AN OPTIMIZED DESIGN SOLUTION OF AN OLEO-PNEUMATICALLY ACTUATION DEVICE FOR HIGH POWER CIRCUIT BREAKERS

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Abstract **– The operation of high-power circuit breakers with oleo–pneumatically (nitrogen-oil) actuation device was subject to large debates of the designers and users of circuit breakers, the main concern being the synchronization between the three phases of the circuit breaker. When each of the three phases of circuit breaker are driven by a separate oleo-pneumatically device (MOP) it's assumed that their synchronization of the three breakers poles it's impossible to be achieved. In this paper it's elaborated a design algorithm (for the optimization of these actuators) that leads to the possible simultaneous drive with a single MOP of all the three poles of the high power circuit breakers achieving both the price reduction (because there are two MOPs less) and the synchronous drive of the three poles circuit breakers.**

Keywords: *circuit breaker, oleo-pneumatically device.*

1. THE MOVEMENT EQUATION OF THE CIRCUIT BREAKERS' MOBILE EQUIPMENT INTRODUCTION

The operating diagram of the oleo-pneumatic actuator it's presented in fig. 1 where the gas reservoir (under pressure 1) that is the energy accumulator and, by gas expansion, it creates the necessary oil pressure for the circuit breaker piston movement [1].

The oil tank (3) is separated through a piston (2) from the reservoir tank gas. The supply pipe (4), one for the circuit breakers opening, and the other one for its closing, connects the oil tank and piston cylinder (5). The piston cylinder (5) has the same movement as the mobile equipment of the breaker (for direct drive) or a different movement when an intermediary mechanism is used. The drive piston (6) and the exhaust circuit pipe (7) are also presented in figure 1.

The hydraulic circuit consists of a single pipe for all the three breakers' poles and three individual pipes, one for each pole. Both distinct circuits are interconnected through a distributor. Taking into account that both pipes having equal diameters and that the flow ratio is 3/1, the oil speeds are in a 1/3 ratio, all these lead to different hydraulic resistances and different flow regimes.

The movement equation of the mobile equipment is:

$$m_o \cdot \frac{d^2x}{dt^2} + k^f \cdot S_1 \cdot \left(\frac{dx}{dt}\right)^2 - \\ -\left(p_{ac} \cdot S_1 - p_o \cdot S_2 - F_{rez} \pm m_o \cdot g\right) = 0 \qquad (1)$$

where [2]: m_o - the reduced mass of the mobile equipment at the drive piston; (when there is no intermediary mechanism, it represents the total mass of the mobile equipment of the circuit breaker); x is the mobile equipment displacement/movement, k^f is the equivalent coefficient of hydraulic resistance, p_{ac} is the nitrogen gas tank pressure, p_o is the atmospheric pressure (pressure from the expansion tanks of the exhaust systems), S_1 is the opening/closing piston surface, S_2 is the closing/opening piston surface, F_{rez} is the sum of all resistance forces, the most important of them being: the friction forces in fittings and contacts, as well as the electro-dynamic force.

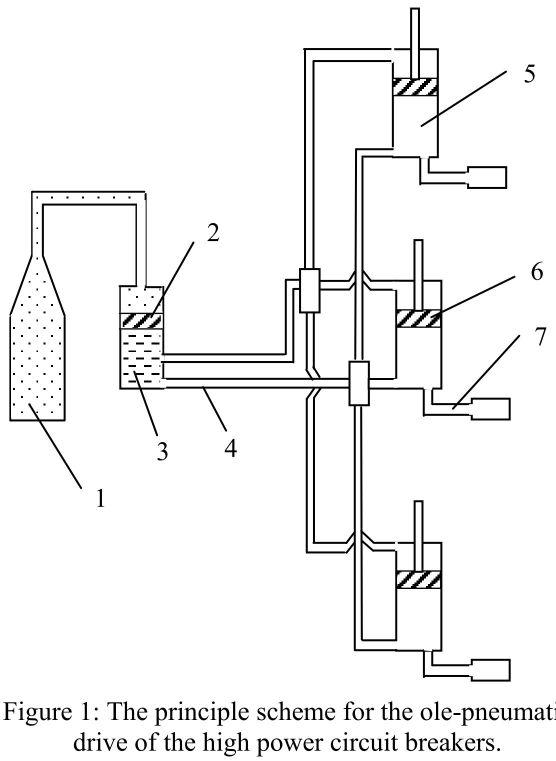

Figure 1: The principle scheme for the ole-pneumatic drive of the high power circuit breakers.

