ABOUT THE BRAKING OF LINEAR HYDRAULIC MOTORS AT THE END OF MOTION

Ilare BORDEAŞU[1], Victor BĂLĂŞOIU[1], Anton HADĂR[2], Rodica BĂDĂRĂU[3], Mircea POPOVICIU[1], Johann-vVALENTIN KRIER[4]

Abstract: *The braking systems of LHM at the end of motion can be found in any places where the load in motion has a large kinetic energy and a high safety is needed. This paper aims to propose a method and an algorithm for the dimensioning of the braking systems of the LHM at the end of motion. This methods experimentally verified will be the base of a calculus methodology for the dimensioning of LHM with braking at the end of motion in view of operating of hydro mechanical equipment.*

Keywords: linear hydrostatic motor, equation, braking at the end of motion, parameter

1. Introduction

The linear hydraulic motors (LHM) braking systems at the end of motion can be found in the hydro drives of all equipment characterized by large mass (tens or hundreds of tones) and displacement speeds relatively large (up to 0.5 m/s); the hydro mechanical equipments of the hydro energetic and navigation arrangements, rolling mills, machine tools, industrial robots.

Briefly, the braking systems of LHM at the end of motion can be found in any places where the load in motion has a large kinetic energy and a high safety is needed.

Given the importance of these braking systems, this paper aims to propose a method and an algorithm for the dimensioning of the braking systems of the LHM at the end of motion. This methods experimentally verified will be the base of a calculus methodology for the dimensioning of LHM with braking at the end of motion in view of operating of hydro mechanical equipment.

2. The delimitation of the filed of existence of breaking at the end of motion

The criterion that makes the delimitation of the inferior limit of the domain of existence of breaking at the end of motion is the energetic one, given by the relation:

[1] Prof., Hydraulic Machinery of Department, University “Politehnica” of Timisoara, Romania
[2] Prof., Strength Materials of Department, University “Politehnica” of Bucharest, Romania
[3]Asist.drd., Hydraulic Machinery ofDepartment, University “Politehnica” of Timisoara, Romania
[4] Ing., Sraβer 4, 90431 NURNBERG, Germany

$$\varepsilon_m = \frac{E_i}{E_f} \tag{1}$$

where:

$$E_i = F_{\max} U_M^2 / (2 \cdot g)) \text{ (J)} \tag{2}$$

$$E_f = (18 \div 20) \text{ (J)} \tag{3}$$

and:

$F_{\max}$ (N) is the maximum calculus force at the vertical LHM (VLHM). In the case of horizontal LHM (HLHM) instead of $F_{\max} / g$ one takes M_{red} – the rediced mass (kg).

$U_M - C_S \cdot C_A \cdot U_N$ (m/s) - the maximum speed of calculus;

$C_S = 1{,}25 - 1{,}5$ - safety coefficient;

$C_A = 1{,}2 - 1{,}25$ - damage coefficient;

$U_N =$ (m/s) - the calculus rated speed.

If $\varepsilon_m \leq 1$ one must not introduce braking at the end of motion.

If $\varepsilon_m > 1$ one must introduce braking at the end of motion. (BEM)

It must be specified the fact that the value of 18-20 J taken for the final kinetic energy of the load represents the average volume of the breaking energy for steels (kV).This because it is considered that the part undertaking the shock at the load braking is made in steel. For other metals to be used one must evaluate the breaking energy or the mechanical work of deformation.

