

BUILDING AN INNOVATIVE, MODULAR, FLEXIBLE, HIGH PRESSURE (70 MPa) HYDRAULICALLY ACTUATED EQUIPMENT FOR FITTING SCREWED MOUNTINGS FOR HYDRO-ELECTRIC POWER STATIONS

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Lucrarea își propune să dezvolte un echipament modular flexibil inovativ pentru montajul asamblărilor filetate la centralele hidroenergetice. Caracterul de nouitate al soluțiilor este centrat pe capacitatea de generare și control a unei presiuni înalte (700 bar) și de utilizare a acesteia în construcțiile inovative ale echipamentului. Este dezvoltat un echipament complex și o tehnologie de lucru pentru montajul asamblărilor filetate la centralele hidroenergetice. Sunt elaborate modele teoretice ale pozițiilor de montaj adaptate diverselor geometrii și configurații, utilizându-se softuri specializate. Lucrarea face parte dintr-un proiect mai larg de integrare a echipamentelor de înaltă presiune în procesul de mențenanță a centralelor hidroenergetice.

This study stresses on the development of an innovative, flexible, modular equipment for the fitting the screwed mountings for the hydra-electric power stations. The newness of the findings consists in the capacity of generating and controlling the high pressure of 700 bar and using it in the innovative constructions of the equipment. There will be developed a complex equipment and a working technology for fitting the screwed mountings in the hydra-electric power stations. There will be created theoretical models of the mounting positions adapted to various geometries and configurations, using specialized software. The present study is part of a larger project of integrating high pressure equipments in the hydra-electric power station maintenance process

Key words: dynamometric keys, high pressure, hydra-electric power stations

1. Introduction

The creation of the equipment goes toward creating a new range of new innovative, complex, high technical level products, consisting of a system of modular high pressure hydraulic equipments, for fitting screwed mountings in hydra-electric power stations, using a software for the maintenance process.

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The modular hydraulic equipment for fitting screwed mountings in hydrotic power stations, software supported in the maintenance process, is made up of:

- the programmable, multiplex, metrologically approved 700 bar electric-hydraulic source;
- one- stage, stud bolt hydraulic stretcher;
- collar move away hydraulic device;
- nut screw up/ unscrew hydraulic device;

The users of this range of products will benefit from using the modular hydraulic equipment system for fitting screwed mountings in the hydrotic power stations by increasing the work quality and eliminating the accidental damage, thanks to the software assisted maintenance process in energetics (hydrotic / thermal power stations.)

The innovative character of this equipment relies on the idea of offering both the modern technology by controlling the maintenance process using a software and the modern equipments for various operations proper to energetics. The software will be developed by using the newly created data base by gathering information at a national and international level.

The equipment is in accordance with the EU standards and settlements (e.g. quality, environment, risk) beginning with the projection stage and improving the equipment while it is executed and until the new products are manufactured.

2. General overview on the modular dynamometric equipment

For a screwed mounting the most important parametre is the value of the torque imposed by the designer. Therefore, in the case of a dynamometric hydraulically actuated machine, the main parametre is precision.

For the unscrewing operation, the torsion moments must be greater than the screwing ones (depending on the placement of the mounting, maintenance manner, atmospheric conditions etc) which imposes in many cases the use of a hydraulic key with a greater torsion moment or the construction of a device attached to the hydraulic key intended to increase the torsion moment.

In the cases in which the mountings were preserved in chemical environments or even outdoors for a long time but without an adequate maintenance, at the level of the contact between the threads of the two mounting elements (nut and screw), there appear a series of micro-solders. In some cases, the soldering of the two elements is so strong that their unscrewing by applying a greater torsion moment will determine the screw shearing. In order to eliminate this obstacle, one should use a device intended to destroy the micro-solders existing at the level of thread contact between the two elements.

When the entrance of the under pressure fluid in the hydraulic cylinder chamber, a sudden displacement of the piston takes place, thus, determining its work start by a shock. In the case when the pawl is not well positioned, the pawl's or the ratchet's sprokets are destroyed depending on the material they are made of. A great importance regarding the correct positioning of the pawl comes from the pawl's pushing spring: both from its placement and its pushing force. Another influence comes from the pawl's guiding surfaces. A great gradient can lead to an „interruption angle” when it gears the ratchet.

