ul nostru www.hesper.ro.

# ECOLOGICAL IRRIGATION OF DAMBOVITA HYDROTECHNICAL ENGINEERING ARRANGEMENT

[(1)]Constantin NICOLESCU, [(1)] Gheorghe ŞOVĂIALĂ, [(2)] Lucian ILIESCU, [(3)]Dan PREPELIŢĂ, [(1)] Ioan LEPĂDATU, [(1)] Gabriela MATACHE, [(1)] Alexandru MARINESCU

[(1)] National Institute of Research and Development for Optoelectronics (INOE 2000 - IHP) – Hydraulics and Pneumatics Research Institute, Bucharest, cons_nicolescu@yahoo.com
[(2)] The Research and Development Institute for Horticultural Products Processing and Marketing of the Horticultural Products " HORTING " Bucharest, lucian20051@yahoo.com
[(3)] Faculty of Hydrotechnique, Technical University “Gh.Asachi” Iasi, predans@yahoo.com

**Abstract**
*The article presents the method for calculating the technical elements of the ecologic irrigation performed by underground watering of the sod and sprinkling of the ivy. The two plants have different watering and ecological needs and have a support made of floating panels. These are made for both high and low river banks. The cultures are interlaid, each panel being watered separately through the supply junctions and the same water supply pipeline. The soil stratum is thin and the passing of the pipes from a bank slope to another is made in the zones with proper conditions (bridges, hydrotechnical junctions or other existent engineering). It is required to maintain a constant pressure in the supply network and a low water and electricity consumption. The pressure is set automatically by means of a water pumping system provided with a reservoir with a component part made of rubber in which is maintained air supply. The prompt is given from a regulation device fixed on the reservoir. The technical engineering arrangement was performed for a waterstream of 500 m and for the whole area Lacul Morii Vitan Barzesti, which represents the urban flowing zone. It derives from the concept that irrigation is an ecological engineering activity.*

**Keywords:** river flow, Dambovita, irrigation, technical elements, pilot engineering arrangement

## 1. Introduction

Potable water used for the various human needs represents less than 1% from the entire global water resources. The allotment of the scarse water resources is not made in accordance with the needs. For example Europe which with 20% of the world s population benefits of only 7% of the global potable water resources. From the entire amount of water used on global scale 70% is used in irrigations.

The studies made in the world during the interval 1881-2005 emphasize the global atmosphere heating phenomenon. There are taking place short or long term droughts, being required to use irrigation. According to the specialists estimations about 2% from the entire agricultural land of 15 mil.ha is affected annually by extremely severe drought and 28% of severe drought with a frequency of over 40 in 100 years. The vulnerability of these cultures increses in the same time with the raising water demand and it depends on the vegetation phase and the intensity and frequency of the pluviometric disturbances.

## 1. Material and method

*1.1. Sod requirements in relation with the main vegetation factors*

1) The optimum air temperature is different depending on the seed age, species and variety being generally of 15 – 35°C .The min temperature is of 1-3°C but can resist below 0 degrees till -20°C.
2) The transipration coefficient of the sod is 1,5 2 times higher than at cereals, varying between 500-1000 .The differences depend on the stage of growth, cutting time, conditions of growth, variety etc.

3) Relative tolerance at droughts of certain types of sod seeds is excellent for Cynodon dactylon and good for Festuca ovina, Festuca arundinacea or rubra. When sod is short cut the need for nutrients is higher.

*2.2. Technological aspects regarding sod*

a) Sod is a perennial culture with high water needs especially during summer.
b) Sod is planted on a soil with a thickness of min 10 cm reducing the capacity of retaining water and resulting a shorter interval between waterings.
c) Irrigation is performed by sprinkling and by underground watering. Excessive irrigation may have negative impact upon culture, more than lack of humidity, the sod transforming itself into swamp.
d) A recommended type of sod is Poa pratensis with a very good resistance at droughts and a superficial radicular system.
e) The depth recommended for the system is of 10-15 cm for 80-85% of the edaphic volume.

*2.3. Mixed sod varieties*

a) The use of a single variety is generally avoided being preffered the mixed varieties.
b) Are prevalent two varieties Lolium perenne and Festuca rubra.

**The sod for green public space has a very high and varied traffic depending on season: Lolium perenne - 20% and Festuca arundinacea - 80%, Festuca arundinacea - 60% and Cynodon dactylon - 40%.**

c) It is estimated that worldwide there are over 200 different varieties from Holland, Germany Denmark, England, USA, France etc.
d) **In our country** the production of original sod seeds is irrelevant.

Seeds are generally purchased from import companies, without knowing the behavior and needs of these varieties in the conditions offered by our land.

*2.4. Technical elements of the watering*

The watering methods used are underground watering for sod and sprinkling for the ivy. At the underground watering, the watering pipe is provided witj pore like supply elements and at sprinkling with sprinklers placed in derivation. The performances offered which represented the main criteria of selection are the watering capacity of 0,95 at underground watering and 0,90 at sprinkling and the water transport capacity which is of 0,85 for the underground watering and 0,80 for sprinkling. The watered surface is of 6m$^2$ at the panel which is watered from the underground and of 1,12 m$^2$ for the panel watered by sprinkling. Below are presented the main technical elements of the watering process: the gross watering rate, the interval between waterings, the watering time, specific daily water consumption for sod and ivy.

$$m_b = \frac{100 \bullet H \bullet DA(CC - P_{mn})}{\eta_u \times \eta_t} \quad [m^3/ha] \qquad (1)$$

Where H is the watering depth for the radicular profile, expressed in m; DA - the apparent density, expressed in t/m$^3$ ; CC - water amount in the field expressed in % ; CO - soil fading coefficient expressed in % ; Pmin = CO + β(CC-CO); β - coefficient which depends on the soil texture variety ; $\eta_u$ - watering capacitz ; $\eta_t$ - water supply capacity ,on the track pumping system - watering installations.

*2.4.2 Interval between waterings ( Iu )*

It is used the following relation :

$$I_u = \frac{m_b}{q_d} \quad [days] \qquad (2)$$

Where $q_d$ represents the dailz need of the plant in the month with max consumption, in dm$^3$/m$^2$ day;

*1.1.1. Watering time span ( T )*

It is calculated using the following formula>

$$T = \frac{m_b \bullet S}{q_d} \quad [hours] \qquad (3)$$

Where S is the watered surface water being poured through a porous tube or sprinkler expressed in $m^2$.

S = e · d,

e = being the equidistance of the watering pipes expressed in m; d = the distance between distributors on the watering pipe in m; $q_d$ = the flow of the supply element in $dm^3/h$.

*2.5. Solution for watering sod and ivy panels*

*2.5.1. S od panels*

Are used sets of 5 flotable panels covered by sod alternatively with panels covered by ivy. The surface of such a panel is of 6 $m^2$ and the dimensions 2,0 mx 3,0 m. The watering inside panels is performed by means of porous pipes with a DN of 13 mm (1/2") placed in the existent gorges on the panel bottom at a distance of 0,40 m one from the other.

Over the pipes is put a geotextile sheet which is water permeable. Then is laid the soil layer with a thickness of 7 cm, then another geotextile sheet and a sod layer of 4 cm. The radicular system of the sod passes throgh the textile sheet without distroying it.

The lifespan is of 5/10 years , according to HORTING Bucharest.

*2.6.2. Ivy panels*

A plant is watered from a set of 5 sprinklers with a watering capacity of 10 $dm^3/h$, consuming 50 l/h at the operational pressure of 1 bar.

The watering pipe is made of PEJD with a DN of 16 mm and a length of 10 m. The lifespan is estimated at 20/25 years, according to HORTING Bucharest.

## 3. Obtained results

*3.1 )* Values used for the performed operation regarding the technical elements of the watering: H=0,11 m, DA - 1,25 $t/m^3$; CC - 22%; Pmin - 17%; $\eta_u$ – 0,95; $\eta_t$ – 0,85.

In these conditions by applying the relations (1) ,(2),(3) it results a gross rate of

$$m_b = 85{,}2\ m^3/ha = 8{,}52\ dm^3/m^2 = 8{,}52\ mm.$$

For a flotable panel with a surface of 6 $m^2$ and for reaching the gross rate is required the specific volume Mpg of 51,12 $dm^3$ per panel.

$$M_{pg} = m_b \times S_p \quad [dm^3 / panel] \tag{4}$$

$$M_{pg} = 8{,}52 \cdot 6 = 51{,}12 \qquad [dm^3/ panel] \tag{5}$$

3.2.*)* For $m_b$ = 8,52 $dm^3/m^2$ and q = 1,5 $dm^3/m^2$ · day, it results $I_u$ = 5 days.

The daily consumption is equal at sod and ivy according to Iliescu L, 2010 from HORTING Research Institute from Bucharest.

For the 2 types of supply elements analyzed results:

Porous pipe S= 0,40 x 3,0 = 1,20 $m^2$ ; $q_d$ = 91 $dm^3/h$ ; T= 1,12 ore = 65 min;

Sprinkler S= 1,12 $m^2$ ; $q_d$ = 10 $dm^3/h$; T = 0,95 hours ≈ 1 hour.

*3.3. ) Specific daily consumption of the cultivated species*

| Nr. crt. | Cultivated plant | $\sum(e+t)_{med}$ [$dm^3/m^2 \cdot day$] | Technological specifications | $\sum(e+t)_{max}$ [$dm^3/m^2 \cdot day$] | Mentions |
|---|---|---|---|---|---|
| 1 | Sod | 1,5 | Layer of 4 cm | 1,5 | Local varieties; Poa pratensis; Festuca rubra |
| 2 | Ivy | 1,0-1,4 | 2 m between plants | 1,5 | L = 12m; l = 4 m |

*3.4.) Equipping for the flotable panels covered with sod*

Specific technical elements overall length of a pipe from a panel - 19,6 m; Qm= 91 $dm^3/h$; $q_d$=4,64 $dm^3/h$ x m operational pressure of the porous pipe from the panel is of 1 bar.

*3.5.) equipping for the panel covered with ivy*

It is used a watering pipe made of PEJD with a DN 16 mm, a length of 10 m, provided with sets of 5 sprinklers, placed at equal distance of 2,0 mm, with a flow of the sprinkler of 10 $dm^3/h$. A plant is watered by a set of sprinklers consuming 50l/h at an operational pressure of 1 bar.

*3.6. ) Water supply from a watering installation provided with reservoir*

The variant is Comfort type 5000/5, code 1774, produced by Gardena, the device being provided with rubber element and pneumatic insulation. This ensures a max flow of 4500 $dm^3/h$ (1,25 L/s = 0,00125 $m^3/s$ ) at a pressure of 5 bars, starting from 2 bars on aspiration.. The aspiration and discharge dyameter is of 32 mm.

The consumed power is of 1100 W, the operation being mono phasic.

The electropump starts automatically when watere is required and stops when consumption ceases.

Beside electropump in the installation are included reservoir, against return valve pre filter rotative shut switch, integrated manometer,pressure shut switch mounted in the housing, lid for drainage, protection against temperature excess at the motor.

Other elements: weight 16 kg, max height of aspiration 8 m ( as distance until the water lustre) max temperature of the pumped water 35°C. Water temperature was of 26-28°C in the zone Podul Izvor [1].

The loss of longitudinal load for the main typical dimensions of the pipes of low density, may be determined from the graphic images shown in fig.1.

The supply installation is placed in the hydrotechnical junction, or on the riverbank. Are analyzed 2 kinds of situations : arrangement for the pilot project with a length of 500 m, for each slope(fig.2) and the total arrangement of the urban river flow of Dambovita between Lacul Morii Vitan Bârzeşti, with a total length of 8786 m.

The watering methods and equipping for the pilot project are presented in fig.3.

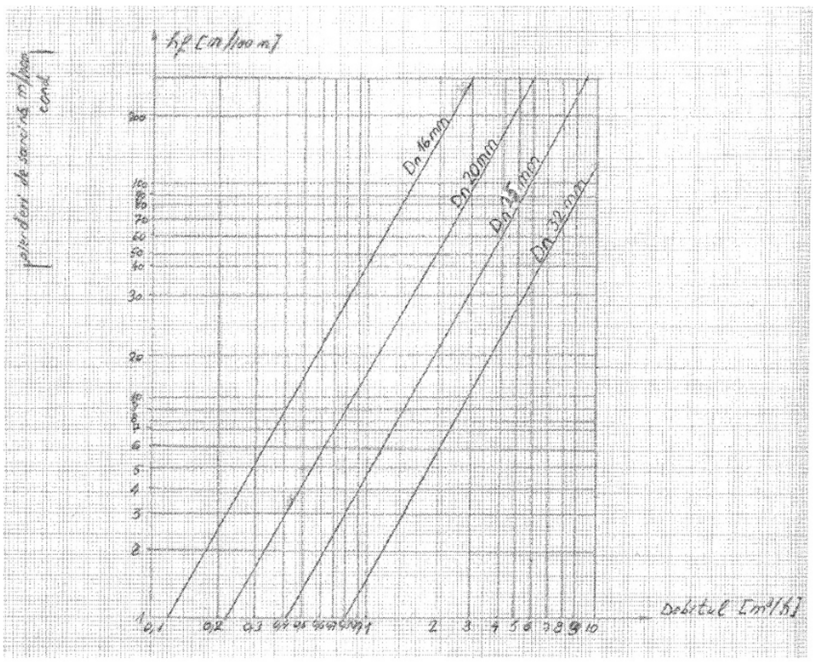



**Fig.1** *Load losses at the watering pipe PEJD*

*3.7.) Irrigation arrangement*

*3.7.1.) Arrangement for the pilot project*

The engineering work has a length of 500 m, in relation with NH Grozăveşti (fig 2.3).