The upper limit of the domain of existence of the BEM is given by the maximum mechanical work that can be obtained using the BEM. The constructive variants have in their building, elements for energy dissipation, (throttles) and:

$$L_{fM} = M \int_0^{l_{fM}} a_f(Z) dZ = K_L \cdot M \cdot a_{fM} \cdot l_{fM} \tag{4}$$

where M is $F_{\max} / g$ at the VLHM or M_{red} at HLHM;

a_{fm} – the maximum braking acceleration – resulting from the condition that the maximum effort in the parts (the rod of the LHM), connecting bars, bolts, junction plates etc.) to not overcome the maximum admissible value.

$$\tau_{ech} = \tau_{0,2} / c \qquad c = 2{,}5 \text{ - safety coefficient}$$

where

$$\tau_{02} = R_{p0,2} \text{ - the flowing limit.}$$

In the case of pushing LHMs with the rod submitted to buckling, during braking a safety coefficient for buckling must be taken 3,5, value with what the critical buckling force is being determined, and from it a_{fm}:

$$a_{fM} = F_{cr} / M \quad \left(\frac{m}{s^2}\right)$$

In what concerns the maximum barking length: l_{fM}, it results from constructive and technological reasons. Thus, in the case of long LHMs, at which the active run $c \geq 8 \cdot (D+d)$ where D is the diameter of the cylinder hole and D the diameter of the rod it can be taken:

$$l_{fM} = 0{,}8 \cdot (D+d)$$

In the case of short LHMs with $c < 8 \cdot (D+d)$ it will be taken:

$$l_{fM} = 0{,}1 \cdot c \text{ but } l_{fM} \geq 100 \text{ mm.}$$

Finally K_L – under unitary coefficient the value of which depends on the law of variation of the acceleration by report with the displacement; $a_f(Z)$. It is rational to adopt a sin shape dependency $a_f(Z)$:

$$a_f = a_{fM} \cdot \sin\left(\pi \cdot \frac{Z}{l_f}\right) \tag{5}$$

with $Z \in [0, l]$

or, without dimensions:

$$a_r(x) = \frac{a_f(Z)}{a_{fM}} = \sin(\pi \cdot x) \tag{6}$$

where $x = \dfrac{Z}{l_{fM}}$

In this case $K_L = \int_0^1 a \cdot r(x) dx = \int_0^1 \sin(\pi \cdot x) dx = \dfrac{2}{\pi}$

The criterion that delimitates the upper limit of the braking domain at the end of motion is given by the ratio:

$$\varepsilon_m = \frac{E_i - E_f}{L_{fM}} \tag{7}$$

$$L_{fM} \geq E_i - E_f$$

In $\varepsilon_m \leq 1$, the load can not be stopped at the end of motion using the considered procedures.

In $\varepsilon_m > 1$, the load can not be stopped at the end of motion using the considered procedures.

In the case of sin shape laws of variation of the acceleration, imposing the condition $\varepsilon_m = 1$ it results a supplementary relation useful in designing:

$$a_{fM} \cdot l_{fM} / U_M^2 = \pi \cdot \left(1 - \frac{l}{\varepsilon_m}\right) / 4 \tag{8}$$

with the condition a_{mf} and l_{fm} to not overcome the above mentioned limits.

The title of each chapter will be numbered. All figures (charts, screenshots, drawings, illustrations etc) must be numbered consecutively with Arabic numerals and cited in the text. Illustrations may be submitted either as drawings or photographs and placed within the text along with descriptive and concise legends. Figures must be centred, and one blank line before and after them must be left in the text. Figure number and caption must be placed just below the relevant figure. The figures must be in black and white (grey scale). All tables must be numbered and cited in the text. The caption of a table must be placed just above the relevant table.

The total length of the paper should not exceed 8 pages and this should include all text, figures, tables and literature references. Use of subtitles is recommended only if the results section contains several parts (one blank line must be placed before and after main and subheadings, and paragraphs).

Presentation will be clear and concise. Symbols will be defined within a Nomenclature if it is necessary. The SI units will be used.

Descriptions of equipments, apparatus and installations cannot be accepted.

3. Choice of the constructive solution

The choice of the optimum constructive solution is essential because it ensures, on one hand the correct operation of the braking system and on the other hand it avoids the applying of a complex solution in the place where it would not be necessary.