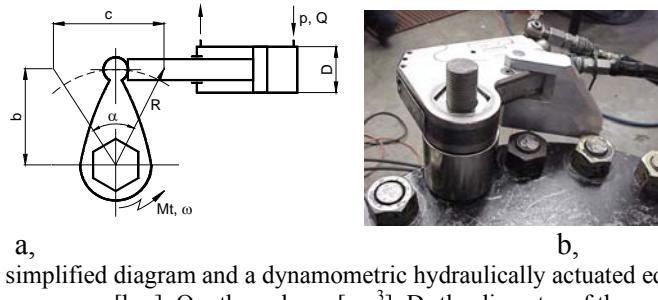


Fig 1. A simplified diagram and a dynamometric hydraulically actuated equipment
 p - is the entry pressure [bar]; Q – the volume [cm^3]; D - the diameter of the cylinder piston;
 R - the maximum lever [cm]; C – the stroke [cm]; α – the angle of the stroke [$^\circ$];
 ω – the angular speed; Mt - the torsion moment [Nm].

2.1 Designing typical structures for hydraulic keys;

For the device that turns the translation motion into rotation motion, a monoblock pawl mechanism – ratchet type.

There are 5 functional blocks, that are going to be analyzed next.

2.1.1 The linear hydraulic engine

The performance of the torsion depends directly on the type of hydraulic engine used, and the double effect hydraulic engines have a high performance.

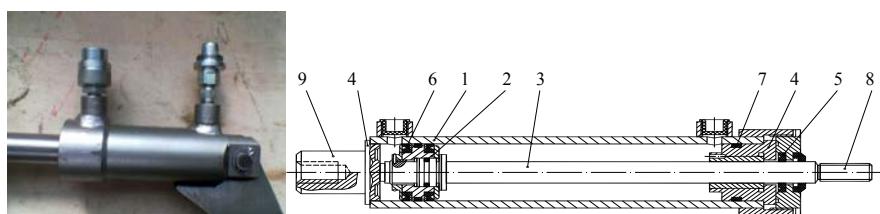


Fig. 2. Linear hydraulic engine used in the construction of the dynamometric hydraulic keys.
Where: 1 – cylinder body; 2 – piston; 3 – rod; 4 – bell; 5 – rod joint sealing; 6 – piston joint sealing; 7 – bell joint sealing; 8 – rod sealing thread; 9 – cylinder sealing thread.

Table 1

CRT/ NO	NAME OF THE FUNCTIONAL PARAMETRE	SYMBOL	MEASURING UNIT
1	Rated pressure	p_n	daN/cm ² , bar
2	Main sizes: - Piston diameter (surface) - Rod diameter (surfacea) - The relation between the active surfaces of the piston in the case of differential cylinders - Piston overall work	D(S) d(S) $\varphi=S/(S=s)$ L	mm (mm ²) mm (mm ²) - mm
3	Rated pushing force	F	daN
4	The average speed of the piston (volume)	V(Q)	m/min, (l/min)
5	Total performance	η	%
6	Cylinder mass	m	kg

2.1.1.1 Calculation of the pushing force

The pushing force F of the piston of the hydraulic engine is calculated using the formula:

$$Mt=F \cdot b \Rightarrow F = \frac{Mt}{b} \quad (1)$$

Where: Mt – moment of torsion; F – pushing force (formula 4.2); b – lever arm.

$$F = F_R + F_F + F_m \quad (2)$$

Where: F_R – the resistance force of the mounting; ; F_F – the friction force; F_m – device resistant force.

$$F_F = \mu \cdot \pi \cdot D \cdot h \cdot p \cdot z \quad (3)$$

Where: μ - friction coefficient; D - the boring of the hydraulic cylinder; h - the height of the fitting levers; z - the number of the fitting levers.

Pressure can also be expressed as in the relation 4, which, if equaled with the relation 2 we obtain the real force that needs to be developed by the hydraulic engine, thus taking into account pressure loss, friction etc.

$$F = p \cdot S = \frac{\pi \cdot D^2}{4} p \quad (4)$$

$$F_R + F_m + \mu \cdot \pi \cdot D \cdot h \cdot p \cdot z = \frac{\pi \cdot D^2}{4} p \quad (5)$$

$$\Rightarrow F_R = \frac{\pi}{4} \cdot p \cdot (D^2 - 4 \cdot D \cdot \mu \cdot h \cdot z) - F_m$$

2.1.1.2 Calculation of the cylinder section and the piston diameter

Cylinder sectioning (boring) is calculated according to the way the rod is working. In our case, the rod works under compression, the section of the cylinder can be rendered by the relation:

$$S = \frac{F_r}{p} \quad (6)$$

Where: p - maximum working pressure; S - active surface of the piston.