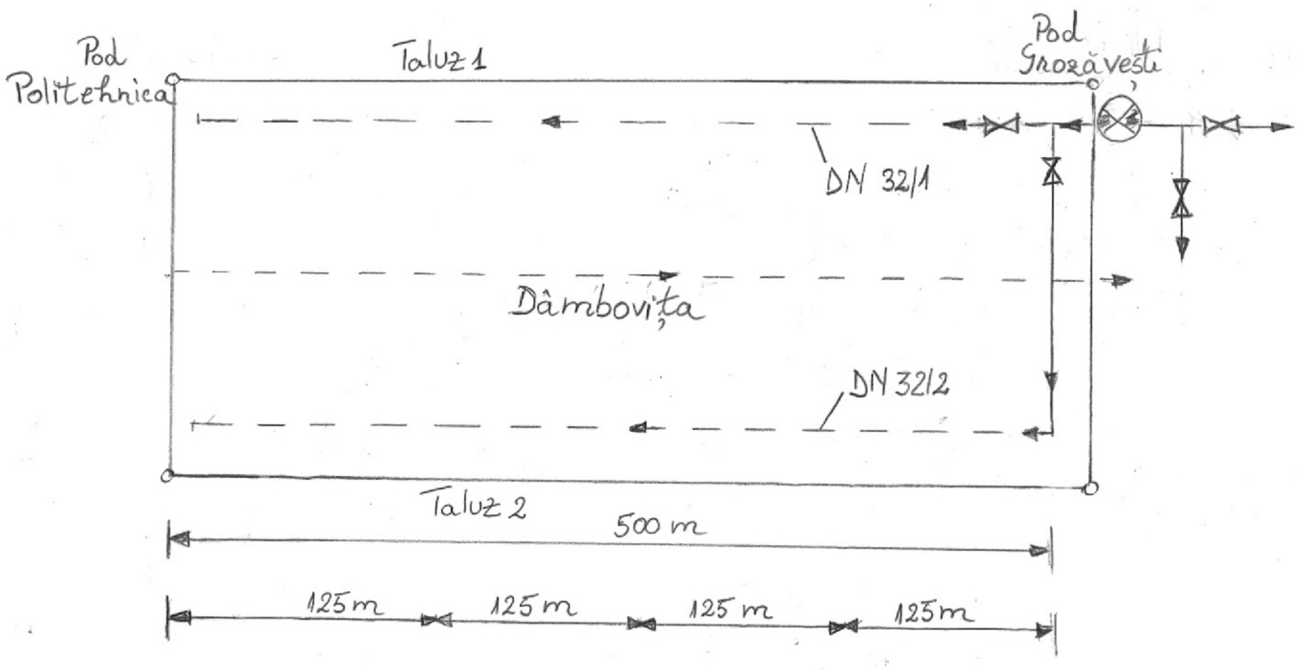



**Fig.2** *Emplacement for the pilot project*

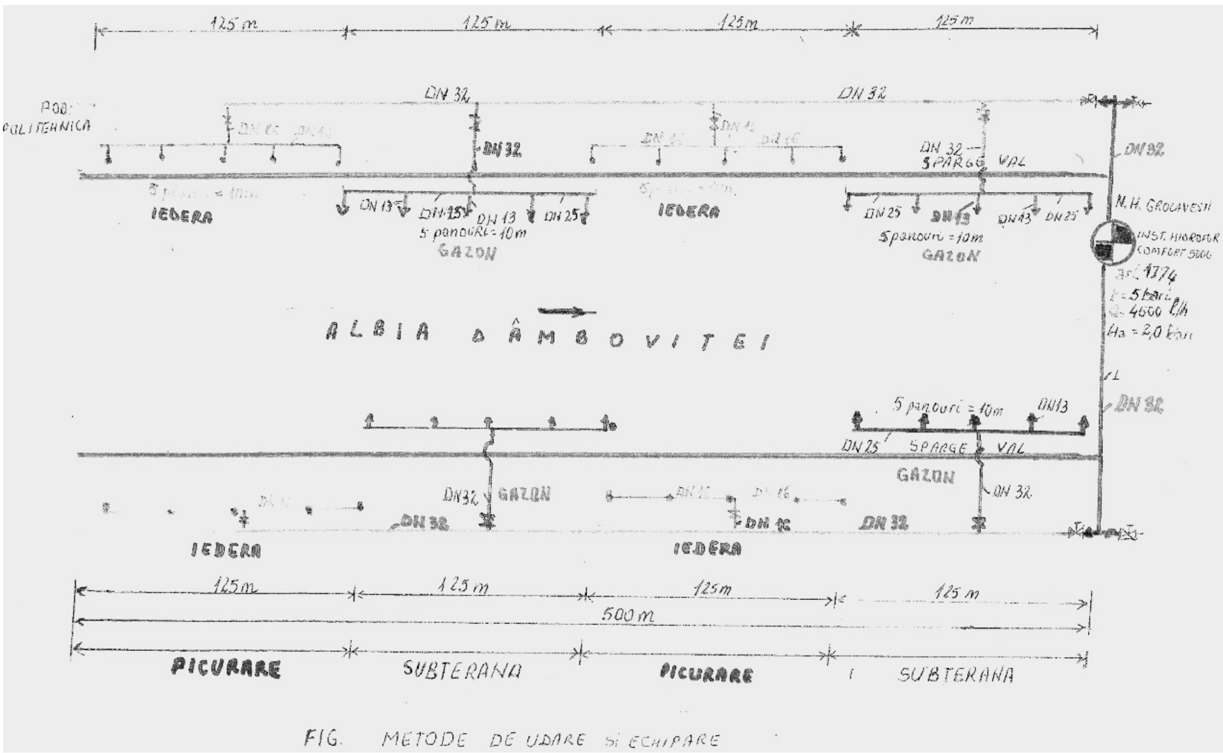



**Fig.3** *Watering and equipping methods*

Each slope is divided in 4 segments of 125 m ,each with 64 panels. On the overall length of the slope of 500 m, are placed 250 panels grouped alternatively (125 panels with sod and 125 panels with ivy). Are placed 5 panels with sod then 5 with ivy.

The capacity of the watering system is of 4500 $dm^3/h$ , supplying simultaneously 64 panels grouped in 13 groups of 5 panels or 128 m as follows:

For $q_{sod\ panel} = 91\ dm^3/h$ ;

$q_{ivy\ panel} = 50\ dm^3/h$ .

$91 + 50 = 140\ dm^3/h$

$4500\ dm^3/h : 140\ dm^3/h$ = 32 groups of 2 panels

32 x 2 = 64 panels= 128 m

Operational conditions:

It works with a segment of 125 m, of 63 panels during an hour.

Time required for watering 8 segments is of 8 x 1h = 8,0 hours.

Watering starts from the upstream edge of the pipes DN 32/1 si DN 32/2.

*3.7.2.) Irrigation arrangement for the entire surface (fig.4)*

*3.7.2.1.) The watering method*

Watering is performed on one side of the hydrotechnical junction, with a length of 500 m and is continued on the other side with a length of 500 m. On each side, are watered the two slopes, by turns, starting with the upstream one. On each slope are put 5 sets, each with a length of 125 m.

In these conditions it results:

500 m : 2m (panel width = 250 panels, from which 125 ivy panels;*pgq* - 91 $dm^3l/h$; *piq*- 50 $dm^3/h$;*pgq*- flow required for watering a sod panel;*piq* -flow required for watering the ivy panel;

The number of sod panels which may be watered in the same time: 32 panels

The number of ivy panels which may be watered in the same time 32 panels

*3.7.2.2.) Watering modality*

1) 500 sod panels, in 8 intervals ,with 125 m on each interval, with a flow of 91 $dm^3/h$ for each panel, operational pressure of 1 bar, watering time an hour.

2) 500 ivy panels with a length of 1000 m, (500 on the left and 500 on the right ), with a watering flow for a panel of 50 $dm^3/h$ and 1 bar, pressure during 1 hour time.

*3.7.2.3.) Normogram of the watering for a hydrotechnical junction*

The pumping system is operated 16 h/day ,6 days/month , during 6 months April – September.

The daily water needs is of 72 $m^3$/day, 1 installation x 16 h / day x $4,5m^3h$.

Annual operation of the pumping system 16x6x6=576 h/year.

Energetic consumption 1,1 kW/h x 576 hours/year = 633,6 kW /year.

Energy cost 0,5 lei/kW.

Energy cost = 633,6 x 0,5 = 316,8 lei/year ≈ 73,7 Euro/year. 1Euro = 4,3 lei

*3.7.2.4. ) Land prioritizing as presented in (fig.4)*

*Zone I (5 sod panels 5 ivy panels )*

$I_a$ Pasarela Politehnică –Orhideea = 445 m AP-2

$I_b$ Haşdeu – Izvor = 394 m AP-5; APT-2

$I_c$ Biblioteca Națională (Călcâi) – Podul Mărăşeşti = 562 m AP-6, APT-3

$I_d$ Spitalul Municipal - Pod Operă = 248 m AP-4

**Σ zone I = 1650 m (825 sod panels + 825 ivy panels) (5 AP, 2APT )**

*Zone II (5 sod panels 5 ivy panels )*

$II_a$ Pod Cotroceni – Spitalul Municipal = 393 m AP-3, AP-4

$II_b$ Pod Opera – Haşdeu = 402 m APT-2

$II_c$ Izvor – Calea Victoriei = 638 m AP-5, APT-2

$II_d$ Podul Mărăşeşti - Pod Timpuri Noi = 691 m AP-6, AP-7

**Σ zona II = 2124 m (1062 sod panels 1162 ivy panels) (4 AP, 1 APT)**

*Zone III ( ivy panels )*

$III_a$ Pod Ciurel – Politehnică = 1136 m AP-1, APT-1

$III_b$ Carrefour– Podul Cotroceni = 837 m AP-2, AP-3

$III_c$ Calea Victoriei – Unirea (Tribunal) = 373m APT-3

$III_d$ Timpuri Noi – Vitan Bârzeşti = 2666 m AP-7, AP-8, AP-9, APT-4

**Σ zona III = 5012 m (5012 ivy panels) (6 AP, 3 APT)**

**Total arrangement zones I,II, III = 8786 m**

It is presented for each of the 3 zones : number for each type of panel, pumping units placed in the hydrotechnical junctions AP pumping units placed on the slope riverbank APT

*3.2.2.5 ) Total necessary of water for 1 hydrotechnical junction is of 72 $m^3$/day for 16 hours/day x 4,5 $m^3$ hour.*

*3.7.2.6.)irrigation water needs for the entire arrangement.*

Watering is performed during an hour.

13 installations x 576 hours/yearx1,1,kWx0,5lei kWh=4118 lei/year=957,8 euro/year. From the 13 pumping units 9 are placed in hydrotechnical junctions and 4 on the riverbank ,as it is shown in fig.4. The daily water needs on the entire arrangement is of 936 $m^3$/day (13 units x 4,5m3 / hour x 16h / day ).

## 4.Conclusions

It is proposed a new approach of irrigation as ecological factor and as engineering work as well.

Water as a vegetation factor has a different meaning than the common one taking into account the relation water – soil – plant - climate present in the biochemical and phzsical processes influencing directly the air, water and biologic condition.

An operational arrangement answers, positively to the variable environmental factors, by the changes which lead to behavior variations (the system is very flexible) and the resistance at the respective factors, in certain limits, depending on the safety threshold taken into account.

The climatic hazard is alleviated, the effect of droughts is reduced and the ecologic conditions of the environment are significantly improved.

The irrigation works comprise proper correlations with the environmental conditions and the engineering works previously performed, dammings, drainages, irrigations.

An update, performant irrigation system exploited and maintained properly in accordance with the most recent technological acquirements supports sustainable development and protects the environment.

The irrigation works must be adapted to the new challenges determined by the climatic changes.

The irrigation technique is alternative watering.

Between irrigation and environment exists an ineluctable bond, caused by the thermal island phenomenon on the bed of Dambovita river.

Were determined the technical elements of the irrigation process on flotable panels with underground watering for sod and sprinkling for the ivy.

Water supply of the watering installation is done by a pumping equipment with hydrophore.

Were realized two arrangements for the pilot project with a length of 1000 m and for the entire surface from Lacul Morii to Vitan Bârzeşti with a total length of 8786 m.

## Bibliography

1.- Nicolescu C. , Şovăială Gh. , Marinescu Al., Vişan Alexandra - „Effects of the thermal aisle on the urban flow of Dambovita”, Hervex, 2010.

2.- Nicolescu C. And partners - „Complex use of water in agriculture”, AGIR publishing house, Bucureşti 2008.

3.-Grumeza N. Kleps Cr - „Irrigation work arrangements in Romania” , CERES publishing house, Bucureşti 2005.

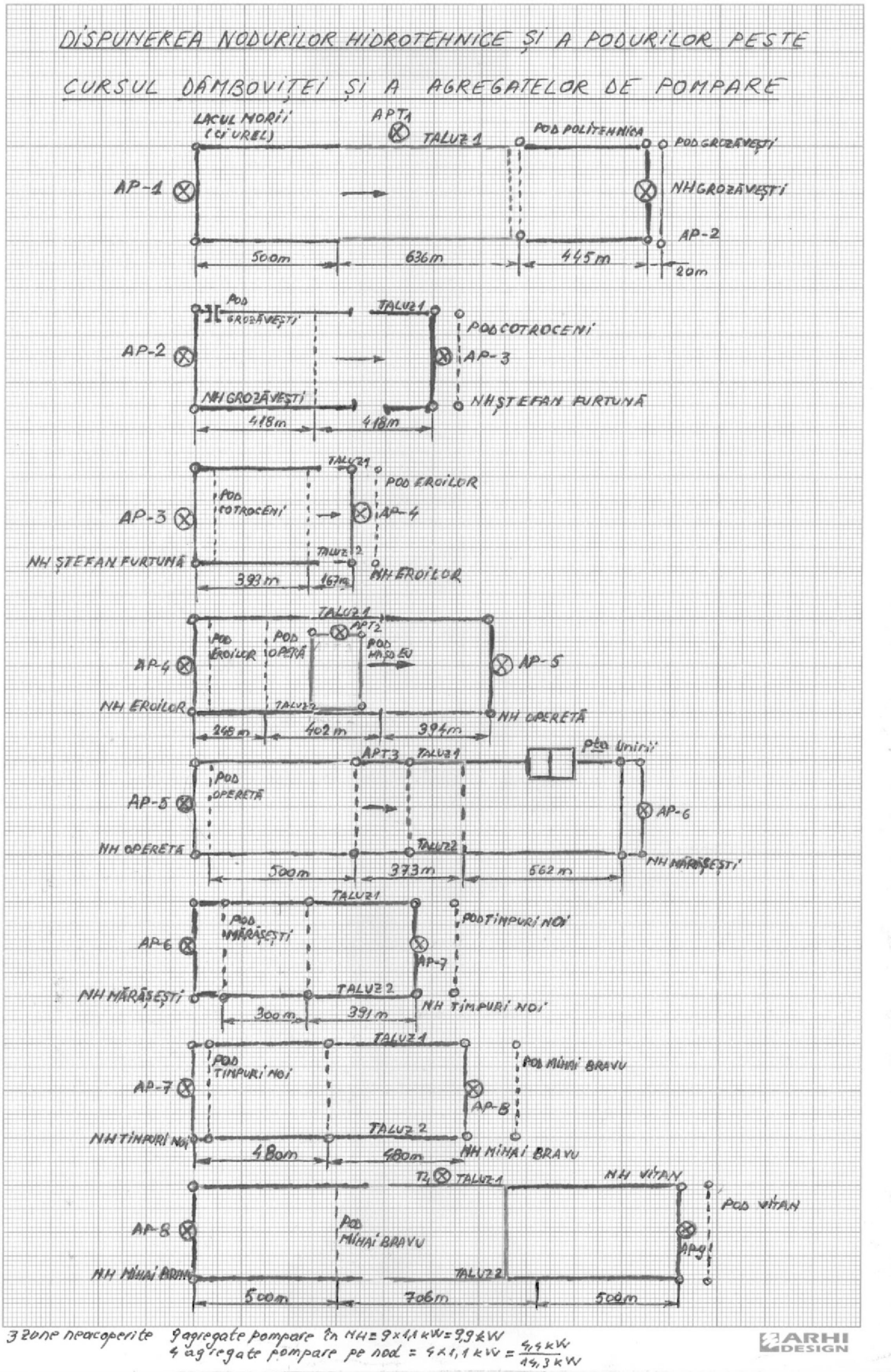


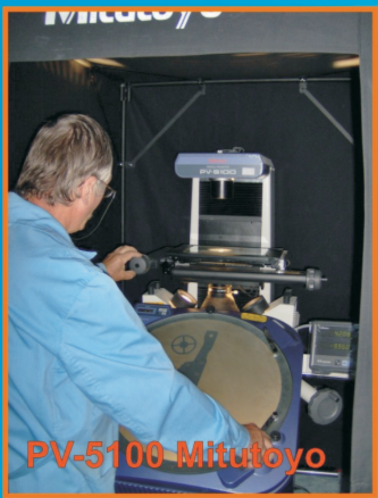

Fig.4 Layout of hydrotechnical nodes and bridges over Dambovita river and of pumping units