In this paragraph it will be given a criterion of delimitation of the sub-domain of existence of braking at the end of motion with a simple throttle. This can be obtained from the differential equation that describes the braking using the simple throttle:

$$w(dW / dx) = AXW - B(1 - w^2) \tag{9}$$

imposing the boundary conditions

$$x = 0 \quad \Rightarrow \quad W = 1 \wedge dW / dx = 0 \tag{10}$$

$$x = 1 \quad \Rightarrow \quad W = W_m = 1/\varepsilon_m \wedge dW / dx = 0 \tag{11}$$

in the equation (9) $W = u / U_M$

$x = z / l_f$

A is the ratio between the mechanical work of the pressure forces (of braking) and the initial kinetic energy of the load and is described by:

$$A = 12(l_b / \delta)^2 \rho v S_{if} / (MU_M)(S_{if} / (\pi d_f \delta) - 0{,}5) \quad (12)$$

B is the ration between the mechanical work of the load on the braking distance and the initial kinetic energy of the load:

$$B = F_{\max} \cdot l_b / (M \cdot U_M^2) \quad (13)$$

The symbols of the items from the relations (12) and (13) correspond to the notations from fig. 1. and:

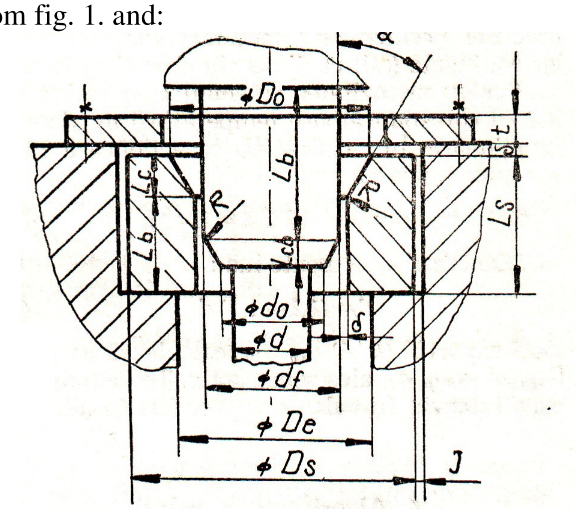

Fig.1

$$S_{if} = \pi(D^2 - d_f^2)/4$$

$$S_i = \pi(D^2 - d^2)/4$$

Because of the boundary conditions (10) and (11) the differential equation (9) can be written:

$$A/R = 1/W_m - W_m = (\varepsilon_m - 1)/\sqrt{\varepsilon_m} \quad (14)$$

With this one can get the delimitation criterion of the sub domain of existence of braking at the end of motion using a simple throttle:

$$\mu = (B/A)_{disp}(\varepsilon_m - 1)/\sqrt{\varepsilon_m} \quad (15)$$

where $(B/A)_{disp}$ – the value available, computed with the available parameters, of the throttle.

If $\mu<1$ one can use, in view of braking at the end of motion a simple throttle.

If $\mu>1$ one can no longer, use, in view of braking at the end of motion a simple throttle, a combined throttle being needed.

In view of correct use of the criterion some considerations must be stated, considerations on some parameters l_b, δ, d_f and v, parameters that enter into the structure of A and B.

For the value of l_b it will be taken the maximum value admitted from the technological and constructive point of view:

$$l_{b\max} \cong 0{,}25 \cdot (D-d) \tag{16}$$

The recommended value for l_b is:

$$l_b \cong 0{,}15 \cdot (D-d)$$

In what concerns the radial gap JOC δ, the minimum value, from the operational and technological point of view is:

$$\delta_{\min} = 0{,}1 \text{ mm} \tag{17}$$

in case of steel/steel couples with polished surfaces.

In the case of using antifriction materials, $\delta_{\min}$ can be diminished but not much, because of the fact of implication of the manufacturing precision of the conjugated coupling surfaces (both sizes and shapes).