By replacing the force and pressure values in relation (6), the active surface of the piston is determined.

The diameter of the boring D of the cylinder is determined from the following relation:

$$D = \sqrt{\frac{4S}{\pi}} \quad (7)$$

Section s of the rod results from the relation:

$$\varphi = \frac{S}{S-s} \quad (8)$$

Where: $\varphi = 1,4 \div 1,6$ - section quotient; S - cylinder section; s - rod section;

By replacing the values S și φ , in relation (8) the value of the section s of the hydraulic cylinder rod is obtained.

$$s = S - \frac{S}{\varphi} \quad (9)$$

The diameter of the rod d is determined from relation 10:

$$d = \sqrt{\frac{4s}{\pi}} \quad (10)$$

The movement between the piston and the cylinder is established according to the cylinder diameter as follows: $J=0.07$ mm. For $D \leq 70$ mm, then $J=0.1$ mm. For $D=70-120$ mm, $J=0.15$ mm. For $D > 120$ mm.

2.1.2. The mechanism of turning the translation movement into an intermittent rotation movement

2.1.2.1 Ratchet admeasurement

The pawl mechanism is an elementary mechanism, different from the ratchet mechanism, both from a kinematical as well as from the designing point of view, as a series of pawl mechanisms cannot be used in force mechanisms.

In order to project the ratchet, we initially adopt the exterior diameter of the ratchet depending on the opening with the S – key, the material type used and the number of sprokets; then the number of sprokets geared z_a is taken into account according to the construction manner chosen.

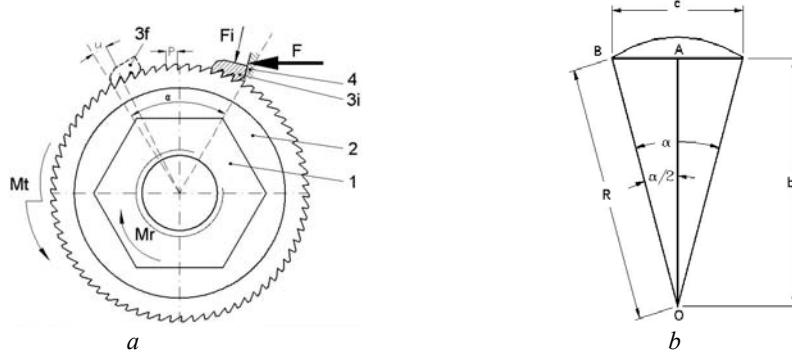


Fig. 3 the Constructive-functional diagram of the pawl- ratchet mounting

1 – screw or nut; 2 – ratchet; 3i- ratchet in the initial and final position; 4 – pawl gearing element (arm); M_t – torsion moment developed by the key; M_r – the resistance torsion moment developed by the mounting element; F_i - pawl pushing force (produced by the spring); F – the pushing force produced by the hydraulic engine; u – the angle corresponding to one step; α - the angle of the active stroke.

The total number of the ratchet sprokets is calculated by the relationship:

$$n = \frac{360}{u} \quad (11)$$

Where u is the angular step.

By applying the trigonometrical functions *sinus* and *tangent* in the right angle ΔOAB we will obtain the value of the angle α (relation 12) and of the lever arm br (relation 13).

$$\sin \frac{\alpha}{2} = \frac{c}{R} \Rightarrow \alpha = \frac{1}{2} \cdot \arcsin \frac{c}{2 \cdot R} \quad (12)$$

$$\tg \frac{\alpha}{2} = \frac{c}{br} \Rightarrow br = \frac{c}{2} \cdot \ctg \frac{\alpha}{2} \quad (13)$$

Angle α can also be determined by the number of sprokets of the ratchet:

$$\frac{z_u}{z_t} = \frac{\alpha}{360^0} \Rightarrow \alpha = \frac{z_u}{z_t} \cdot 360^0 \quad (14)$$

Where: z_u – the number of sprokets used in a stroke; z_t – the total number of sprokets of the ratchet.