Tacking into account the small values of the circuit breaker operating time, one might consider the nitrogen adiabatic expansion and in these conditions the gas pressure depends on the mobile equipment movement and on the circuit breakers' pipes as in the following equation:

$$p_{ac}(x)=\left(\frac{V_o}{V_o+S_1\cdot x}\right)^{\gamma}\cdot p_{ac\max} \qquad (2)$$

$V_o=30\cdot10^{-3}\,m^3$ - Nitrogen minimum volume, $\gamma=1{,}4$ - the adiabatic exponent of nitrogen, $p_{ac\max}=3\cdot10^7\,N/m^2$ - nitrogen pressure at V_o.

The force that drives the circuit breaker mobile equipment has the equation:

$$F_a=p_1\cdot S_1 \qquad (3)$$

where p_1 is the oil pressure in the piston cylinder and it is computed as the subtraction between the nitrogen expansion and pressure losses due to the hydraulic resistances between the nitrogen tank and the driving piston. This has the expression:

$$p_1=p_{ac}-\Delta p_{lin}-\Delta p_{loc} \qquad (4)$$

where: Δp_{lin} are the linear pressure losses calculated with:

$$\Delta p_{lin}=\sum_{k=1}^{2}k_{1_k}^{l}\cdot v_{c_k}^2=k_1\cdot v^2\cdot S_1^2 \qquad (5)$$

v_{c_k} is the oil speed through a pipe section with constant speed, $k_{1_k}^{'}$ - coefficient for linear pressure losses, on homogenous pipe sections , k_1 - global coefficient for linear pressure losses.

$$\Delta p_{loc}=k_2\cdot v^2\cdot S_1^2 \qquad (6)$$

represents the local pressure losses (through turnings, variable section pipes, speed regulating diaphragms); k_2 is the global coefficient for local pressure losses.

In the equations (5) and (6), v represents the momentary mobile equipment speed. From the equations (1), (2), (5) and (6) one can obtain:

$$m_o\cdot\frac{d^2x}{dt^2}+k\cdot S_1^3\cdot\left(\frac{dx}{dt}\right)^2-\left(\frac{V_o}{V_o+S_1\cdot x}\right)^{\gamma}\cdot p_{ac_{\max}}\cdot S_1+p_o\cdot S_2+F_{rez}\mp m_o\cdot g=0 \qquad (7)$$

In the equations (1) and (7), F_{rez} represents the sum of the resistance forces as follows:

$F_{fr_{MOP}}\cong300\,N$ - friction force in the piston's fitting, that separates the oil from the nitrogen, contact friction force, electro-dynamical force, friction forces in mobile equipment fittings (about $300\,N$). So, for a first approximation, $F_{rez}=600\,N$ and also S_2 might be neglected ($p_o\cdot S_2 << p_{ac}\cdot S_1$). Because the global coefficient k that characterizes the pressure losses depends on the oil's flow regimes, one may hypothesize: the reduced mass m_o is constant and the losses coefficient k is constant, even in the un-stationary flow regime.

If the drive piston is directly linked (united) to the mobile contact, the piston-mobile equipment ensemble concentrates almost all the moving mass and has a translation movement with the same speed as the drive piston speed. In case of an intermediary mechanism, the different motion equipment speeds lead to a nonlinear variation of the reduced mass m_o. In this case, the maximal value for the reduced mass can be considered. The losses coefficient k is constant only for a laminar flow, when the linear and local losses coefficients λ and ξ respectively have constant values. For turbulent oil flow, λ decreases as the speed increases, while ξ increases. In order to obtain satisfactory results, one might consider for these parameters values corresponding to the minimum respectively maximum speed, in the analyzing time duration of the mobile equipment movement (the least advantageous ones). Tacking into account the fact that for a circuit breaker operation (with one MOP for three poles) the nitrogen tank volume variation is about 3 % from the maximum volume and that $p_o\cdot S_2 << p_{ac}\cdot S_1$, the movement equation (7) becomes:

$$\frac{d^2x}{dt^2}+\frac{k}{m_o}\cdot S_1^3\cdot\left(\frac{dx}{dt}\right)^2-\frac{1}{m_o}\cdot\left(p_{ac_{\max}}\cdot S_1+p_o\cdot S_2+F_{rez}\mp m_o\cdot g\right)=0 \qquad (8)$$