For the kinematical viscosity it will be considered the value corresponding to the maximum operating temperature. In the case of LHMacting EHM it will be considered the value:

$$v_{\min} = v_{50^o C} = 10Cst = 10^{-5} \text{ m}^2\text{/s} \tag{18}$$

The diameter d_f can be taken using:

$$d_f = (1{,}1 - 1{,}5) \cdot d \tag{19}$$

So the ratio $(B/A)_{disp}$ can be computed with L_{bmdx}, $\delta_{\min}$ and $v_{\min}$ d_f chosen constructively. All the items from this calculus must be taken in SI units.

4 Calculus algorithm

4.1. The dimensioning of the simple throttle

The simple throttle is being dimensioned from the condition that at the end of braking, the mobile assembly (the load) to be in a rectilinear and uniform motion with the speed $U_f = \sqrt{2 \cdot E_f / M}$. So:

$$F_{\max} = (\Delta p_d)_f S_{if} + (\Delta p_{Dr})_f S_i \tag{20}$$

$$\text{or } AW_m - B(1 + Wm^2) = 0 \tag{21}$$

from where we get:

$$(B/A)_{nec} = \sqrt{\varepsilon_m} / \varepsilon_m - 1 \tag{22}$$

Considering the relations (12) and (13) of A, respectively B, we can get the ration:

$$\left(\delta^3 / l_b\right)_{nec} = \left(B/A\right)_{nec} \cdot k \cdot \delta \cdot \rho \cdot v \cdot U_M \cdot S_{if}^2 / \left(F_{\max} \cdot \pi \cdot d_f\right) \tag{23}$$

where $K_\delta = 12 \div 16$

Taking l_b constructively with the restriction $l_b \le l_{b\max}$, given by the relation (16), one can determine the value of the gap JOC δ:

$$\delta = \left(l_b \left(B/A\right)_{nec} k \delta v U_M S_{if}^1 / \left(F_{\max} \pi d_f\right)\right)^{1/3} \tag{24}$$

for $K_\delta = 12$ we get the minimum necessary value of δ and for $K_\delta = 16$ the maximum value.

It will be checked the restriction $\delta_{\min}$ calculated $\ge \delta_{\min}$ given by (17) or determined technologically or operationally. In case of not checking, l_b will be enhanced with the restriction $l_b \le l_{b\max}$ and the calculus is made again.

In the designs it will be written the diametric gap JOC which must enter the limits $2\delta_{\min} \div 2\delta_{\max}$.

In what concerns the tapered section of engaging, on its geometric parameters there can be made a series of considerations not included in the frame of this work.

The gaps J and S are being taken so the whole volume of oil to pass during lifting through the holes at a oil speed of $(2{,}5 \div 5)$ m/s.

A very important parameter in the operation of the throttle is the sliding surfaces roughness. It must not exceed 0,8 μm.

The same recommendation is in force for the insulating and sitting surfaces.

4.2. Dimensioning of the combined throttle

If $\mu > 1$ it results it must be used a combined throttle in view of braking at the end of motion. This is made from a simple throttle and a variable one, linked in parallel from the hydraulic point of view (fig 2).