For the annular keys, the width of the spokets should not be bigger than the height of the screw/ unscrew nut. The width of a sprocket is calculated using relation 15.

$$B = \frac{F}{q_a} = \frac{F_t}{q_a \cdot \cos \alpha} \quad (15)$$

Where: q_a - is the pressure admitted on the width unit, F_t - tangent force;

The step of the ratchet's sprokets is calculated using relation 16 from the condition of flexure resistance:

$$p = k_p \cdot 3 \sqrt{\frac{M_t}{z_t \cdot \psi \cdot \sigma_{ai}}} \quad (16)$$

Where: k_p - coefficient with values between 1,75 and 1,8; σ_{ai} - flexure admitted resistance;

Ratchet in depth diameter is calculated with relation 17

$$D_i = D_e - 2 \cdot h_d \quad (17)$$

Checking of the sprocket at flexure is done with relation 18:

$$\sigma_i = \frac{12 \cdot M_t \cdot h_d}{z \cdot m \cdot b \cdot a^2} \leq \sigma_{ai} \quad (18)$$

Where: σ_{ai} - is the admitted flexure resistance; h_d - sprocket height; z - number of sprokets; m - module; b - sprocket width; a - sprocket thickness;

The sprocket width is checked upon both at the contact with the pawl and at the contact with the screw (according to fig. 4).

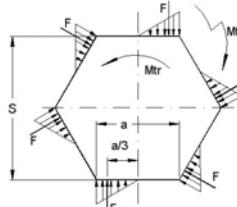


Fig. 4 Force distribution at the contact between the nut and the contact element

2.1.2.2 Pawl admeasurement

The exterior diameter is adopted, the angle at the centre corresponding to a sprocket, the width of the denture and the number of sprokets geared.

The exterior diameter is chosen on clearance grounds, taking into account the key opening.

The angle at the center corresponding to a sprocket is chosen so that we could benefit from the whole piston stroke: α_d [°]

According to the tilting arm equation of moment, one can write:

$$\sigma_i = \frac{M_i}{W_z} \quad (19)$$

Where : W_z – polar moment of inertia ; M_i – flexure moment.

$$W_z = \frac{b \cdot h^2}{64} \quad (20)$$

Where : b – pawl width, equal with the width of the ratchet; h – sprocket height.

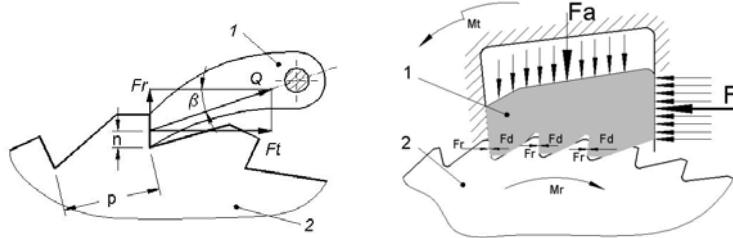


Fig. 5 Forces that appear at the pawl – ratchet mounting

$$M_i = F_d \cdot \frac{h_d}{2} \quad (21)$$

Where: $F_d = \frac{F}{z_a}$ is the radial force that operates on a sprocket (fig. 6)

$$\text{Check upon shearing is done using the relation: } \delta_f = \frac{F}{A_f} \leq \delta_{adm} \quad (22)$$

Where: δ_f - is the real shearing effort; A_f – the surface affected by shearing;

$$A_f = \pi \cdot d \cdot \frac{n_z \cdot \alpha_d}{360} \cdot b \quad (23)$$

$$\text{Check upon squeezing is done with the relation: } \sigma_c = \frac{F_d}{A_c} \leq \sigma_{ac} \quad (24)$$

Where: σ_c – real effort of compression; $A_c = b \cdot h_d$ - the surface affected by squeezing;

In order to obtain a greater reliability for the pawl- ratchet gearing, these two elements will be processed by plastic deformation and then submitted to a rolling in order to shape the two elements at the contact sprokets' flanks. Following the rolling, these pieces will be submitted to a heat treatment to obtain the prescribed hardness.

Lever admeasurement:

The admeasurement calculation is done because of the condition of resistance to bursting. In order to resist the bursting effort, the shape of the lever,

needs to coil the solid body of an equal resistance, whose thickness equals the minimum lever thickness.

$$A = \frac{F}{\sigma_a} \text{ [mm}^2\text{]} \quad (25)$$

Where: A – critical section area; F - the force that operates on the lever; σ_a – the admitted tension at bursting;

By replacing the admitted values of the force and tension at bursting in relation (25), one obtains the value of the critical section A [mm²].