The last term from the (8) equation represents the maximum acceleration of the mobile equipment. It is denoted:

$$a_o=\frac{1}{m_o}\cdot\left(p_{ac_{\max}}\cdot S_1+p_o\cdot S_2+F_{rez}\mp m_o\cdot g\right) \qquad (9)$$

The coefficient:

$$r=\frac{k}{m_o}\cdot S_1^3 \qquad (10)$$

characterizes the global hydraulic resistance of the circuit and depends on the S_1 surface section of the actuation piston. Hence, equation (8) becomes:

$$\frac{d^2x}{dt^2} + r \cdot \left(\frac{dx}{dt}\right)^2 - a_o = 0 \quad (11)$$

This equation allows the study of the opening/closing circuit breaker movement for the monopole drive, as well as for the drive with three poles.
The mobile equipment speed will act in accordance with the differential equation:

$$\frac{dv}{dt} + r \cdot v^2 - a_o = 0 \quad (12)$$

The analytical solutions for the equations (11) and (12) as well as the mobile equipment acceleration equation will be:

$$v(t) = \sqrt{\frac{a_o}{r}} \cdot \frac{e^{2\cdot\sqrt{a_o \cdot r}\cdot t} - 1}{e^{2\cdot\sqrt{a_o \cdot r}\cdot t} + 1} \quad (13)$$

$$x(t) = \frac{1}{r} \cdot \ln \frac{e^{2\cdot\sqrt{a_o \cdot r}\cdot t} + 1}{e^{2\cdot\sqrt{a_o \cdot r}\cdot t}} \quad (14)$$

$$a(t) = \frac{4 \cdot a_o \cdot e^{2\cdot\sqrt{a_o \cdot r}\cdot t}}{(e^{2\cdot\sqrt{a_o \cdot r}\cdot t} + 1)^2} \quad (15)$$

The mobile speed variation as a function of travel contact is given by the formula:

$$v(x) = \sqrt{\frac{a_o}{r}} \cdot \left(1 - e^{-2\cdot r \cdot x}\right)^{1/2} \quad (16)$$

The previous relations completely describe the mobile equipment movement.
The maximum speed and acceleration have the expressions:

$$v_{\max} = \sqrt{\frac{a_o}{r}} \quad (17)$$

$$a_{\max} = a(0) = a_o \quad (18)$$

Computing the maximum mobile acceleration allows the computation of the admissible stresses in the cinematic chain in order to verify the mechanical resistance.

2. THE GLOBAL HYDRAULIC RESISTANCE COEFFICIENT

The reduced mass m_o is determined from energetically consideration, using the expression:

$$m_o = \sum_{i=1}^{n} m_i \cdot \left(\frac{v_i}{v_p}\right)^2 \quad (19)$$

where: v_p - the driving piston's speed, v_i and m_i are the speed and the mass of the i-th order element being in translation movement, n - the number of masses in movement. If there is no intermediary mechanism, m_o is the mobile equipment mass. The pressure losses in the hydraulic system have the expression [3], [4]:

$$\Delta p = \Delta p_{lin} + \Delta p_{loc} = \frac{\rho \cdot v^2 \cdot d^4}{d} \cdot \left(\sum_{i=1}^{n} c_1 \cdot \frac{l_i}{d_i^5} \cdot \lambda_i + \sum_{i=1}^{n} \xi_i \cdot \frac{1}{d_i^4}\right) \quad (20)$$

where: $c_1 = 9$ on common pipe portions, $c_1 = 1$ on individual pipe sections, $d = \sqrt{\frac{4S}{\pi}}$ is the equivalent diameter of the driving piston surface.
The k coefficient will have the expression:

$$k = \frac{S_1 \cdot \Delta p}{v^2 \cdot S_1^3} = \frac{8 \cdot \rho}{\pi^2} \cdot \left(\sum_{i=1}^{n} c_1 \cdot \frac{l_i}{d_i^5} \cdot \lambda_i + \sum_{i=1}^{n} \xi_i \cdot \frac{1}{d_i^4}\right) \quad (21)$$

The relations (19) and (21) allow the computation of the global hydraulic resistance coefficient.