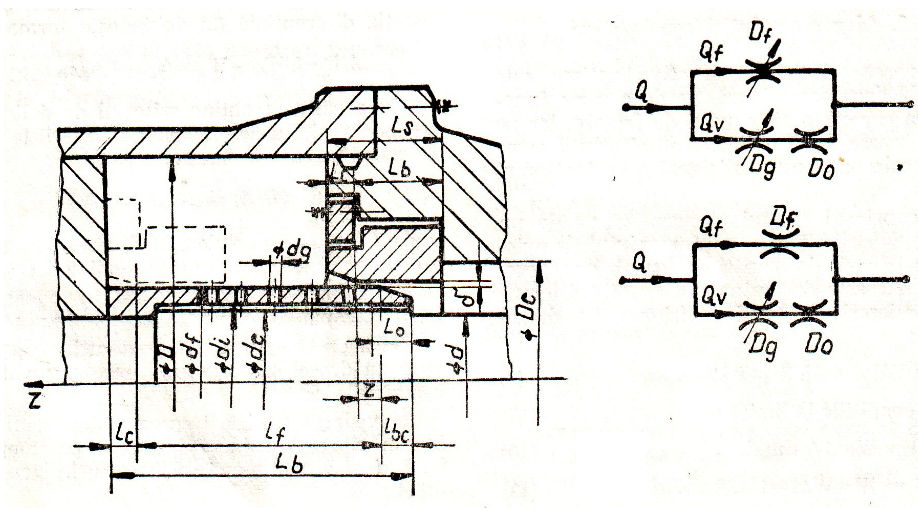

Fig.2

The dimensioning of the combined throttle can be done in two stages:

- the dimensioning of the simple throttle;
- the dimensioning of the variable throttle.

The dimensioning of the simple throttle

This is made using the procedure described at the paragraph 4.1.

The dimensioning of the variable throttle

This can be done using two methods:

A) the analytic method

B) the graphic-analytic method.

A) The analytic method of dimensioning of the variable throttle is done using the following algorithm:

a) The choice of the rule of variation of the acceleration.

It can be chosen any rules of variation of the acceleration ensuring variations without jumps and the boundary conditions (10) and (11). We recommend two variants of laws of variation for the acceleration:

a1) The sin shaped low of variation of the acceleration having the advantage of a optimum dynamic behavior of the system, having the expression:

$$a_f = a_{fM} \sin\left(\pi \frac{Z}{l_f}\right)$$

Or, in relative measures:

$$a_r = \frac{a_f}{a_{fM}} = \sin(\pi \cdot x)$$

a2) The second possibility is that of ensuring the enhancing of the acceleration from 0 to the maximum value a_{fM} by means of the conic section of engaging and of a fraction from the cylindrical part of the braking rove, i.e. for $0 \le x \le (l_f + Kl_b)/l_f$. Then the acceleration is being held constant and equal to a_{fM} up to $x = (0.5 - 0.6)$. On the last section the acceleration diminishes from a_{fM} to 0 using a cos shape law or following a polynomial law.

If the a.1 variant can be applied in any situation, the variant a.2 is recommended especially at relatively long braking distances.

b) Establishing of the maximum braking acceleration.

- For $\varepsilon_m > 0.7$, and stresses in the most loaded part during braking not overcoming 30% of the flowing limit of the material, it will be taken $a_{fM} = (1{,}5 - 2{,}5)$ m/s^2.
- For ε_m between 0.25 and 0.7, and stresses in the most loaded part during braking among 30% and 40% of the flowing limit of the material, it will be taken $a_{fM} = (0.5 - 1.5)$ m/s^2.
- For ε_m < 0.25, and stresses during normal operation (not during braking) in the most loaded part during braking reaching 40% of $R_{p0,2}$, it will be taken $a_{fM} = (0.1 - 0.5)$ m/s^2.

c) The determination of the braking length (distance)

This can be made starting from the obvious equality:

$$18(\varepsilon_m - 1) = K_L M a_{fM} l_f \tag{25}$$

from where we get:

$$l_f = 18(\varepsilon_m - 1)/(K_L M a_{fM}) \tag{26}$$

where:

$$K_L = \left(\int_0^1 a_f(x) dx \right) / a_{fM}$$

and the expression $a_f(x)$ is established previously at the point a).

In the hypothesis of sin shaped variation of the acceleration, l_f can be determined using the relation:

$$l_f = U_M^2 \pi (1 - l/\varepsilon_m)/(4 a_{fM}) \tag{27}$$

After the determination of the braking distance l_f one must do the checking $l_f \leq l_{fM}$ determined according the paragraph 2.

d) The determination of the law of variation of speed. By definition $a_f = du/dt = udu/dz$, from where we get the differential equation:

$$wdw/dx = c \cdot ar(x) \tag{28}$$

which integrated, in the boundary condition (10) and (11) offers the law of variation of speed as a function of z(x).