In the critical area, the section has a rectangular shape : $A=b \cdot l$ (26)

$$\text{Out of relation (26) one obtains : } l = \frac{A}{b}.$$

$$\text{Lever checking is done using the relation: } \sigma_{ef} = \frac{F}{A} \leq \sigma_a \quad (27)$$

Where: σ_{ef} is the real bursting effort.

The force achieved by the hydraulically actuated dynamometric key is calculated by relation 28:

$$P = Mt \cdot \omega \text{ [kW]} \quad (28)$$

The torsion moment can be a parametre that can be referred to according to more factors: pressure, piston diameter, stroke, roughness, UAH rated capacity, the features of the oil used, the type of application, the precision of execution of the component elements, sealing, temperature, anti-coupling arm etc.

$$Mt = \frac{\pi}{8} \cdot p \cdot c \cdot \left(4 \cdot b^2 \cdot \frac{\sigma_a^2 + p \cdot \sqrt{4 \cdot \sigma_a^2 - 3 \cdot p^2}}{\sigma_a^2 - 3 \cdot p^2} - 8 \cdot b \cdot \sqrt{\frac{\sigma_a^2 + p \cdot \sqrt{4 \cdot \sigma_a^2 - 3 \cdot p^2}}{\sigma_a^2 - 3 \cdot p^2} \cdot \mu \cdot h \cdot z} - k_m \right) \cdot ctg \left(\frac{z_u}{z_t} \cdot \pi \right) \cdot k_c \quad (29)$$

The hydraulic key and the operating unit diagram is presented in fig. 6 and is made up of:

- D1 – electric manifold diagram 01; D2 - electric manifold diagram 01, both having the nominal port DN6, charged at 24 V or 110 V (by replacing their spools);
- Valves: Sp – safety valve; S1, S2, S3 – one way valves
- M- manometer 0.....1000 bar;
- SD - reversible valve
- ME - electric engine having the rotor in a 1,5 kW short circuit, a revolution n=1500 rot/ min, symbol ASI 90 L -24 – 4.
- MH – hydraulic multiplier

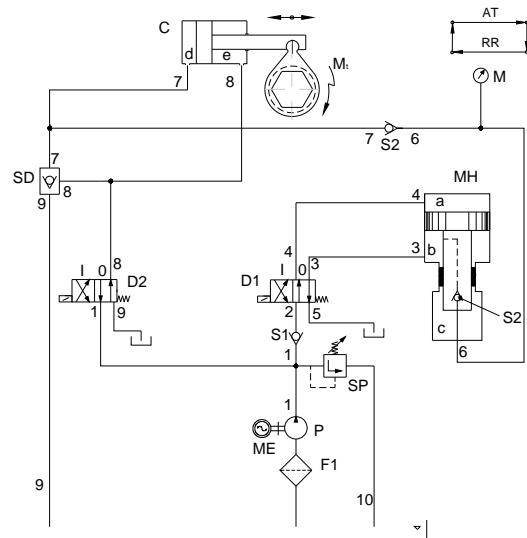


Fig. 6 Hydraulic diagram of the key-pressure unit functional mounting [6]

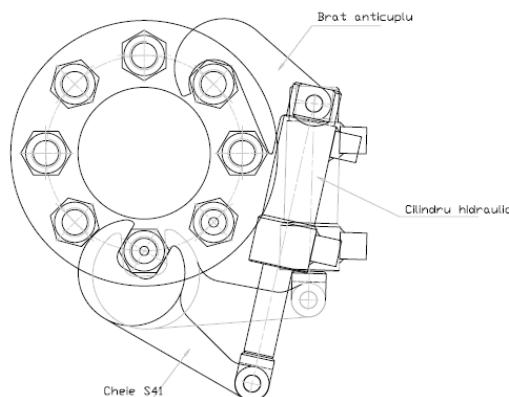


Fig. 7 Assembling diagram of the functional mounting : dynamometric key [6]