3. ALGORITHM FOR THE DESIGN OF AN OLEO – PNEUMATICALLY ACTUATION DEVICE

The differential equation (11) with the solutions (13), (14), (15) and (16) has a physical significance, if the following condition is fulfilled: $a_o > 0$, meaning:

$$p_{ac_{\max}} \cdot S_1 + F_{rez} \mp m_o \cdot g > 0 \quad (23)$$

The minimum surface of the drive piston is:

$$S_{1_{\min}} = \frac{1}{p_{ac_{\max}}} \cdot \left(F_{rez} \mp m_o \cdot g\right) \quad (24)$$

The proposed algorithm has the following steps:

a) the computation of the reduced mass m_o;
b) the computation of the global pressure losses coefficient k;
c) the computation of the maximum (initial) acceleration a_o (eq.9) considering the piston surface S_1 as a parameter;
d) the computation of the r coefficient (eq.10) with S_1 as a parameter;
e) The graphical representation of the drive speed for an imposed time (e.g. the circuit breaker's opening time), considering S_1 a continuous variable;
f) The computation of the minimum section of the piston surface, in order to obtain the imposed speed for the imposed or for an imposed travel contact value.

The paper allows an exhaustive analysis of the oleo-pneumatically driven circuit breaker mobile equipment dynamics.
If we impose the contact movement x_d and the v_d speed of the circuit breaker (the speed value at the moment time when the disconnecting arc appears), one might determine the opening speed under these circumstances as follows:

1. $t = t_{d1} = \frac{x_c}{v_d}$ is initialized;
2. $v = f(S_1)$ for the time t_{d1} is plotted;
3. $S^1{}_{1\min}$ is determined;
4. $x(t)$ for $S_1 = S^1{}_{1\min}$ is plotted and $x(t_{d1})$ is determined
5. if $x(t_{d1}, S_1) \neq x_c$, then t_d takes the value $t_{d2} = t_{d1} \mp \Delta t$;
6. we continue by recalculating steps 1-3 obtaining $S^2{}_{1\min}$, until
   $V(S_1^k)|t_{dk} = v_d$ and $x(t_{dk}, S_1^k) = x_c$

4. NUMERICAL APPLICATION

In order to illustrate the analysis of the drive piston, the following numerical example (associated to a high voltage SF6 circuit breaker IHK 123kV) is taken, for which:
$p_{ac_{\max}} = 300 \cdot 10^5\ N/m^2$, $F_{rez} = 600\,N$, $m_o = 47{,}5\,kg$, $x_c = 90\,mm$ - the contact follow, $x = 230\,mm$ - total travel of the mobile contact, $v_d = 4{,}5\,m/s$ - speed at the time of the mobile and fix contact separation, $k = 10^{12}\ kg/m^7$ - the global pressure losses coefficient.
The speed variation versus section of the surface piston, having as parameter the value t , corresponding to the designed values of the contact follow and to the imposed value of the speed, $v_{imposed} = 5m/s$, is shown in figure 2. It can be noticed that the imposed speed value is reached for two different values of S_1: $S_{1\min} = 4\,cm^2$, $S_{1\max} = 11\,cm^2$ for $t_d = 25\,ms$. The maximum speed value is reached for $S_1 = 7\,cm^2$.
The $k = 10^{12}$ value has been computed according to a given configuration of the hydraulic circuit of MOP device, manufactured by Electroputere. The variation of the contact movement in time, having parameter *k* is plotted in figure 3.
The different values of the global coefficient *k* have been obtained by changing the control diaphragm placed on the hydraulic circuits of the MOP device. From figure 4 it is observed that the speed is increasing as the *k* coefficient deacreases, at t_d given value. The curves from figure 3 and 4 are plotted for $S_1 = 7\,cm^2$, the section for which the maximum value of speed is reached.