In the hypothesis of sin shaped variation of the acceleration, the law of variation of speed will have the shape:

$$U(Z) = \left(U_f^2 + U_M^2\left(1+\cos\left(\pi \cdot Z/l_f\right)\right)/2\right)^{1/2} \tag{29}$$

or

$$W = \left(l/\varepsilon_m + \left(1-1/\varepsilon_m\right)\left(1+\cos\left(\pi \cdot x\right)\right)/2\right)^{1/2} \tag{30}$$

e) The calculation of the pressure drop on the throttle.

In view of determining the pressure drop - necessary for braking - on the throttle one can consider the equation of dynamic equilibrium of forces acting on the piston:

$$\Delta p_d(Z) = Ma_f(Z)/S_{if} = F_{\max}\left(1-w^2(x)\right)/S_{if} \tag{31}$$

f) The calculus of the flow rates through throttles.

This is achieved by successive approximations following the subsequent algorithm:

f.1. One considers an approximate value of the oil speed through the simple throttle: V_s.

If $Z = 0 \Rightarrow i = 0 \quad V_{so}^0 = u$

If $0 < Z \leq l_f \;\Rightarrow\; 0 < i \leq m \quad V_{si}^0 = V_{si-1}^f$

i.e. the last approximation made is being taken at the calculus in the previous point $i-1$.

f.2. There are being calculated step by step:

$$\mathrm{Re}_{si}^j = V_{si}^j 2\delta/\nu$$

$$\zeta_{si}^j = 0{,}75\left(64/\mathrm{Re}_{si}^j\right)\left(l_b/2\delta\right)\left(2-U/V_{si}^j\right)$$

$$V_{si}^{j+1} = \left(2\Delta p_{di}/\left(\rho\zeta_{si}^j\right)\right)^{1/2}$$

f.3. One checks the difference:

$$\left|V_{si}^{j+1} - V_{si}^j\right| \leq \Delta \cong 10^{-2}\ \mathrm{m/s}$$

If Yes we go to f.4

If NO, the calculus must be done again starting from f.2.

f.4. With the value $V_{si}^{j+1} = V_{si}^f$ one calculates:

$$Q_{si} = \pi \cdot d_f \cdot \delta_{si}^f$$
$$Q_{vi} = u \cdot Sif - Q_{si}$$

g) The calculus of dependency $n(2)$.

The dependency of the number of holes on the position of the barking bushing by report with the braking rove: $n(2)$ can be obtained after an approximate calculus by successive approximations.

g.1. The determination of the total number of holes in the bushing:

For $Z = 0 \Rightarrow i = 0$ we calculate:

g.1.1 $N^0 = \left(\zeta_{vo}\rho Q_{vo}^2 / \left(2\Delta p_{do} S_g^2\right)\right)^{1/2}$

$\zeta_g^j = 0{,}5 \cdot \left(1 - N^j S_g / S_{if}\right) + \left(1 - N^j S_g / S_1\right)^2$

$\zeta_0^j = 0{,}5 \cdot \left(1 - N^j S_g / S_1\right) + \left(1 - N^j S_g / S_e\right)^2$

where

$$S_g = d_g^2 / 4$$

and $\zeta_{vo} \cong 2{,}45$

g.1.2. With the value N^j one calculates:

variables for Re_g, Re_0 10^5

where:

$$S_l^1 = \pi\left(D_1^2 - d_e^2\right)/4$$
$$S_e = \pi\left(D_e^2 - d^2\right)/4$$
$$V_g^j = Q_{vo} / \left(N^j S_g\right) \cong v_o^j \quad ; \quad d_o^j = 4N^j S_g / P_o$$
$$\mathrm{Re}_g^j = V_g^j dg / v \quad ; \quad \mathrm{Re}_o^j = V_o^j d_o^j / v$$

g.1.3. There are being calculated or adopted from tables λ_g^j respectively λ_o^j, as a function of Re^j, respectively Re_o^j.

g.1.4. The values of ζ_g^j, ζ_o^j are being transposed with function of Re_g^j, Re_o^j if these are les then 10^5.