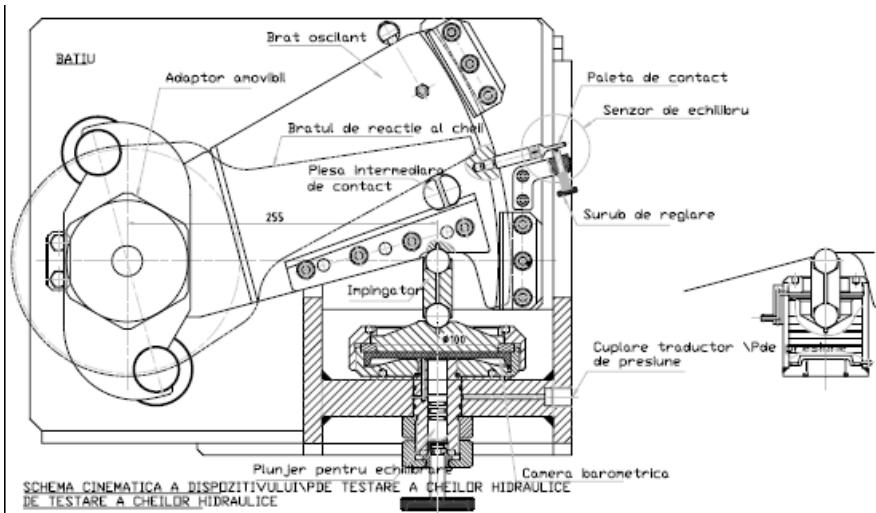


Fig. 8 Assembling diagram of the functional mounting: hydraulic key – essay stall

3. Conclusions

In this study, we presented and analyzed the need of controlled threading and that of the use of dynamometric keys in the threading of the screwed mountings.

We presented a model of organological calculation for the dynamometric hydraulically actuated keys, as well as a series of phenomena that are characteristic to the use of these keys. Regarding the information presented on the present development tendencies in the field of equipments of screwing/unscrewing screwed mountings, we can say the following:

- there is the tendency of increasing the work pressure and also of producing small size elements per power unit;
- there is the tendency of using the high quality materials and fluids, heat treatments and thermic-chemical treatments, or both, as well as of using processing equipments that would ensure the highest precision for the key components;
- improvement of the reliability is achieved by using special materials and especially superior oils that would preserve their properties in time;
- there is the tendency of broadening the practicability field of the hydraulic dynamometric keys;
- the separation of the key from the pressure source proper to the hydraulically and pneumatically actuated dynamometric keys, can be considered a disadvantage from the point of view of the key compactness but also an advantage from the point of view of maneuverability;
- the innovative results obtained and their applications.

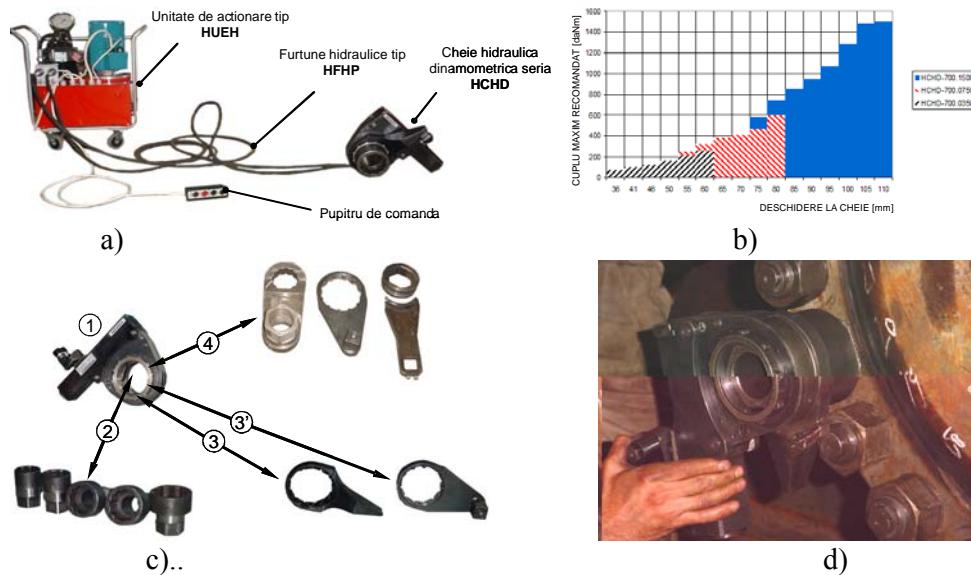


Fig 8. Dynamometric hydraulic keys, contract no. 93 PNCDI 1 [6]

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