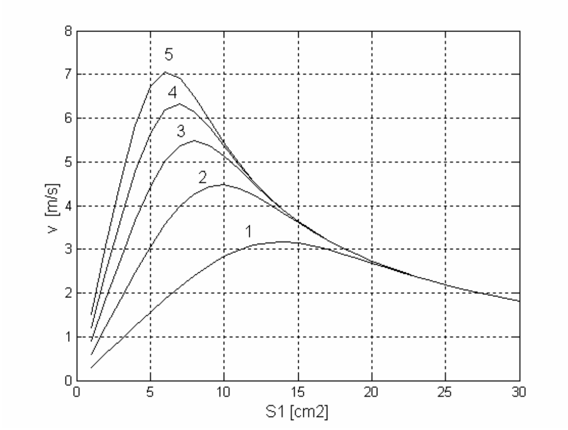

Figure 2: Speed variation versus section of the piston surface, *t* parameter: $k = 10^{12}$,1- $t = 0{,}0050s$, 2- $t = 0{,}010s$, 3- $t = 0{,}0150s$, 4 - $t = 0{,}0250s$, 5- $t = 0{,}0300s$

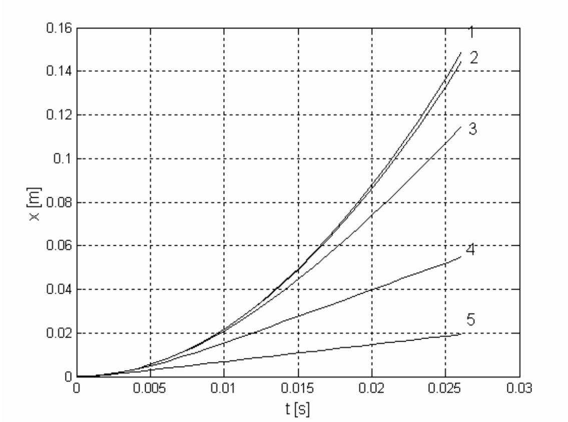

Figure 3: Contact movement- time variation, parameter k: 1- $k = 10^{10}$; 2 - $k = 10^{11}$; 3 - $k = 10^{12}$; 4 - $k = 10^{13}$; 5 - $k = 10^{14}$; $v_d = 5m/s$; $S_1 = 7\,cm^2$

A similar variation is obtained for acceleration, as shown in figure 5. The speed variation in time, considering S_1 as parameter, is shown in figure 6. It can be noticed that for $S_1 = 7\,cm^2$, the speed is approx. constant. If it is necessary to use the MOP device to actuate other constructive variants of the circuit breaker mobile contacts, we have to take into account the different values for the reduced mass. Consequently, as shown in figure 7, it is noticed that for the same t_d value (e.g. $t_d = 20\,ms$) the speed increases as the reduced mass m_o decreases. For small values of m_o, the acceleration increases and the s maximum speed value is reached in short time.

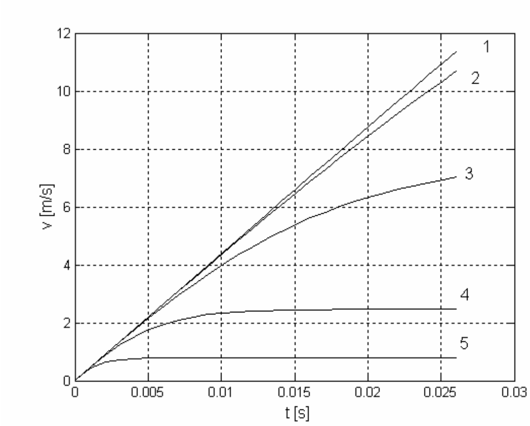

Figure 4: Speed variation versus time, k parameter
1- $k=10^{10}$; 2 - $k=10^{11}$; 3 - $k=10^{12}$; 4 - $k=10^{13}$;
5 - $k=10^{14}$; $v_d=5 m/s$; $S_1=7 cm^2$

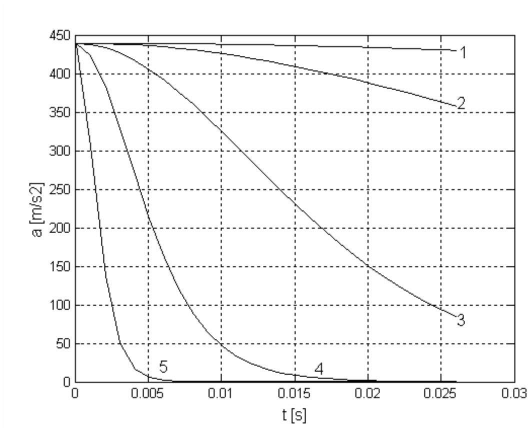

Figure 5: Acceleration time variation, parameter k
1- $k=10^{10}$; 2 - $k=10^{11}$; 3 - $k=10^{12}$; 4 - $k=10^{13}$;
5 - $k=10^{14}$; $v_d=5 m/s$; $S_1=7 cm^2$