g.1.4. With these values one computes ζ_v^j in the formula:

$$\zeta_v^j = \zeta_g^j + \lambda_g^j l_g / d_g + \zeta_o^j + \lambda_o^j l_o / d_o^j$$
$$N^{j+1} = \left(\zeta_v^j / \zeta_v^{j-1}\right)^{1/2} N^j$$

g.1.5. One checks:

$$\left|N^{j+1} - N^j\right| \le 10^{-1} \div 10^{-2}$$

If yes, we go to point g.2.
If not, we retake the calculation from g.1.2.
g.2. The determination of the dependency $n(Z)$
For $0 < Z \le l_f \;\Rightarrow\; 0 < i \le m$
g.2.1. One approximates $n_i^o = n_{i-1}^f$
g.2.2. This value is computed:

$$\zeta_{gi}^j = 0{,}5\left(1 - n_i^j S_g / S_{if}\right) + \left(1 - n_j^i S_g / S_1\right)^2$$
$$V_{gi}^i = Q_{vi} / \left(n_i^j S_g\right)$$
$$v_{oi} = Q_{vi} / \left(N S_g\right)$$
$$\mathrm{Re}_{gi}^i = V_{gi}^j d_g / v$$
$$\mathrm{Re}_{oi} = v_{oi} d_o / v$$

g.2.3. There are being calculated or are taken from tables the values λ_{gi}^j ,respectively λ_{oi}^j, and they are being transposed ζ_{gi}^j as a function of Re_{gi}^i, if $\mathrm{Re}_{gi}^j < 10^5$.

g.2.4. One calculates:

$$\zeta_{vi}^j = \left(\zeta_{gi}^j + \lambda_{gi}^j l_d / d_g\right)\left(N / n_i^j\right)^2 + \zeta_o + \lambda_{oi} l_o / d_o$$
$$\Rightarrow \quad n_i^{j+1} = \left(\zeta_{vi}^j \rho Q_{vi}^2 / \left(2 \Delta p_{di} S_g^2\right)\right)^{1/2}$$

g.2.5. One checks:

$$\left| n_i^{j+1} - n_i^j \le 10^{-1} \div 10^{-2} \right|$$

If yes, one goes to point h.
If not, the calculus is retaken from g.2.2.
h) The determination of the coordinates of holes.

This is done either by Lagrange interpolation of the dependency $n(Z)$, resulting the values Z, corresponding to the integers n, or by graphic interpolation.

B) The graphic analytical method of dimensioning makes appeal at the graphic solving of the motion equation of the mobile parts during barking,

$$M \cdot a_f = \Delta p_d \cdot S_{if} - F_{\max}\left(1 - w^2\right) \tag{32}$$

where: M – the reduced mass at the LHM rod eye of all mobile parts (including of the oil from the cylinder).

5. **Verification**

After the dimensioning of the constructive solution a verification if it is necessary. This can be made either by testing on the model or by an another method.

The analytical calculus method, by its volume and complexity of calculi, is recommended for being programmed on a computer and used at dimensioning, its efficiency being higher as it is used more frequently.

In exchange, the graphic analytical method of calculus, by its specificity, is recommended more for checking a given solution, especially if some of the values of the lost coefficients are being approximated with average values, but enough precise for this purpose. This method too can be programmed on a computer with corresponding graphic capabilities.

6. Conclusions

The calculus method and the algorithm offers the possibility of dimensioning (designing) of braking systems at the end of motion for LHMs – the variants with simple throttle and or with combined throttle.

The calculus method shown above was verified on the braking systems at the end of motion existing at the VLHM from the SHEN Porțile de Fier I at the fast valve of the central water inlet and at the plan gates of the lock) the results being good.

In view of constructing of a methodology of designing of the LHM with braking systems at the end of motion, the experimental verification of the proposed method must be done. This program is developed in the laboratory of hydro drives from CCSITEH Timişoara.

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