# MECATRONICĂ ŞI TEHNICA MĂSURĂRII INTELIGENTE


Laboratorul
Micro-Nanotehnologii
"Rapid Prototyping"


Micro - Laboratorul
Măsurări dimensionale
ultraprecise în 3D


Laboratorul nanometrie
"Microscop de Forță Atomică"





Echipanente mecatronice
HIGHTECH


- SISTEME ŞI ECHIPAMENTE MECATRONICE DE MĂSUTARE
- APARATURĂ INTELIGENTĂ MEDICALĂ
- TEHNOLOGII INOVATIVE
- MERKETING, STRATEGII, TRANSFER TEHNOLOGIC
- CERCETARE DEZVOLTARE ŞI INOVARE
- STUDII, STRATEGII ŞI CONSULTANȚĂ
- ANALIZE, TESTĂRI ŞI CERTIFICARE

# CONSTRUCTIVE AND FUNCTIONAL IMPROVEMENT OF PLANE DISTRIBUTION FOR THE AXIAL PISTONS PUMPS WITH TILT BLOCK AT 40º

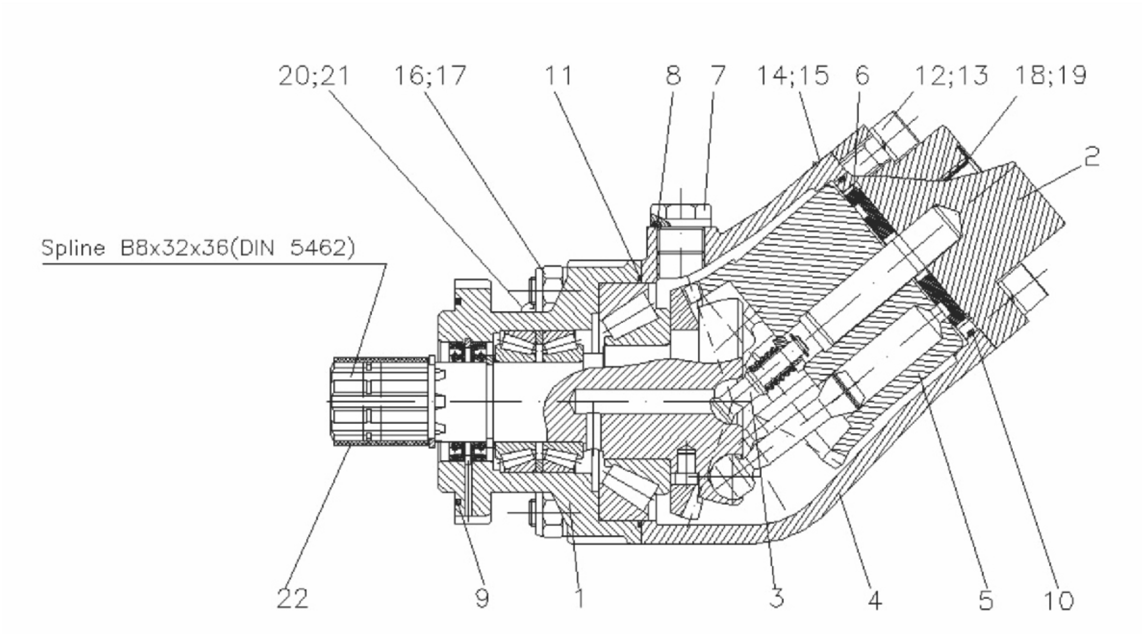
- **Ionel DINU –** General Manager - S.C. HIDRAULICA UM Plopeni S.A.
- **Petru BREAZ** – Technical Manager S.C. HIDRAULICA UM Plopeni S.A.
- **Gheorghe ZAMFIR** – Chief of Technical Department S.C. HIDRAULICA UM Plopeni S.A.

**Abstract**

*This document presents the results of a study about the constructive and functional improvement of the plane distribution for axial piston pumps with tilt block at 40 °, type PPA40, available for mass production at S.C. HIDRAULICA UM PLOPENI S.A..*
*Concerning to the distribution, the pump (Fig. 1) was designed using classical solutions, namely: the distribution plate made of bronze, in combination with a steel cylinder block.*
*The study was done as a result of company objectives, namely:*

- *Improving of functional qualities;*
- *Raise of technical characteristics;*
- *Reduction of production costs.*

**Keywords:** axial piston pump, distribution plate, seal angles




| | |
|---|---|
| 11. „O”- ring | 22. Protection clutch |
| 10. „O”- ring | 21. Safety wire |
| 9. „O”- ring | 20. Stamped seal |
| 8. „O”- ring | 19. Protection plug |
| 7. Threaded plug | 18. Protection plug |
| 6. **Distribution plate** | 17. Plate washer |
| 5. Cylinders block | 16. Nut |
| 4. Pre-tightening assembling | 15. Rivet POP |
| 3. Assembled control valve | 14. Label |
| 2. Assembled distributor | 13. Washer |
| 1. Driving assembly | 12. Screw |

**Fig. 1.** *Axial piston pump with tilt block at 40 °*

## 1. Distribution Slots

The shape and size of distribution slots must ensure the flow of liquid within the characteristic sections, with moderate speeds in order to avoid excessive pressure losses.

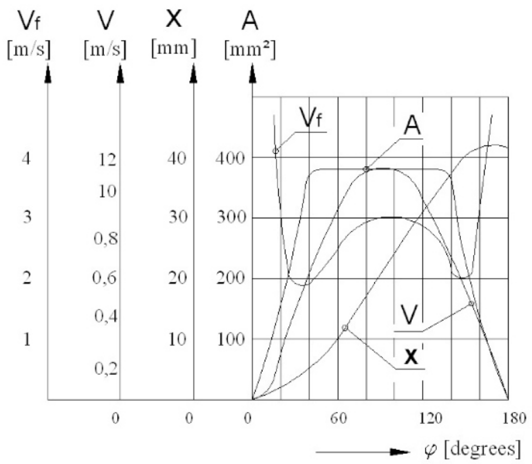



**Fig. 2**. *The variation of stroke and speed of piston, range of the opening of the slot and speed of the liquid within slot, depending on the angle of rotation of the shaft*

The speed $V_r$ is maximum (Fig. 2) at the beginning and at the end of discharge (aspiration) and much higher than the corresponding value of the maximum speed of piston, for $\varphi = \pi / 2$

It is recommended as the flow speed by slots to not exceed 8 m/s. The flow speed by slots is calculated using the following formula:

$$V_{f(\pi/2)} = v_{(\pi/2)} \cdot \left( A_f / A_c \right) = \omega R \sin\alpha \left( A_f / A_c \right)$$

**<u>Observation</u>**: The speed "$V$" of the piston is maximum at $\varphi = \pi / 2$ (Fig. 2)

$V_{\max} = \omega R \sin\alpha$, - practical is recommended to not be exceed the value of 4 m/s

$A_c$ - area of the cylinder (Fig. 3)

$A_f$ - area of the slot (Fig. 3)

Report $A_f / A_c$ for the usual constructions is between 0.42 and 0.48.

The average diameter of the slots, Do (Fig. 3), is required from constructive point of view and is generally equal with the diameter of the cylinders layout.

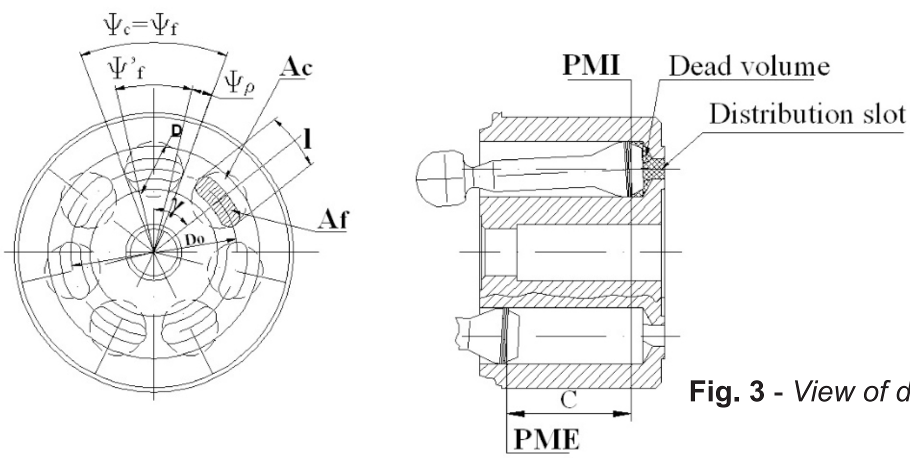



**Fig. 3** - *View of distribution area*

To ensure sealing of the cylinders by the distributor, in the vicinity of the pistons dead points, is required a distribution with "positive coverage" expressed by the condition: $\psi_e - \psi_f > 0$

At the setting of the seal angles $\psi_{a1}$ and $\psi_{a2}$ we had in consideration several reasons, like:

- <u>small values</u> of these angles alter the <u>volumetric efficiency</u> but decrease <u>the operating noise</u>
- <u>high levels</u> of sealing angle $\psi_{a1}$, will increase the volume of liquid enclosed in the cylinder, causing <u>a pressure drop in the cylinder</u> (Fig. 5; Fig. 6), thus leading to the <u>phenomenon of cavitation</u>.

To <u>avoid cavitation</u> at the beginning and end of aspiration, it is necessary:

- the speed limitation;
- the supercharging pump.

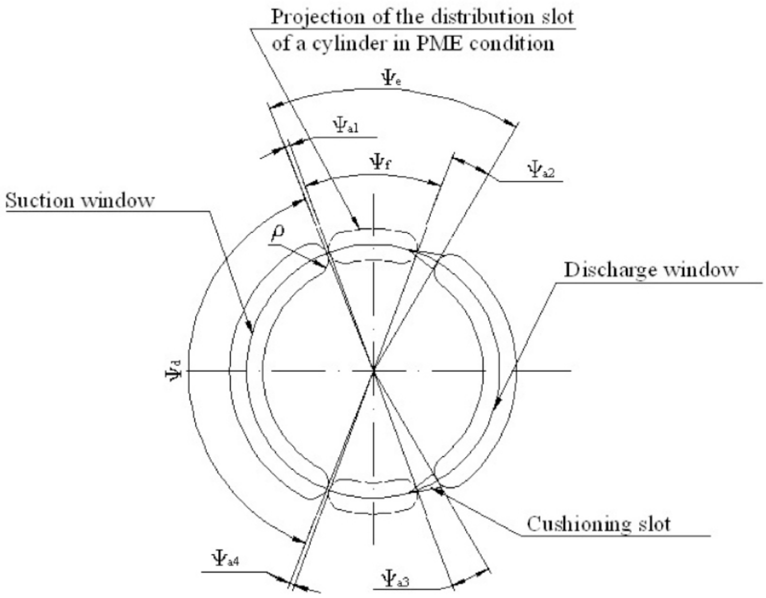



**Fig. 4** – *The plane distributor of the pump*

In this case it should be as:

$$\psi_{a1} = \psi_{a2} = \psi_{a3} = \psi_{a4} = \psi_a = 0,5...2^{\circ}$$

According to Fig. 4, at the rotation of the cylinder block with angle $\psi_{a1}$, as stated above, the phenomenon of cavitation appears, by the pressure drop in the cylinder (Fig. 6). The connecting of the cylinder at the discharge window brings to a rapid pressure increase, which after several high-frequency oscillations, practically reaches an average value.

Closing again of the cylinder, when the slot becomes again outside tangent at the discharge window has as consequence the increasing of pressure until the piston reaches the PMI. This happens after the rotation with angle $\psi_{a3}$.

So, we can see that <u>during the connection of the cylinders</u> at the two distribution windows (Fig. 5 and Fig. 6), <u>shock waves of high frequency</u> are generated, vibrations and liquid leaks, with high speeds, resulting in erosion phenomena due to abrasive particles in the liquid. For attenuation of these phenomena, the windows of distributor are provided at the ends with triangular chamfers (Fig. 7), ensuring the progressive connecting and disconnecting of cylinder. The dimensions are determined empirical.

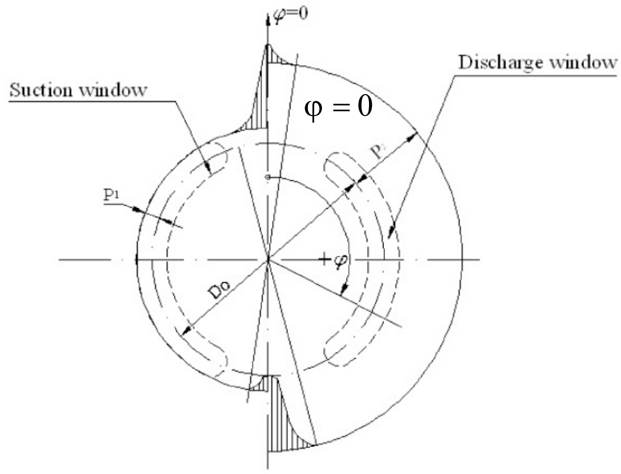



**Fig. 5** *Variation of average pressure on the circle of diameter Do.*

Reduction of the noise level and amplitudes of pressure oscillations can be achieved by "delaying" the beginning of discharge and suction.

By proper choice of the value of $\psi_{a2}$, the pressure from cylinder can be increased by the average pressure from the discharge slot. The speed of the decreasing of pressure in the cylinder at the end of discharge can be reduced by appropriate choice of angle $\psi_{a4}$. Values of the angles $\psi_{a1}$ and $\psi_{a3}$ in this case may be zero or even negative.