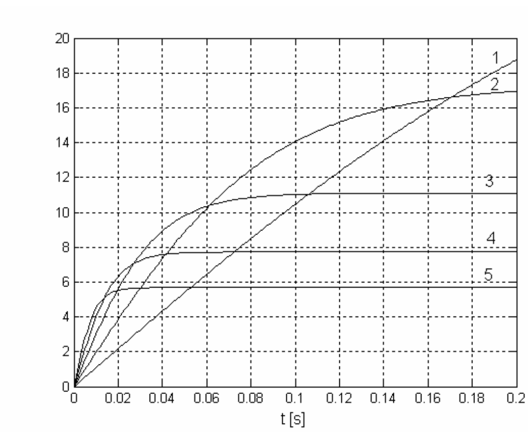

Figure 6: Speed variation versus time, S1 parameter:
1- $S_1=1{,}76 cm^2$, 2- $S_1=3{,}15\ cm^2$, 3 - $S_1=4{,}9 cm^2$,
4 - $S_1=7{,}06\ cm^2$, 5 - $S_1=9{,}62 cm^2$, $k=10^{12}$

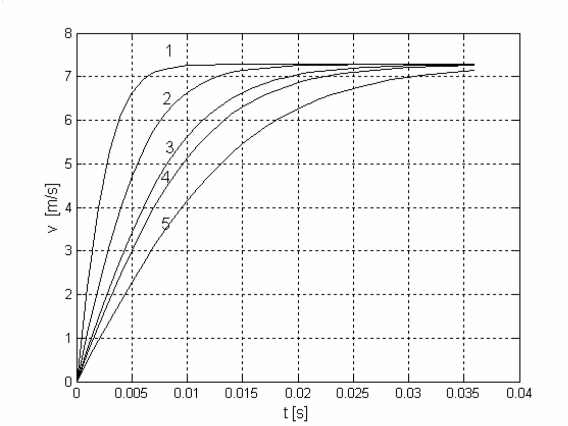

Figure 7: Speed variation versus time, m_o parameter:
1- $m_o=10 kg$, 2- $m_o=20 kg$, 3 - $m_o=30 kg$,
4- $m_o=30 kg$, 5 - $m_o=47{,}5 kg$

The numerical values of the maximum speed v_{max}, when the contact follow value is changed as the designed one ($x_c=90 mm$), due to the technological reasons, are presented in table 1.

$x_c\,[mm]$	85	90	95	100
$t_d\,[s]$	0,0170	0,0180	0,0190	0,0200
$v_{d\ impus}\,[m/s]$	5	5	5	5
$m_o=47{,}5 kg$				
$v_{d\max}\,[m/s]$	5,79	5,98	6,15	6,31
$S_1\,[cm^2]$	7	7	7	7

Tabel 1 The maximum speed value for a given section, contact follow and t_d values .

5. CONCLUSIONS

The paper deals with the optimal design of the oleo-pneumatically device MOP meant for the actuation of medium and high voltage circuit breakers manufactured by the Electroputere Craiova. Knowing the circuit breaker dimensions, the minimum section of the actuation piston surface, for which the maximum value of the mobile contact speed is reached, has been computed.

In order to point out the impact of the section surface, the global losses coefficient or the reduced mass over the movement, speed or acceleration, several curves have been plotted.

Acknowledgments

The paper is dedicated to the 1975- 1980 period, when both the authors have worked, experienced and

tested high voltage circuit breakers at the former Design and Research Institute - Electroputere Craiova (ICMET Craiova nowadays).

References

[1]. Gh. Hortopan, *Aparate electrice*, Editura Didactica si Pedagogica, Bucuresti, 1980

[2]. J. Maksymiuk, *Mecanismele aparatelor electrice de conectare*, Editura Tehnica, Bucuresti, 1970

[3]. N.N. Drăghici, *Conducte pentru transportul fluidelor*, Editura Tehnica, Bucuresti, 1971

[4]. C. Iamandi, s.a., *Mecanica fluidelor*, Editura Didactica si Pedagogică, Bucuresti, 1977

[5]. D. Ionescu, *Indroducere in hidraulica*, Editura Didactica si Pedagogică, Bucuresti, 1977, Ed.II.