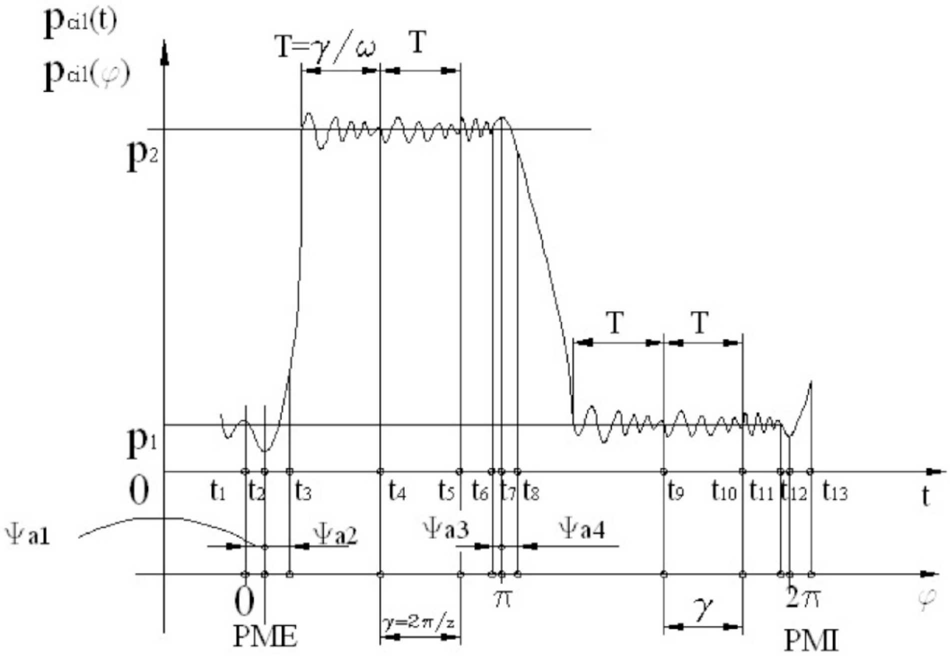



**Fig. 6** – *Indicated diagram of the pump*

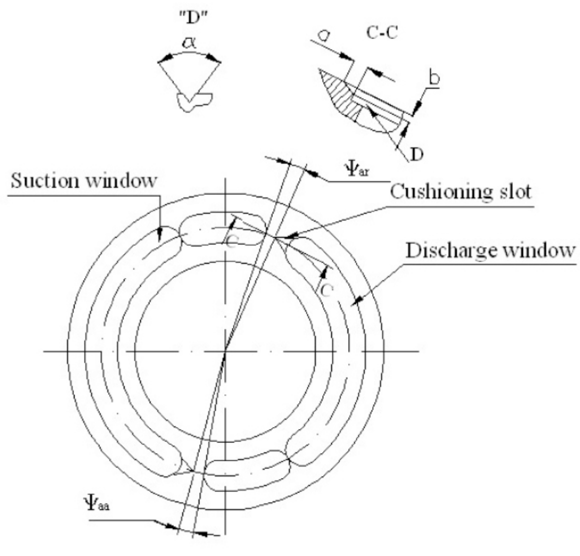



**Fig. 7** *Plane distributor with cushioning slots*

By choosing of some small values of angles $\Psi_{a1}$ and $\Psi_{a2}$, the occurrence of cavitation was avoided and low noise in operation was reduced.

Therefore, the distribution slot having the shape of Fig. 8 distribution exceeds the cylinder overall dimensions.

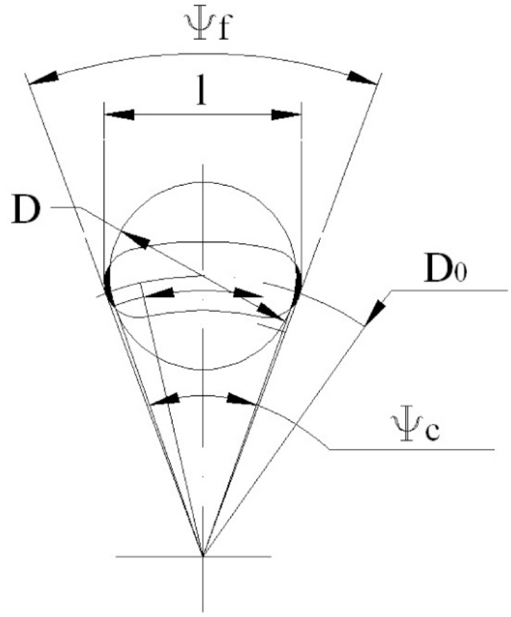



**Fig. 8** - *Geometric elements of the distribution slot*

## 2. Couples of materials

Proposing raising of the technical parameters of our product, respectively pressure and driving speed, it is necessary the increasing of the strength characteristics of surface contact elements, that means of the cylinder block and distributor.

At the choice of friction couples were taken into account the safety criteria of tribological nature to ensure high reliability of the entire mechanical system. Thus, considering the surface interactions occurring in the relative movement of surface contact elements (cylinder +distributor), was established a tribo-system , having as tribo-elements, the steel for the cylinder block and respectively the cast iron cylinder block for the distributor.

The optimizing of the conditions for such a friction torque of materials was obtained by using of a plasma treatment, respectively oxi-nitro-carburize, appeared as a result of the development of the process IONIT OX.

After this treatment for the used couple of materials were obtained the following characteristics:

- high corrosion resistance;
- improved surface hardness;
- good sliding-friction properties;
- improved fatigue-resistance;
- good adhesion behavior due to similar characteristics similar to ceramics.

These results have led to exclusion from the pump component of the bronze distribution plate poz.6, Fig.1, thus fulfilling the last objective, namely the reducing of production costs.

**References:**

Vasiliu N., Catana I., 1988 – *Hydraulic and Electro-hydraulic*, Transmissions Technical Publishing , Bucharest;

Vasiliu N., Vasiliu D., Seteanu I., Radulescu V. 1999 – *Fluids Mechanics and Hydraulic Systems*, Technical Publishing, Bucharest;

Ionescu Fl. s.a., 1980 – *Fluids Mechanics and Hydraulic and Pneumatic Drivings*, Didactic and Pedagogical Publishing, Bucharest;

Alexandru I. s.a. 1997 – *Selection and use of metallic materials* - Didactic and Pedagogical Publishing Bucharest

Olaru D.N. 1993 – *Tribology. Fundamentals of Friction, Wear and Lubrication* - Iaşi Technical University Publishing

Bercea I., Olaru D.N. – *Tribology of Mechanical Systems 1998* – Iaşi Technical University Publishing

7- *Heat Treatments and Engineering of the Surfaces* -Edition III Nr.5-6/2003

ASOCIAŢIA NAŢIONALĂ PROFESIONALĂ DE HIDRAULICĂ ŞI PNEUMATICĂ FLUIDAS

HERVEX

# CONCURS

In cadrul Editiei a -XIX-a
a Simpozionului International
**HERVEX 2011**
care se va desfasura in perioada
9-11 noiembrie va avea loc concursul
pentru studenti si tinerii ingineri
(pana in 30 de ani) pe teme de
hidraulica si pneumatica.

Concursul este la nivel national.
Se acorda premii in bani si obiecte.

**www.hervex.eu**

# OPTIMIZATION OF THE ROBOTS SPATIAL TRAJECTORY BY USING INVERSE KINEMATICS RESULTS, NEURAL NETWORK AND LABVIEW SIMULATION

**Prof.univ.Ph.D.Eng.Adrian Olaru[1, a] and Ph.D.Eng.Serban Olaru[2,b]**

[1]University Politehnica of Bucharest, Romania
[2]Romsys Mechatronics Company, Bucharest, Romania
aolaru_robotics@yahoo.com, bserban1978@yahoo.com

**Abstract.**
*This paper open the way to the assisted research and optimizing the robots spatial trajectory by using the controller with inverse kinematics results. To find the exactly results of the internal coordinates are the complicated problem to obtain the extreme precision of the end-effector movement. To obtain easily these results in the paper was proposed one method by using the proper neural network. In the paper are shown the generalities about the neural networks, some important neural network types and they simulation by proper LabVIEW instrumentation, to choose the optimal network solving this problem and to optimize the obtained errors. Was proposed and used one proper neural network type Bipolar Sigmoid Hyperbolic Tangent Neural Network with time Delay and Recurrent Links (BSHTNN(TDRL)). With these proper virtual LabVIEW instrumentation were established some influences of the network parameters to the obtained the decreasing of the error of the inverse kinematics results and to can use them in to the intelligent control of the spatial trajectory. By on-line used the virtual instrumentation was established that the more important parameters with the bigger influences to the decreasing the trajectory errors were the amplifier factor, the number of neurons in each layer, and the target of the hidden layer, like we can see in the paper research. By using the results of the inverse kinematics problem in the trajectory controller will be obtained the decrease of the errors and the convergence time to the trajectory target.*

**Keywords:** inverse kinematics; neural network; optimizing process; LabVIEW instrumentation.

## Introduction

In many applications where will be necessary to obtain the extreme precision of the driving or of the guidance in a very short time, and when the robot have more than 5 degree of freedom, will be necessary to apply the neural networks solving the inverse kinematics problem with extreme precision. A first wave of interest in neural networks was the introduction of simplified neurons by McCulloch and Pitts in 1943. These neurons were presented as models of biological neurons and as conceptual components for circuits that could perform computational tasks. Many other application of the neural networks has try to developing this field, most notably were Teuvo Kohonen, Stephen Grossberg, James Anderson and Kunihiko Fukushima [1…10].

Neural network are composed of simple elements operating in parallel, like a biological nervous systems. As in nature, the connections between elements largely determine the network function. You can train a neural network to perform a particular function by adjusting the value of the connections (weights) between elements and biases. Typically, neural network are adjusted, or trained, so that a particular input leads to specific target output. There, the network is adjusted, based on comparison of the output and the target, until the network output matches the target. Typically, many such input/target pairs are needed to train a network.

**Mathematical model to solve inverse cinematic problem**

The experimental research and the theoretical cinematic analyze was realized using one arm type robot, fig.1 and some proper virtual LabVIEW instruments. The structural cinematic schema is shown in fig.2. Using the recurrent matrix method were obtained all joints positions of the robot structure.

The recurrent general mathematical model contains the following matrix form:

$$(r)_i^0 = (r)_{i-1}^0 + [D]_{i-1}^0 (r)_i^{i-1} \qquad (1)$$

where the $r_i^0$ is the matrix form of the absolute position vector of the *i* joint; $r_{i-1}^0$- the matrix form of the absolute position of the *i*-1 joint; $r_i^{i-1}$- the matrix form of the relative position vector between the *i* and *i*-1 joints; $D_{i-1}^0$ coordinates transform matrix from the *i*-1 joint to the base Cartesian system. After applying the recurrent mathematical model to the presented robot structure, were been obtained the following position vectors of all robot's joints:

$$(r)_1^0 = \begin{pmatrix} 0 \\ 0 \\ l_1 \end{pmatrix}; (r)_2^0 = \begin{pmatrix} 0 \\ 0 \\ l_1 + l_2 \end{pmatrix}; (r)_3^0 = \begin{pmatrix} c_1 s_2 l_3 \\ s_1 s_2 l_3 \\ l_1 + l_2 + c_2 l_3 \end{pmatrix};$$

$$(r)_4^0 = \begin{pmatrix} c_1 s_2 l_3 + (c_1 c_2 s_3 + c_1 s_2 c_3) l_4 \\ s_1 s_2 l_3 + (s_1 c_2 s_3 + s_1 s_2 c_3) l_4 \\ l_1 + l_2 + c_2 l_3 + (-s_2 s_3 + c_2 c_3) l_4 \end{pmatrix} \qquad (2)$$

where the $l_i$ is the lengths of each robot modules; $c_i$, $s_i$ are the cosines and sinus trigonometric functions of the relative angle and $q_i$ is the relative robot coordinates between *i* and *i*-1 robot bodies. The direct kinematics LabVIEW results are shown in fig.3 for some values of internal coordinates.

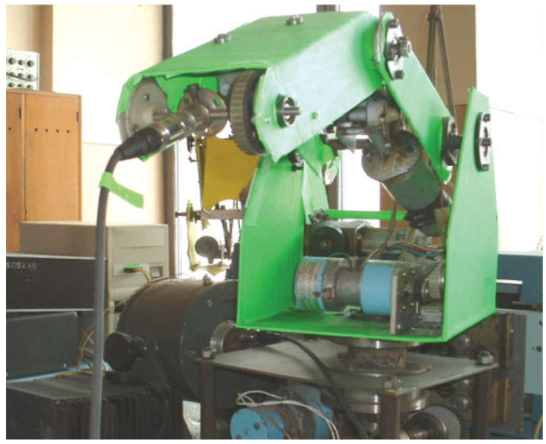


**Figure 1.** *The didactical arm type robot used in the theoretical and experimental research*

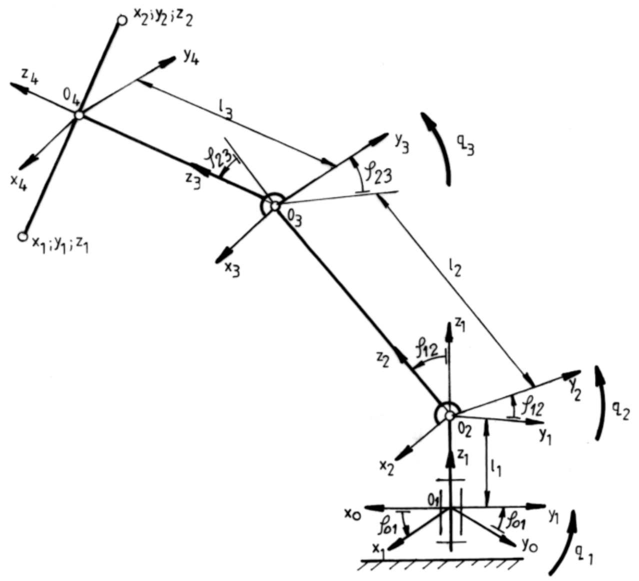


**Figure 2.** *The cinematic- structural schema for the researched arm type robot*

**Research of some important neural network and establish the optimal one for solving the inverse kinematics problem**

To solve the complex cinematic inverse problem were analyzed a lot of some more known neural networks [1-5]. The mathematical models of these networks shown in figs.4...8, are [11...13]:

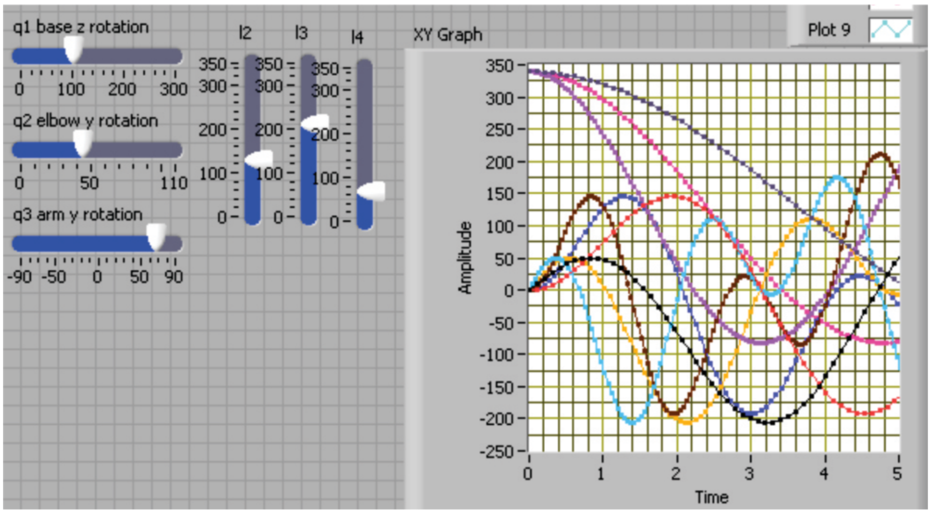



**Figure 3.** *The variation of the absolute coordinates versus the internal coodinates $q_1$, $q_2$ or $q_3$ variation*

$$
\begin{aligned}
a^1(t) &= f^1[IW^{1,1}p^1(t)+b^1+LW^{1,1}a^1(t-1)+LW^{1,3}a^3(t-1)]\\
a^2(t) &= f^2[LW^{2,1}a^1(t-1)+b^2+LW^{2,3}a^3(t-1)]\\
a^3(t) &= f^3[LW^{3,2}a^2+b^3]
\end{aligned} \tag{3}
$$

$$
\begin{aligned}
a^1(t) &= f^1[IW^{1,1}p(t-d+1)+b^1]\\
a^2(t) &= f^2[LW^{2,1}a^1(t)+b^2]
\end{aligned} \tag{4}
$$

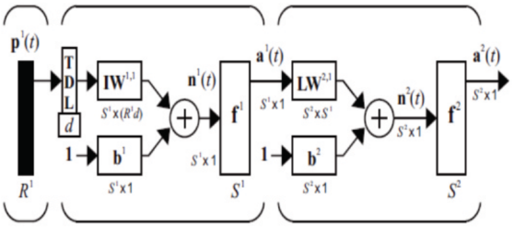


**Figure 5.** *Focused Time-Delay Neural Network (FTDNN)*

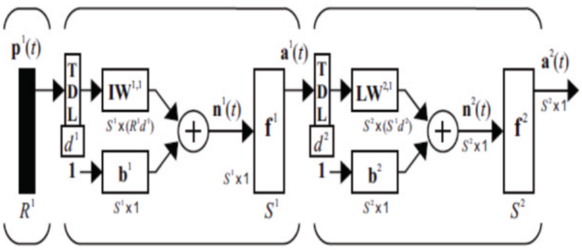


**Figure 6.** *Distributed Time-Delay Neural Network (DTDNN)*

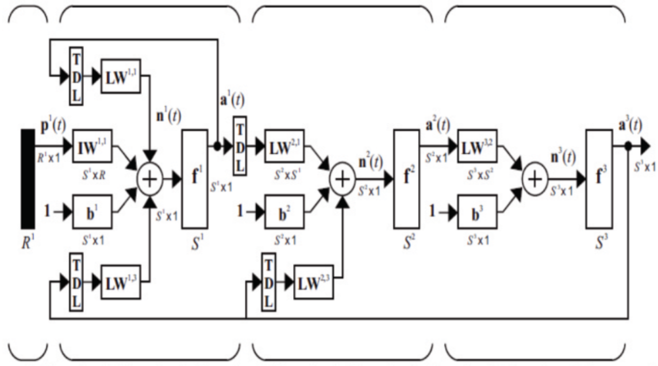


**Figure 4.** *Layered Digital Dynamic Network (LDDN)*

$$
\begin{aligned}
a^1(t) &= f^1[IW^{1,1}p^1(t-d^1+1)+b^1]\\
a^2(t) &= f^2[LW^{2,1}a^1(t-d^2+1)+b^2]
\end{aligned} \tag{5}
$$

$$
\begin{aligned}
a^1(t) &= \frac{1}{1+e^{-[IW^{1,1}u(t-1)+b^1+LW^{1,3}a^2(t-1)]}}\\
a^2(t) &= \frac{1}{1+e^{-[LW^{2,1}a^1(t)+b^2]}}
\end{aligned} \tag{6}
$$

$$
\begin{aligned}
a^1(t) &= \frac{1}{1+e^{-[IW^{1,1}p^1(t)+b^1+LW^{1,2}a^1(t-1)]}}\\
a^2(t) &= LW^{2,1}a^1(t)+b^2
\end{aligned} \tag{7}
$$

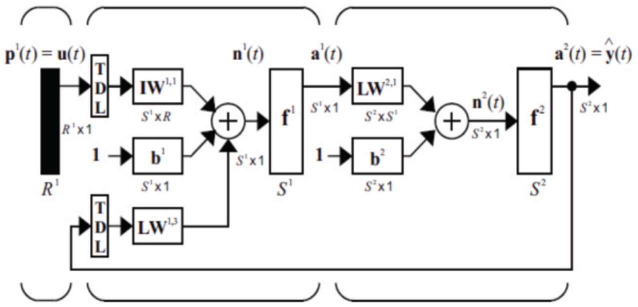


**Figure 7**. *Nonlinear antiregresive with exogene inputs NARX*

After analyzing the results of the inverse kinematics problem, by compare the target (space absolute position of the robot end-effector) with the absolute position obtained by using the relative internal coordinates obtained with neural network for inverse kinematics, we can remarks the following: the calculus is possible to do only for one sensitive function like sigmoid bipolar hyperbolic tangent

**The research of the proper neural network to obtain the optimal results of the inverse kinematics**

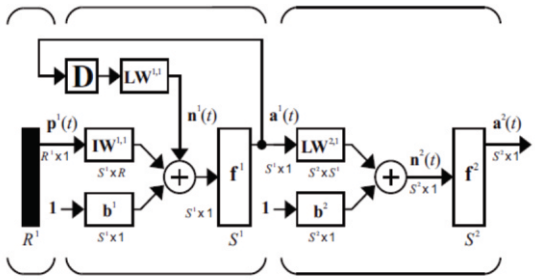


**Figure 8.** *Layer-Recurrent Network (LRN)*

because this function cover the movement in the two direction and determine the convergence process; all the used neural network have the errors between 40-80% and not be possible to use in the analyzed form. For that was proposed one new neural network with some time delay to obtain the different output oscillations in each layer and with recurrent links between the final errors and some points of the proposed network configuration.

To easily obtain the convergence to the target after solving the inverse kinematics problem was applied one proper neural network Bipolar Sigmoid Hyperbolic Tangent type with Time Delay and Recurrent Links and proper teaching law method [11-15].

$$\begin{aligned}
&[w^{3,2}]_{new} = [w^{3,2}{}_{old}] + ((t^3) - (a^3))v^3 && b^2_{new} = b^2_{old} + ((t^2) - (a^2)) \\
&[w^{2,1}]_{new} = [w^{2,1}{}_{old}] + ((t^2) - (a^2))v^2 && b^1_{new} = b^1_{old} + ((t^1) - (a^1)) \\
&[w^{1,1}]_{new} = [w^{1,1}{}_{old}] + ((t^1) - (a^1))v^1 && (t^2) = [f^{-3}(t^3) - b^3]\left[w^{3,2}\right]^{-1} \\
&b^3_{new} = b^3_{old} + ((t^3) - (a^3)) && (t^1) = [f^{-2}(t^2) - b^2]\left[w^{2,1}\right]^{-1}
\end{aligned} \tag{8}$$

The neural network had one 3-8-3-3 neuronal structure to assure the link with the real target and the input data. The base of the teaching law is to determine the error between the target and the output in each layer, by the transfer the target to each layer and adjust the errors by teaching gain $v^i$ and amplifier gain and adjust not only the weights and biases matrix but the target of the hidden layer. The mathematical model was created by using the schema of the proposed neural network for solving the inverse kinematics problem, fig.10, and the proposed teaching law, see relations (8)[14,15].

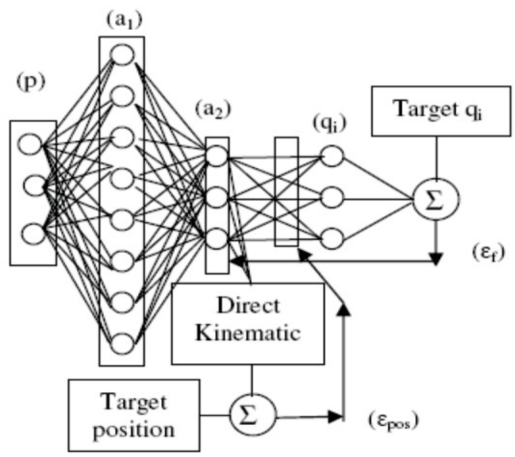



**Figure 9.** *The proposed Neural Network. Simpflified schema*

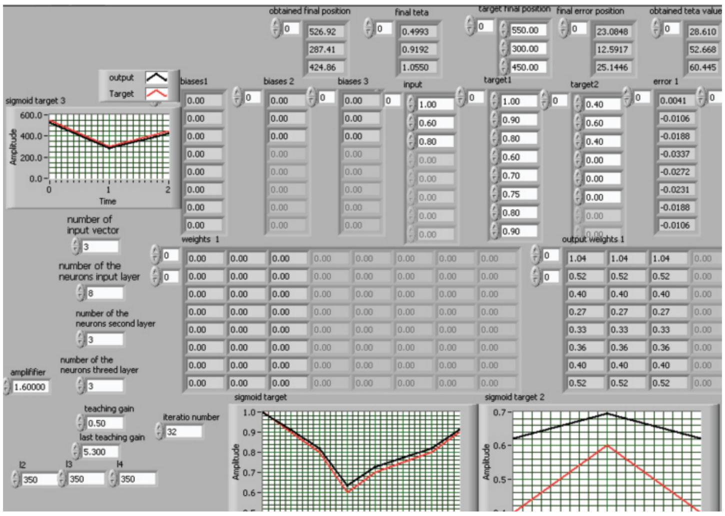


**Figure 11.** *Some results of the space position after applying the changes for the hidden target data, amplifier and teaching gain*

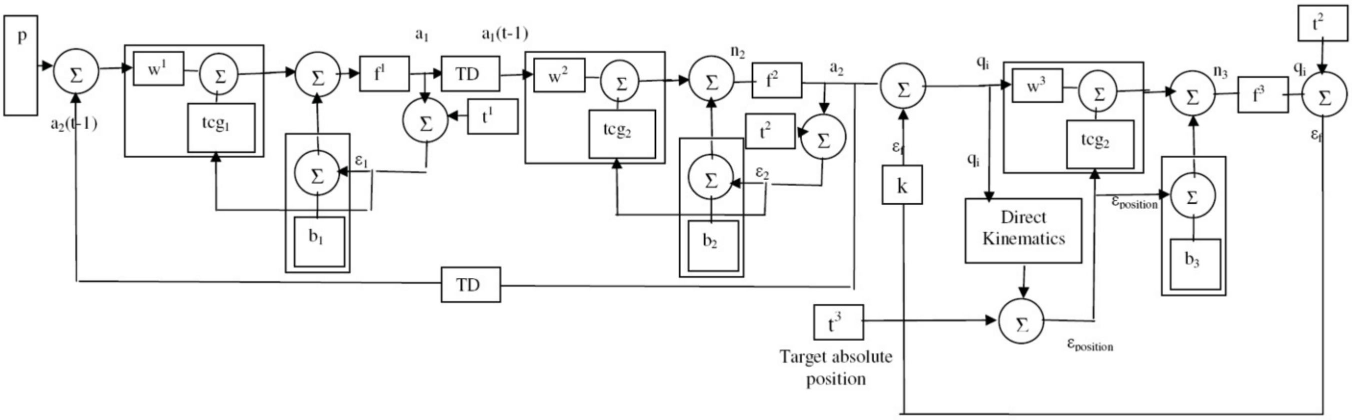



**Figure 10.** *The complex schema of the proper enural network used to solve with minimum errors the inverse kinematics problem to prepare the applying the intelligent controler of the spatial trajectory of the robot's arm*

where $t^i$ is the target and $a^i$ is the output of each layer, $f^i$ is the inverse sensitive function of each layer, $w^{-i,j}$ is the inverse weight matrix between the network layers, $b^i$ is the biases matrices, old and new indices are before and after applying correction.

$$n_1 = [\underbrace{w^1}_{p_1} + \underbrace{tcg_1}_{p_2} \cdot \varepsilon_1](p - a_2(t - p_3 + 1)) + (b_1 + \varepsilon_1)$$

$$a_1 = \frac{p_4(1 - e^{-n_1})}{1 + e^{-n_1}}$$

$$\varepsilon_1 = t_1 - a_1$$

$$n_2 = [w^2 + \underbrace{tcg_2}_{p5} \cdot \varepsilon_2](a_1(t - p_6 + 1)) + (b_2 + \varepsilon_2)$$

$$a_2 = \frac{p_7(1 - e^{-n_2})}{1 + e^{-n_2}}$$

$$\varepsilon_2 = t_2 - a_2$$

$$q_i = p_8(a_2 - \varepsilon_f)$$

$$r_i = \begin{pmatrix} c_1 s_2 l_3 + (c_1 c_2 s_3 + c_1 s_2 c_3) l_4 \\ s_1 s_2 l_3 + (s_1 c_2 s_3 + s_1 s_2 c_3) l_4 \\ l_1 + l_2 + c_2 l_3 + (-s_2 s_3 + c_2 c_3) l_4 \end{pmatrix}$$

$$\varepsilon_{pos} = t_3 - r_i$$

$$n_3 = [w^3 + \underbrace{tcg_2}_{p_5} \cdot \varepsilon_{pos}](q_i) + (b_3 + \varepsilon_{pos})$$

$$a_3 = \frac{p_9(1 - e^{-n_3})}{1 + e^{-n_3}} \qquad (9)$$

$$\varepsilon_f = t_2 - a_3$$

After was applied the proposed neural network were obtained the errors between the target and the proposed end effector space position more that 20%. To obtain the optimization of these errors was researched all influences of some more important parameters of the model to the errors. The parameters what were researched are: $p_1$- the number of neurons; $p_2$ – the first teaching gain; $p_3$- step of the first time delay; $p_4$- the first sensitive function gain; $p_5$- the second teaching gain; $p_6$- the step of the second time delay; $p_7$- the second sensitive function gain; $p_8$- the magnify gain of the proportional error control; $p_9$- the third sensitive function gain and some more recurrent links, see the relations (8).

**Optimization by applying the proper neural network and Numerical simulation with LabVIEW**

The numerical simulation of the network was doing with the LabVIEW soft, version 8.2 from National Instruments. The design of the virtual instruments was done by transpose the mathematical model of the direct kinematics and of the network with proposed teaching law and the proper neural network model (relations (8) and (9)), like we can see in fig.10, in the soft applications. The simulation consisted in the determination of the error after each iteration and the trace of the matrix target t in the comparative characteristics with the output and compare them.

All the virtual instruments work on-line, to see easily what are the changes of the errors or of the output trace, comparing with the target, when was changed the sigmoid bipolar gain, teaching gain v, the inputs p, the weights w, and the bias b. Some results of the assisted research are shown in fig.11. The synthesis of the results is shown in the table 1.

| Config. | Amplif. gain | Teaching gain | $l_i$ | Trget data | Obt. Pos. | Obtained $q_i$ | Target hidden | Iteration number | Relative errors |
|---|---|---|---|---|---|---|---|---|---|
| 3-8-3-3 | 1.8 | 0.2 | 350 | 550<br>300<br>450 | 447.65<br>244.17<br>430.09 | 28.610<br>50.420<br>61.306 | 1<br>0.7<br>0.8 | 132 | 18% |
| 3-7-3-3 | 1.8 | 0.2 | 350 | -‘- | 447.65<br>244.17<br>430.09 | 28.610<br>50.420<br>61.306 | 1<br>0.7<br>0.8 | 132 | 18% |
| 3-8-3-3 | 1.6 | 0.5 | 350 | -‘- | 525.55<br>286.66<br>471.85 | 28.610<br>49.274<br>58.441 | 1<br>0.7<br>0.7 | 132 | 4% |
| 3-8-3-3 | 0.24 | 0.5 | 350 | -‘- | 578.42<br>315.50<br>453.53 | 28.610<br>63.394<br>35.351 | 0.4<br>0.4<br>0.5 | 32 | 4% |
| 3-8-3-3 | 1.6 | 0.5 | 350 | -‘- | 502.04<br>273.84<br>425.81 | 28.610<br>47.947<br>69.003 | 1<br>0.7<br>1 | 8 | 9% |
| 3-8-3-3 | 1.6 | 0.5 | 350 | -‘- | 526.92<br>287.41<br>424.86 | 28.610<br>52.668<br>60.445 | 0.4<br>0.6<br>0.4 | 32 | 4% |

**Table 1.** The synthetic results of the assisted research of the neural network for solving inverse kinematics

**Discussion of the assisted research results**

The numerical simulation shown that one of the most important influences in the optimization of the output way to approach to the target are the teaching algorithm, gain, the number of iterations and the hidden target. We observe that the small gain determine one sensitive and stable approach to the target and the big one determines one oscillation with different magnitude of the output, balanced to the target.

After the assisted research of the sigmoid bipolar neural network parameters, relation (8), we could remark the followings: the change of the number of the neurons in the first layer don't change the errors; the change of the amplifier gain and the teaching gain assured the decreases of the error from 18% to 4% for the 132 number of iteration; one substantial decreasing of the errors and the decreasing of the number of iteration was obtained by on-line changing of the hidden layer target data, 18% to 4% for 32 iteration.; one big difference between the input profile and target data determines divergence; increasing the teaching gain over than one limit determines the instability.

## Conclusions

In the paper was shown some of the more important neuron network types, his mathematical models and how can using these networks to solve and apply the inverse kinematics results in the smart control of the space movement of the robot arm. All created virtual instruments works on-line and it is possible to see the influences of the input elements, weights, biases, number of the neurons in hidden layer or in the input data layer, or some delay time and recurrent links. It is possible to see on-line what is happened when were changed the hidden target profile of the curve, the components of some layers, the different sensitive functions or the teaching gain. With this instrumentation we can choose the optimal form of the neural network concerning the type of the neurons in each layer, the neuron number, input matrix and the teaching gain. By using the on-line work of the virtual LabVIEW instrumentation was possible to obtain on-line, by recurrent links, the optimal values of the hidden target data for one smaller error and for one fast approach. The results and the created virtual LabVIEW instrumentation can be used in many other mechatronic guided applications, to perform the error between the target and the output and to obtain the short time of the convergence. With these new proper virtual instrumentation was open the way to the complex neural network simulation to obtain optimal results.

## References


[1] Blelloch, G., & Rosenberg, C.R. “Network learning on the Connection Machine” Proc. of the Tenth International Joint Conference on Artificial Intelligence. Dunno, pp. 323-326, 1987.

[2] Fukushima, K. Cognitron: A self- organizing multilayered neural network, Biological Cybernetics, 20, pp.121-136,1975.

[3] Hopfield, J.J. Neural networks and physical systems with emergent collective computational abilities, Proc.of the National Academy of Sciences, 81, pp.3088-3092, 1984.

[4] Rosenblatt, F. Principle of Neurodynamics, New-York: Spartan Book.

[5] Dimith, H., Beale M.& Hagan M.: Neural Network ToolboxTM 6- User Guide, The MathWorks, Inc. 3 Apple Hill Drive Natick, MA 01760-2098, USA.

[6] P.J.Alsina, N.S.Gehlot, Direct and inverse kinematics of robot manipulator based on modular neural networks, ICARCV, IEEE, pp. 1743-7, Vol.3, 1994.

[7] R. Manseur, D.Keith, A fast algorithm for reverse kinematics analysis of robot manipulator, International Journal of Robotics Research, Vol.7:3, 1998, pp. 622-648.

[8] D. Gorinevsky, T.Connoly, Comparations of some neural network and scattered data approximations: The inverse manipulator kinematic example, Neural computation, Vol.6, Num.3, pp.521-542.

[9] L.Lee, Training feedforward neural networks: An algorithm givin improved generalization, Neural Networks, Vol.10, No.1, pp.61-68, 1997.

[10] C. Schittenkopf, G.Deco and W. Brauner, Two strategies to avoid overfitting in feedforward networks, Neural Networks, Vol.10, No.3, pp.505-516, 1997.

[11] Olaru, A., Olaru S., Ciupitu, L. Assisted research of the neural network by bach propagation algorithm, OPTIROB 2010 International Conference, Calimanesti, Romania, The RPS Singapore Book, pp.194-200 , 2010.

[12] Olaru, A., Olaru, S., Paune D., Ghionea A. Assisted research of the neural network, OPTIROB 2010 International Conference, Calimanesti, Romania, The RPS Singapore Book, pp.189-194 , 2010.

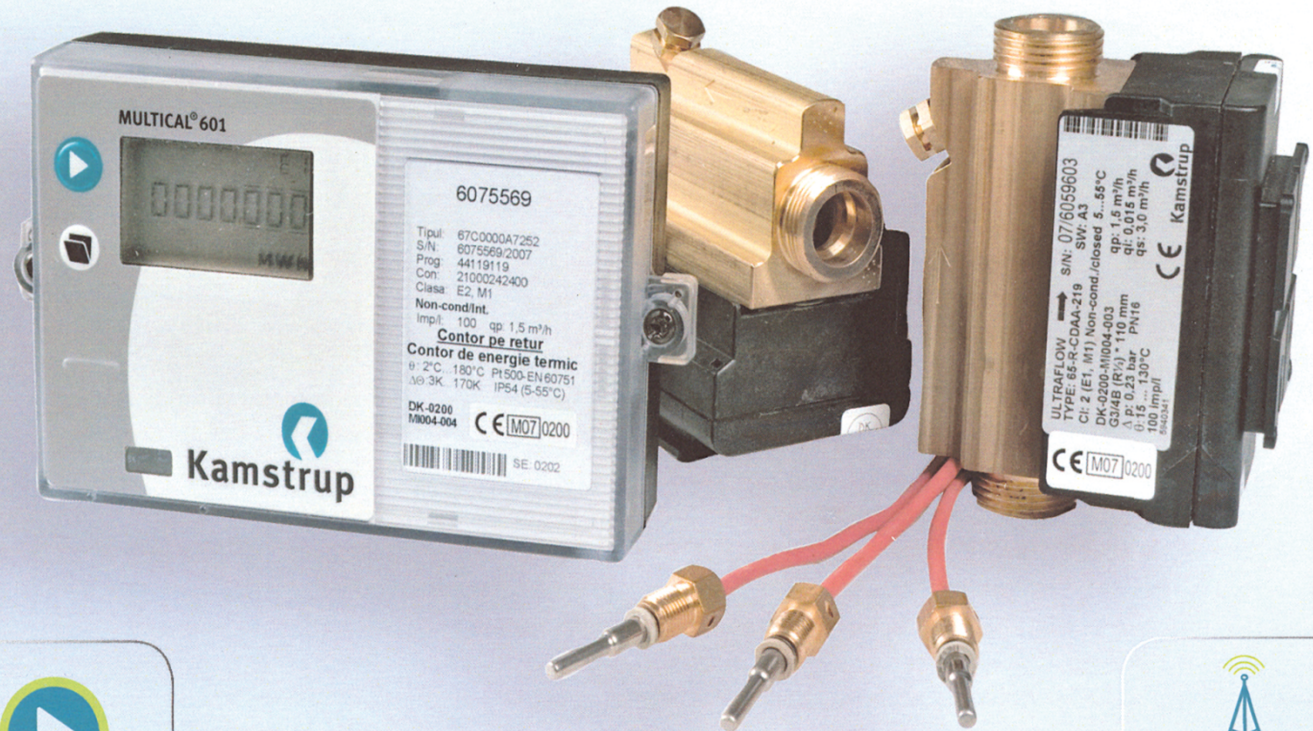
[13] Olaru, A., Olaru, S. Assisted research of the neural network with LabVIEW instrumentation, IEEE ICMENS-2010 Proceedings, Changsha, China, pp.1-8, 2010.

[14] Olaru, A., Oprean, A., Olaru, S., Paune, D. Optimization of the neural network by using the LabVIEW instrumentation, IEEE ICMERA 2010 Proceedings, ISBN 978-1-4244-8867-4, IEEE catalog number CFP1057L-ART, pp.40-44, 2010.

[15] Olaru, A., Olaru, S., Paune, D. Assisted research dynamic neural network with time delay and recurrent links, IEEE ICMERA 2010 Proceedings, ISBN 978-1-4244-8867-4, IEEE catalog number CFP1057L-ART, pp.284-288, 2010.

GENERAL FLUID S.A.

14, Cutitul de Argint Street, district 4, Bucharest - Romania
Phone: 004 021 337 00 78 ; Fax: 004 031 405 40 99
office@generalfluid.ro ; www.generalfluid.ro




# MULTICAL® 601

CONTORIZAREA CONSUMURILOR DE CALDURA PENTRU INCALZIRE SI APA CALDA, PE BAZA PRINCIPIULUI ULTRASONIC DE MASURA

**Mod de operare**
Display intuitiv si simplu de utilizat

**Citire la distanta**
M-Bus, Radio, LONWorks, etc.

**Memorie**
Standard, cu jurnale de date protejate

**Ultrasonic**
Sistem de masura precis, cu durata lunga de viata

**Standarde**
Conform MID si EN 1434:2004

**Flexibil**
Noi functiuni si interfete pot fi oricand incorporate

Kamstrup
– metering solutions

# A SIMPLIFIED MODEL FOR PARTIAL JOURNAL AND WATER LUBRICATION

**Lorena DELEANU***
*Universitatea "Dunărea de Jos" Galati, Romania

***ABSTRACT***
*This article presents a theoretical model for steel shaft sliding against an elastic partial journal, in open circuit water. The model is based on the LHD theory but takes into account the particularities of the bearing material and fluid, as found from experimental data obtained by the author (friction coefficient,almost constant or insignificant temperature variation of water and bearing) and from literature (water viscosity and bearing material properties). Based on this model with the system parameters as load, sliding speed and temperature being kept constant, the results point out that water is capable to form very thin continuous films (~1.5...8 m).*

***Keywords***: EHD model for elastic journal, water lubrication, continuous film.

## 1. A Simplified EHD model for smooth surfaces

Modern engineering uses conformal contact for sliding bearing [2, 3, 11]. The designer would like that tribosystem parameters allow to form and maintain a continuous fluid film of lubricant or, for a mixt regime, that system reliability is less influenced. Studies on bearing failures ask a more accurate evaluation of the contact parameters. Shi and Wang [15, 19] pointed out the importance of theoretical modeling based on a mixt regime. For rough surfaces, there were elaborated models for hydrodynamic regime [5] EHD and TEHD lubrication [8]. Equations for an EHD model are

- average Reynolds equation:

$$\frac{\partial}{R_B\partial\theta}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{R_B\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{R_B\partial z}\right)=6U\frac{\partial\rho h}{R_B\partial\theta}+6U\frac{\partial\rho}{R_B\partial\theta} \quad (1)$$

- equation for force equilibrium

$$F=\sum_j\left(\sum_k\left[p(\theta_j,z_k)\right]\cos(\theta_j)R_B\Delta\theta_j\Delta z_k\right) \quad (2)$$

-equation for lubricant film thickness, with journal and bearing elastic deformations:

$$h(\theta,z)=h_o(\theta)+\delta_j(\theta,p)+\delta_B(\theta,p) \quad (3)$$

Thermal expansion was considered having a linear dependence and in [8] it is calculated for the fluid temperature, included in this model as a constant.

The heat source in energy equation (3) includes both viscous dissipation and friction of directly contacted asperities and characterized TEHD models.

## 2. Simulation of water lubrication

For the theoretical study of the tribotester partial bearing, the model is drawn in Figure 1, boundary conditions and hypothesis are in agreement with water lubrication as they were confirmed by experimental results obtained on PTFE composites sliding on steel journal [3]:

- values of roughness (Fig. 2) allow to consider both surfaces as smooth; average maximum values do not exceed 1 m, after sliding in water, for many plastics or their composites; this hypothesis simplifies the model [8];
- measuring temperatures with transducers mounted as shown in Figure 3, allow to consider the water lubricated tribosystem as being isothermal. The partial bearing has, with enough accuracy, the same temperature as water in open circuit. Figure 2 presents the temperature values within the partial bearing for a composite PTFE + 60% bronze; for low speed temperature rises especially for gauge T3, meaning that the regime is very probably mixt (semi-fluid); at higher speed, the differences among the measured temperatures in this three points and water bath are very small, the regime could be considered of EHD nature.

Temperature variations at the shoe edge demonstrate that contact may intermitently occur in these regions;

- the shaft is considered perfectly rigid in order to simplify the model as there is a great differences between mechanical properties of the two involved bodies (see Table 1) ;
- water is an incompressible fluid; for the loading range [14]. As temperature has small variations (under 1C), its viscosity is also a constant [1, 4, 5];

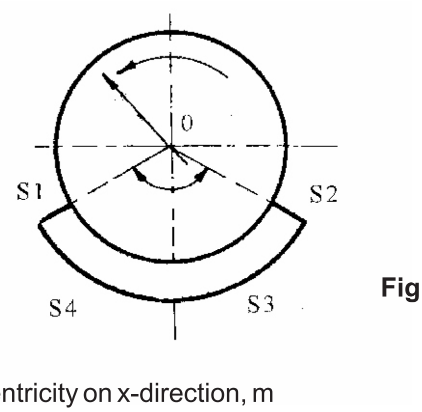


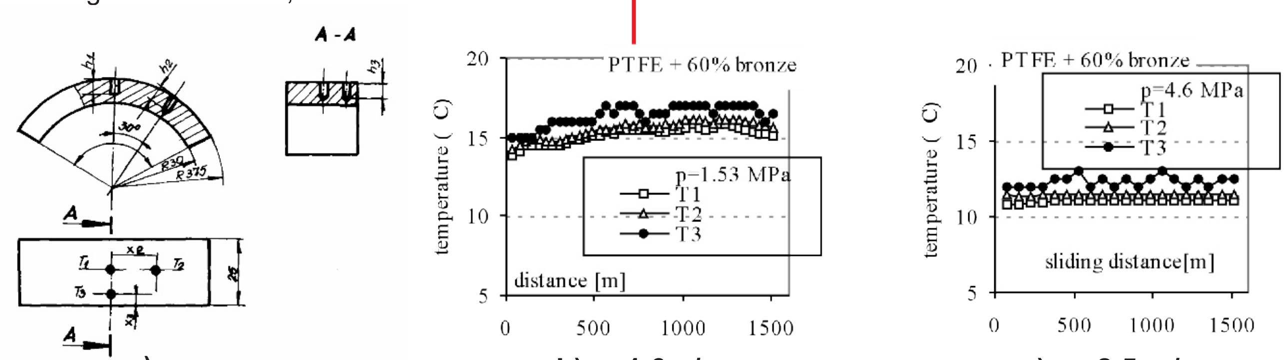

**Fig. 1**

**Notations**

B bearing width, m
C radial clearance, m $C = R_B - R_j$
$C_p$ specific heat, J/(kg.°C)
h film thickness
k thermal conductivity W/(m.°C)
R radius, m
T temperature, °C
U peripheral journal speed, m/s
r, z radial and width co-ordinates, respectively,
F load, N
α coefficient of thermal expansion, mm/m.K
θ angular co-ordinate, radians
$\varepsilon_x$ eccentricity on x-direction, m
δ deformation, μm
Φ heat source
φ angle of the bearing, radians
η viscosity, Pa.s
ρ density, kg/m$^3$
μ friction coefficient
σ or $R_z$ value characterizing the asperities (RMS)
Λ adimensional thickness of the film $\Lambda = h_{\min} / \sigma$

**Index**

o origin, initial; B for bearing, j for journal
r, θ, z co-ordinate on radial, circumferential and width direction, respectively.




a) **b) v=1.6m/s** **c) v= 2.5 m/s**

**Fig. 2.** Temperature variation within the superficial layer for sliding speed. a) Position of temperature transducers: $x_1 = 0.87; x_2 = 1.23; x_3 = 1.64\ [mm]$; $\theta_{water} = 11.0 \pm 0.5\ ^oC$

- friction coefficient is considered constant and relatively small, characterizing steel journal sliding in water on PTFE composites [17], after calculating the normal pressure distribution, on the surface of each finite element situated in contact with water it is introduced a friction effort $p_f(\theta, z) = \mu \cdot p(\theta, z)$ .

Boundary conditions for the presented model are:

- the shaft surface is considered non-deformable, this hypothesis being valid especially because maximum pressure in the water film are small (~25...100 times smaller as compared to conventional flow limit of a heat-treated steel);
- as this model includes a partial bearing (120), eccentricity is not calculated but imposed or initialized in program; for journal having neglectable weight as compared to applied load F, the partial bearing position related to a certain plane does not influence the model behaviour;
- external cylindrical surface of partial bearing (S4) lies on a rigid and fixed surface;
- lateral surfaces of partial bearing S3 (initially situated in planes perpendicular to the bearing ax), are considered free;
- surfaces S1 and S2 are blocked as it is on the tested tribomodel by fixing plates.

Logical diagram of this program is presented in Figure 3.

### 3. Presenting the program

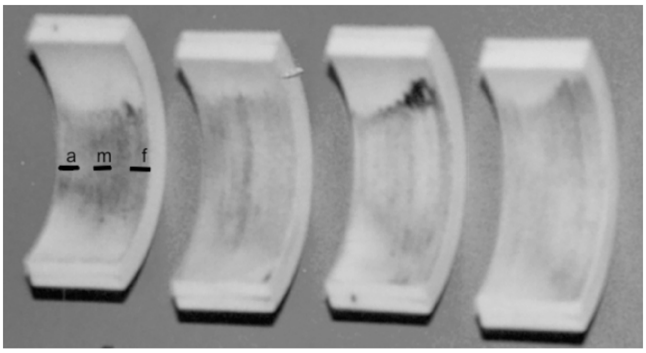
This program for simulating the film formed by low viscosity fluids including water, when a journal is sliding on a deformable partial bearing, was elaborated based on the soft COSMOS I-75 and it may do:

- calculating the minimum thickness and pressure distribution of a continuous water film when a steel journal is sliding on a partial elastic bearing, knowing load F and sliding speed v, it may plot the water film thickness;
- establishing the stress and strain fields for the partial bearing;
- studying the influence of factors as external load F, sliding speed v, water temperature (as a constant in time), bearing material, bearing geometry (angle of partial bearing till 360, width, clearance, eccentricity, thickness of the wall of deformable bearing) for sliding tribomodels having smooth surfaces and small wear, lubricated with low viscosity fluids, including water.

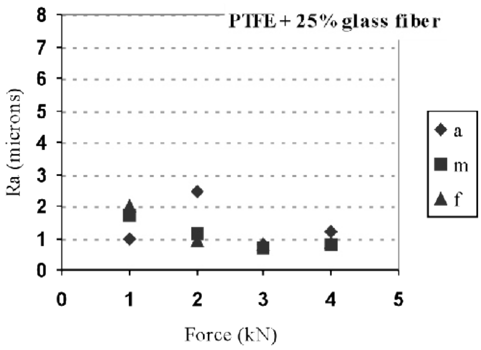
The finite element FLOW3D used for solving the average Reynolds equation and for calculating stress and strain in partial bearing (Figure 4) by finite element method is a volume with eight nodes, also used for 3D fluid flow models. Each node has four degree of freedom including three velocity components (u, v, w) and temperature. The fluid properties may be introduced by functions or constants; dynamic viscosity, density, thermal conductivity, specific heat, coefficient for volume expansion, parameters for power-law characterizing the non-Newtonian fluids. Loading possibilities of this element are: pressure (applied normally or/and tangentially on its faces), thermal conductivity, internal heat generation, applied heat flow.

For each node there are possible to calculate velocity, pressure and temperature. The so-called node pressure is the average of pressures characterizing the element connected in that node.

The fluid film was composed by 950 nodes, set of 190 on each of the five surfaces.




a) 25% glass fibers




b) Ra parameter for the partial journal

**Fig. 4**. Roughness after sliding in water (open circuit ~2 l/min), v=2.5 m/s, after 10500 m [3]. a) codification of the

Partial bearing was fragmented with the same number of nodes. Figure 6 presents symbol-vectors for pressure distribution on the surface element of the partial bearing. Pressure on each element was calculated as an average of forces applied in nodes delimiting the element, characteristic for COSMOS I-75. Friction force on surface element was considered proportional to normal force on the same element and the value of friction coefficient is required in this program as input data. In order to apply the friction forces on each element, a cylindrical system of co-ordinates was used as it is suggested in Fig. 4, left-up.

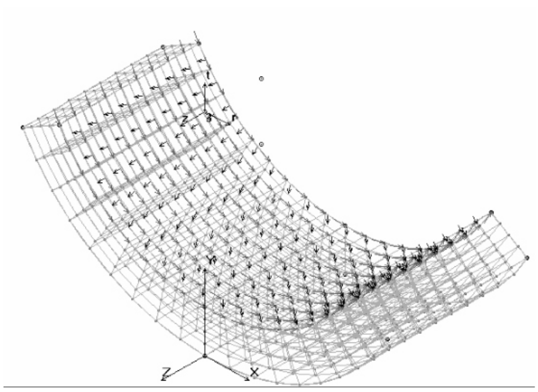


**Fig. 5.**

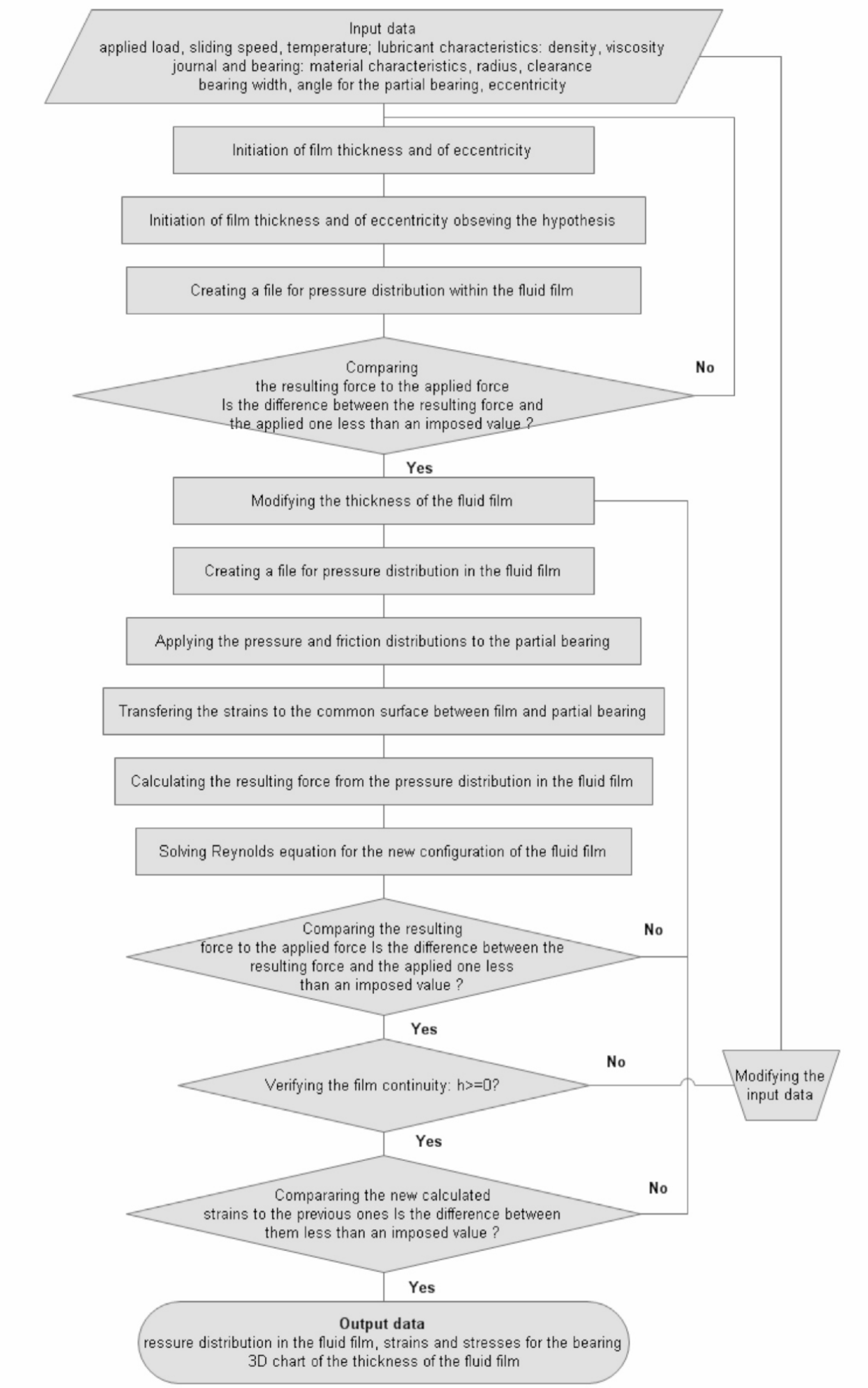
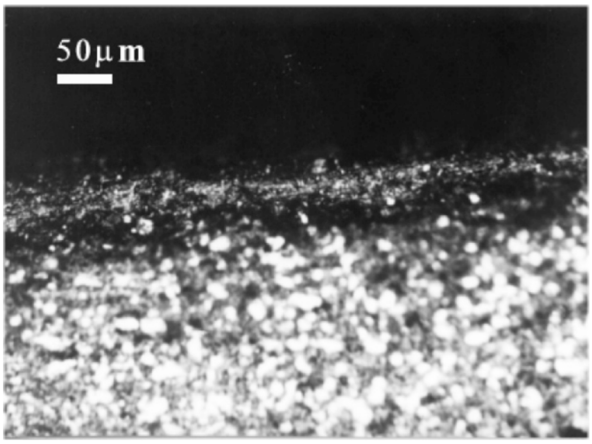


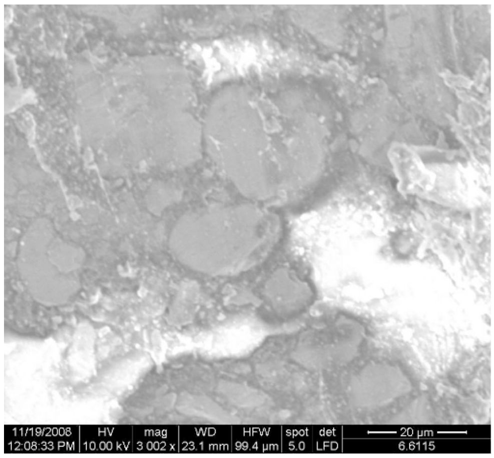

**Fig. 3**. Logical diagram of the program simulating steel journal sliding on a deformable bearing in water.

As it is not known the exactly place for minimum fluid thickness, a subroutine was elaborated for calculating film thickness on radial direction (of the journal) (Figure 7): the film is intersected by fans of straight lines situated in planes perpendicular to journal ax and being equidistant on the initial bearing width. This routine calculates the segment included within the fluid film. Film thickness calculated in orthogonal co-ordinates may give errors due to great strains characterizing the bearing, which are then introduced to the film geometry.
For a quick solving of verifying film continuity, this one was virtually limited by planes parallel to co-ordinate plane y–z, and the partial bearing was also limited by planes perpendicular to journal ax and including extreme nodes of film surface next to the journal.

Programs for calculating stress and strains field do not take into account this virtual limiting necessary for verifying film continuity.
Based on references [10, 12, 16], the author established the initial value for film thickness at 6 or 10 m as it was theoretically demonstrated [5, 10] and experimentally confirmed that water does not form thick film as a consequence of its low viscosity [6, 7. 8, 17, 18, 20].
For these initial values of film thickness, convergence of calculus for external load F and deformations is achieved after 6...10 iterative steps, as the imposed tolerances are of about 1...5‰ for load and 3...5% for deformations.
The program does not take into account processes in the superficial layer that modifies mechanical properties in ~50...200 m depth but the bearing material may be introduced as two or many different layers.




a) shoe-roller tribotester [3]
v=3 m/s and p=4.6 MPa




b) plate (composite) – disc (steel)
tribotester, v=2.5 m/s and p=2.2 MPa

**Fig. 6.** PTFE + 60% bronze, after sliding in water in open circuit

The process of agglomerating the glass fibers in the tribolayer makes the superficial layer rigidity to increase as noticed for PTFE composites but these resulting mechanical characteristics are very difficult to be evaluated. Relationships for an equivalent modulus of elasticity are given in literature but only for bulk material but they may be a starting point for analyzing a composite bearing.

### 4. Results

The characteristics for bearing material are usual ones for plastics composite. As PTFE has a small longitudinal modulus of elasticity (E=400MPa [9]...500 MPa [13]) and traction limit $_r$ ~20 MPa [9], its composites have a higher modulus, traction limit being also greater but less spectacular. Thus, a composite PPS + PTFE + ZnO [13] has E=1650 MPa and $_r$ = 56 MPa. Other ones, PTFE being the filling material may rich E~16000 MPa and $_r$ ~100 MPa [9, 13]. There will be analyzed the results for the data presented in Table 1. Figure 7 presents von Mises stresses in the radial plane containing the maximum water pressure (isothermal regime), bearing and journal are rigid bodies. as a consequence of passing the first loop of the program (Fig. 3).

The minimum thickness of continuous water film was ~6,35 m, for an eccentricity of $\varepsilon_x = 6\mu m$ . (The difference between applied force and that resulting from the film pressure was set at 5‰).
For this conformal contact, the pressure maximum increases with ~20% for an elastic shoe and the depression zone in front of the contact is decreasing as suggested in Figure 8b. When bodies are rigid, due to lateral fluid flows, the higher values of the water pressure are distributed on a smaller area. If the bearing is made of soft material (Fig. 8b), lateral water flows are diminished as a consequence of smaller deformations (and, of course, thinner film thickness) near the bearing edges. Once entered into the contact, the water is forced to pass out on circumferential direction and less on lateral sides, as here the clearance between solid bodies is smaller than in the middle zone. The result is an increase of the area with higher pressures but also an increase of the pressure peak. The strains suggests that an increase of the external load may cancel the water film continuity on the lateral zones of the contact as a confirmation of the model proposed by Wang and Shi [15, 19].

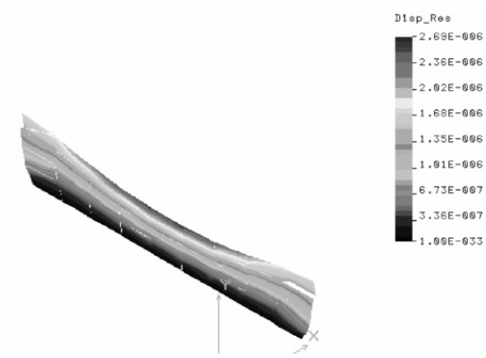
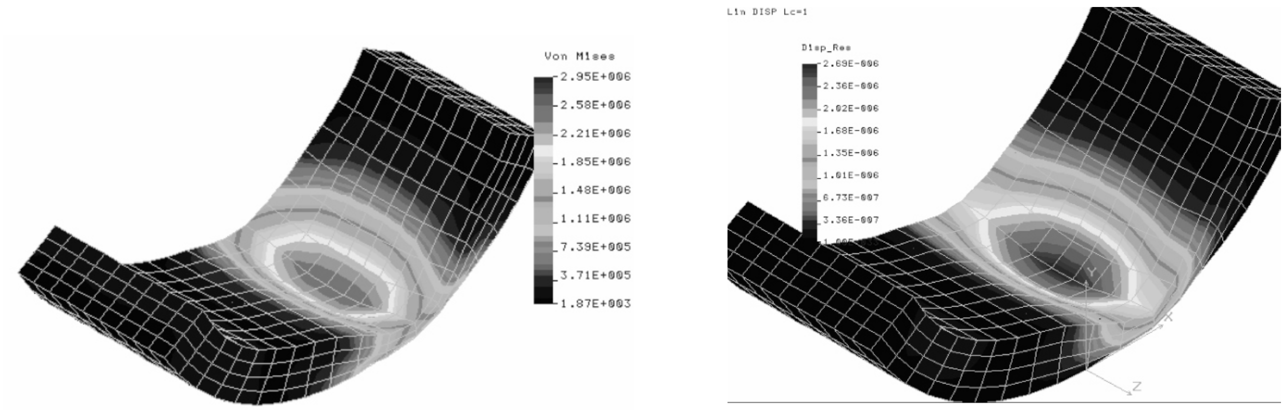



**Fig. 7.** A section in the shoe for the maximum values of stresses

**Table 1**

| | | | |
|---|---|---|---|
| External applied load [kN] | | | 1000 |
| sliding speed [m/s] | | | 2.5 |
| water temperature | | | 20°C |
| Initial eccentricity $\varepsilon_x$ | | | 6 μm |
| Friction coefficient | | μ = 0.02 | |
| **Geometry** | | | |
| partial bearing | radius | | 30.05 mm |
| | width | | 25.00 mm |
| | angle | | 120° |
| journal | radius | | 30.00 mm |
| **Materials** | | **bearing** | **journal** |
| traction limit [MPa] | | 40 | 770 (steel) |
| Poisson ratio | | 0.35 | 0.3 |
| longitudinal modulus of elasticity [MPa] | | 7000 | $2.15.10^5$ |




a) Elastic bearing b) Rigid bodies (steel)

**Fig. 8.**

## 5. Conclusions

The continuous water film allows formulating an isothermal model due to its cooling effect. Temperature recordings also sustain this isothermal solution for water lubrication.
As suggested by the low values of friction coefficient [3, 17], water can generate continuous films for specified ranges of load and speed (kept constant) even if their thicknesses are smaller than those obtained under the same regime using more viscous lubricants. For instance, data in table 1 generate a minimum water film thickness of ~5.33 μm.

Spikes [16] noticed that the continuity of the fluid film depends on an adimensional parameter $\Lambda = h_{\min}/\sigma$ ; as composites with plastic matrix sliding in water may be characterized by $h_{\min} \approx 1...6\mu m$ and $\sigma \leq 0.3...1\mu m$ it is possible to have continuous water film, based on this criterion ( $\Lambda \geq 1.5$ ).

For an elastic bearing, the maximum fluid pressure is higher as compared to the same value obtained for rigid bodies.

The strain analysis allows comparing the obtained values to those admitted by a correct functioning of a tribosystem with conformal contact. For instance, running the program for the data given in Table 1, resultant maximum strains were of 2.69 m, much smaller than dimension tolerances and clearance and, therefore, acceptable.

Stresses within the partial bearing have maximum values in the zone supporting the fluid pressure peak as it may be seen in Figure 8b. Origin of orthogonal system is drawn for the middle of the angle characterizing the partial bearing.

## REFERENCES


1. Arnell R. D., Davies P. B., Halling J., Whomes T. W., Tribology, MacMillan Education Ltd., London, 1991
3. Constantinescu V. N., Nica Al., Pascovici M. D., Ceptureanu Gh., Nedelcu Ş., Lagăre de alunecare, Ed. Tehnică, Bucharest, 1980
3. Deleanu L., On Tribological behavior of PTFE composites in Sliding tribosystems (in Romanian), PhD, University “Dunarea de Jos” of Galati, Romania, 1999.
4. Dowson D., Higginson G. R., Elasto-Hydrodynamic Lubrication, Pergamon Press, Oxford, New York, Toronto, Sidney, Paris, 1977.
5. Frêne J., Nicolas D., Deguerece B., Berthe D., Godet M., Lubrification hydrodynamique. Paliers et butées, Edition Eyrolles, Paris, France, 1990.
6. Hapenciuc M., Contribuții la studiul lagărelor greu încărcate cu cuzineți din materiale plastice, cu ungere cu apă (in Romanian), PhD, University “Dunărea de Jos” of Galati, 1998.
8. Hashimoto, Mongkolwongrojn M., Adiabatic Approximate Solution for Static and Dynamic Characteristics of Turbulent Partial Journal Bearings with Surface Roughness, Transactions of the ASME, vol. 116, pp. 672-680, 1994.
9. Catalogue CEPROINV Focşani, 2010, http://www.ceproinv.ro/pdf/prop_fiz_PVDF_FLUOROSINT_PTFE.pdf
10. Isaksson O., Numerical Analysis of Elastohydrodynamic Contacts Using Power-Law Lubricant With Special Reference to Water-Based Hydraulic Fluids, ASLE Transactions, vol. 30 pp. 501-507, 1987.
11. Johnson K.L., Contact Mechanics, Cambridge University Press, 1985.
12. Moore D.F., Principles and Applications of Tribology, Pergamon International Library, 1975.
14. Neal M. J., Tribology Handbook, London, Newnes Butter Worth, 1973.
15. Shi F., Wang J.Q., A Mixed-TEHD Model for Journal-Bearing Conformal Contacts – Part I: Model Formulation and Approximation of Heat Transfer Considering Asperity Contact, Transactions of the ASME, Journal of Tribology, vol. 120, p 198-205, 1998.
16. Spikes H. A., Advances in the study of thin lubricant films, Proc. of First World Tribology Congress, pp. 355-369, London, 1997.
18. Wan G.T.Y., Kenny P., Spikes H.A., Elastohydrodynamic Properties of Water-Based Fire Resistant Hydraulic Fluids, Tribology International, vol. 17, no. 16, 1984.
19. Wang J.Q., Shi F., A Mixed-TEHD Model for Journal-Bearing Conformal Contacts – Part II: Contact, Film Thickness and performance Analysis, Transactions of the ASME, J. of Tribology, vol. 120, p 206-213, 1998.
20. Watanabe M., Wear mechanism of PTFE composites in aqueous environment, Wear, 158, p 79-86, 1992.

## PNEUMATICON 2011 FAIR PARTICIPATION

National Professional Association - Fluidas - participated at the fair Pneumaticon 2011 from Targi Kielce - POLAND, between 01.03-03.03.2011 and it was represented by PhD.Eng Popescu Costinel from INOE 2000-IHP and eng. Cosmin ZAVOIANU from HERVIL Ltd.



## PRESS CONFERENCE

On the date14/06/2011, at National Professional Association of Hydraulics and Pneumatics **"FLUIDAS"** took place press conference for launch the project **"TRAINING OF SPECIALISTS IN THE FIELDS OF MECHANICS, HYDRAULICS AND PNEUMATICS IN ORDER TO PROMOTE ADAPTABILITY AND INCREASE COMPETITIVENESS" POSDRU/81/3.2/S/47649,** project that runs from April 2011 - April 2013.

The project is co financed from The Social European Fund, by means of the Operational Sectorial Program Human Resources Development 2007-2013, Prioritary axis 3 "Increase adaptability of the workers and enterprises", Main field of intervention 3.2. "Training and support for enterprises and employees in order to promote adaptability".

***Applicant:*** Chamber of Commerce and Industry Valcea, Romania
implement the project together with partners:
***Partner 1:*** National Professional Association of Hydraulics and Pneumatics - FLUIDAS,Romania
***Partner 2:*** Technical University of Cluj-Napoca, Romania
***Partner 3:*** Technical University "Gheorghe Asachi" of Iasi, Romania
***Partner 4:*** PIA e.V. – Development and Assessment Institute in Waste Water Technology at RWTH - Aachen University – Germania

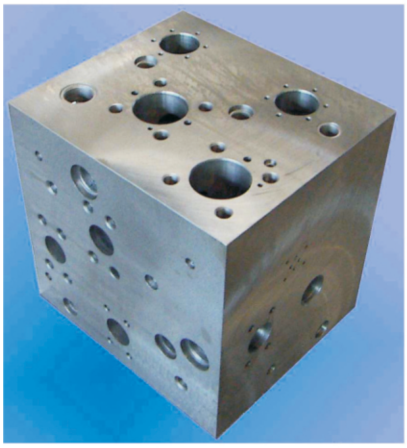
At press conference besides the project team also participated the representatives of the press: T & T Magazine, represented by Mr. Director / Chief Editor Onut ILIESCU, Tehnomarket Magazine represented by editor / DTP Andrei Avram, "Automatizari si Instrumentatie" Magazine - manager PhD. Eng. Horia Mihai Motit and attended by representatives of companies from Fluid Power field and professional associations, such as Festo, SMC, APROMECA, SUSZI Industrial, Romfluid, HYDRAMOLD, HESPER, AAIR, INOE 2000 - IHP.

**HIDROSIB S. A.**

**ROMÂNIA 550316 – SIBIU, STR. ŞTEFAN CEL MARE 193 R 795209**

Cu o vasta experienta in domeniul aparaturii hidraulice S.C. Hidrosib S.A se prezinta si astazi ca un important producator de instalatii, blocuri si centrale hidraulice inclusiv cele de mare complexitate.

Beneficiind de competenta si inaltul nivel calitativ al proiectarii si executiei, produsele noastre sunt recunoscute atat pe piata europeana cat si de importante firme din Romania care se numara printre clientii nostrii.



Atestarile

**ISO 9001** si **ISO 14001** sunt dovada permanentei preocupari pentru cresterea performantelor in vederea satisfacerii tuturor cerintelor clientilor la cele mai inalte cote.

Pentru a veni in intampinarea clientilor nostrii, putem executa lucrari atat dupa proiecte propri realizate la tema, cat si ale beneficiarului.

Pentru rezolvarea problemelor dumneavoastra adresati-va specialistilor nostrii. Avem convingerea ca vom putea identifica impreuna posibilitatile concrete de colaborare intre firmele noastre.

Datele noastre de contact :

Secretariat: 0269/214060 Centrala: 0269/253206 ; Comercial: 0269/245335
Fax: 0269/253216 ; E-mail: tehnic@hidrosib.ro ; Site: www.hidrosib.ro

# I N V I T A Ţ I E

## Al 19-lea Simpozion A.A.I.R.

**28 septembrie 2011, Uzinexport Bucureşti**

(Bd. Iancu de Hunedoara nr. 8, bloc H3, etaj 7, sala ON TOP OF BUCHAREST)

iunie2011

Manifestare naţională de referinţă, Simpozionul A.A.I.R. reuneşte oferta producătorilor/distribuitorilor instrumentaţiei cu cererile utilizatorilor acesteia, în domeniile: automatizărilor, măsurărilor, acţionărilor, prelucrării datelor şi instrumentaţiei virtuale.

Vă aşteptăm în septembrie la Simpozionul A.A.I.R!

Preşedinte
Dr. ing. Horia Mihai MOŢIT

**Lucrările selecţionate până în prezent pentru a fi prezentate sunt următoarele:**

### Gestiunea oprimă a gazelor şi energiei prin automatizări

1. **Sisteme de reglare a presiunii gazelor naturale utilizând acţionări electice cu controler PID integrat.** Ing. Raul ŢETCU - HASEL INDUSTRIAL S.R.L. Tg.Mureş, Ing. Marinel MORARU - Area Sales Representative - PHOENIX CONTACT S.R.L. Bucueşti
2. **Problemeatica determinarii capacităţii unui sistem de transport al gazelor naturale.** Ing. Ioan MOISIN, Dr. ing. Constantin Dorin BICHIŞ – SNTGN TRANSGAZ S.A. Mediaş
3. **Aplicaţii practice ale măsurărilor de calitatea energiei.** Ing. Roxana POPOVICI – RONEXPRIN S.R.L. Bucureşti

### Automatizări & Instrumentaţie

1. **Noutăţi de la ENDRESS + HAUSER în domeniul instrumentaţiei.** Ing. Cristian ANDREI, Director de Vânzări - ENDRESS + HAUSER ROMANIA S.R.L.
2. **Automatizarea unei instalaţii de distilare atmosferică.** Dr. ing. Sorin BROTAC, Ing. Marius SANDU - S_IND PROCESS CONTROL S.R.L.
3. **Sistemul de automatizări de siguranţă PSS 4000.** Ing. Rainald HORNUNG Director, Ing. Andrei HÂNCU Product Manager, PILZ - VDR & SERVICII S.R.L. Bucureşti
4. **Funcţional Safety Management, componentă a Sistemului de Management al Calităţii.** Ing. Gabriel DINESCU FSM Responsible - YOKOGAWA EUROPE B.V. Olanda – Sucursala ROMANIA
5. **Utilizarea standardului IEC 61.499 în programarea sistemelor de automatizare industrială.** Conf. dr. ing. Eugen DIACONESCU – UNIVERSITATEA DIN PITEŞTI
6. **Schemă funcţională pentru reglarea automată a nivelului lichidului din rezervoare.** Prof. dr. ing. Dănuţ ZAHARIEA - Fac. Construcţii de Maşini şi Management Industrial (Catedra Mecanica Fluidelor, Maşini şi Acţionări Hidraulice şi Pneumatice) - U.T. "GH. ASACHI" IAŞI

**Confirmarea participării la Simpozion implică transmiterea la Secretariatul A.A.I.R. a următoarelor:**

**a. Talonul de participare anexat**, completat şi ştampilat de conducerea firmei.

**b. Copia ordinului de plată** (ştampilat de bancă) privind achitarea "Contribuţiei băneşti de participare la Simpozionul A.A.I.R." în contul A.A.I.R. nr. **RO02RNCB0073049975630001** deschis la B.C.R. - Sector 2, Bucureşti. (AAIR are Cod Fiscal RO13289718)

Data limită de plată şi primire a documentelor indicate mai sus: **15.07.2011**

# TALON DE PARTICIPARE

(auditori)

Al 19-lea Simpozion A.A.I.R.
28 septembrie 2011,
Uzinexport - Bucureşti

Firma

Adresa

Participant nume

prenume

Funcţia

Tel/fax :

E-mail

Contribuţia de participare la Simpozion A.A.I.R.: **95 RON**

Anexăm prezentei:
Ordinul de plată nr. ______ din data ______
în contul A.A.I.R. nr. RO02RNCB0073049975630001 deschis la B.C.R.-Sector 2, Bucureşti (A.A.I.R. are Codul Fiscal: RO 13289718).
Ordinul de plată este ştampilat de Bancă.

**Secretariat A.A.I.R.:**
**Str. Viesparilor, nr. 26, et.3; Ap.10; Sector 2; 020643 Bucureşti**

Tel/fax: 021.210.50.55, 031.405.67.99
Mobil: 0745.11.61.99.

# FLUIDAS

**NATIONAL PROFESSIONAL ASSOCIATION OF HYDRAULICS AND PNEUMATICS IN ROMANIA**


fluidas@fluidas.ro