STUDY ON STRAINS AND DEFORMATIONS OCCURRED INSIDE THE ELEMENTS OF THE MECHANISM TOOTHED WHEEL – RACK THAT EQUIP MOBILE DRILLING INSTALLATIONS

DANIEL-ALIN RADU[1], SORIN MIHĂILESCU[2], GABRIEL PRAPORGESCU[3]

Abstract: Drilling installations are used for digging up galleries by drilling - blasting method in the mining industry and for hydro technical constructions, as well for digging up tunnels. They are equipped with one or several handling facilities that support and operate the drilling apparatus with which mine holes are drilled. An important subassembly of the drilling installation is the rotation system that is part of the handling facility; this one achieves a circumferential movement of the drilling apparatus, parallel with the face of the gallery. This system makes the object of this paper.

Key words: mobile drilling installations, rotation system).

1. INTRODUCTION OF THE ROTATION SYSTEM

The researches carried out at the University in Petroşani (Romania) have finalized with the homologation and the production of two types of drilling installations, one equipped with one handling facility (IMP-1 type) and shown in figure 1 and the other one equipped with two handling facilities (IMP-2 type).

The rotation system of the handling facility (Fig. 2) is made of one toothed wheel and two hydraulically driven racks, which transform their movement of translation into a movement of rotation of the toothed wheel to which the handling facility is attached. Figure 2 includes: 1 - rack; 2 - pistons attached to the ends of racks; 3 - toothed wheel; 4 - hydraulic cylinder; 6 - housing; 7 - connecting ears with the handling facility; *p* - pressure of the working fluid. There has to be underlined the fact

[1] *Lecturer, PhD, Eng., University of Petroşani*
[2] *Assoc. Prof., PhD, Eng., University of Petroşani, mihailescus@gmail.com*
[3] *Assoc. Prof., PhD, Eng., University of Petroşani, gpraporgescu@gmail.com*

that special limitations with respect to its overall dimensions have been enforced when devising and designing the rotation system of the handling facility.

Fig. 1. Mobile drilling installation - IMP-1 type

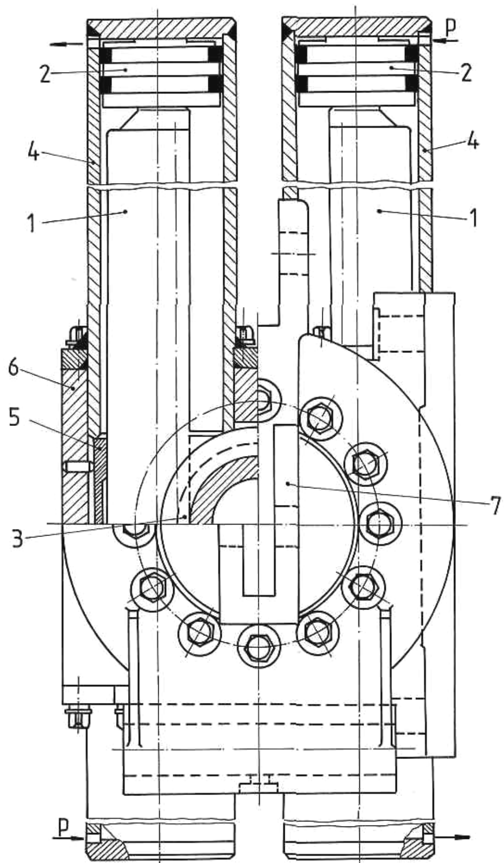


Fig. 2. The rotation system - part of the handling facility

2. THEORETICAL ASPECTS AND RESULTS

Because of limitations imposed by its overall dimensions, special problems regarding devising, designing and production have come up in relation to the rack, reason for which this paper deals only with this aspect.

Figure 3 shows the diagram of the rack, the bearing manner and the load which act upon it and where: F_t - tangential load that act on the teeth of the driving gear made up by the rack ad the toothed wheel and which is equal to the load developed by the piston of the hydraulic cylinder; F_r - radial load born inside the driving gear; 1 - x and x – the momentum position of the pistons located at the ends of the rack (its bearings), compared to the gearing point with the toothed wheel.

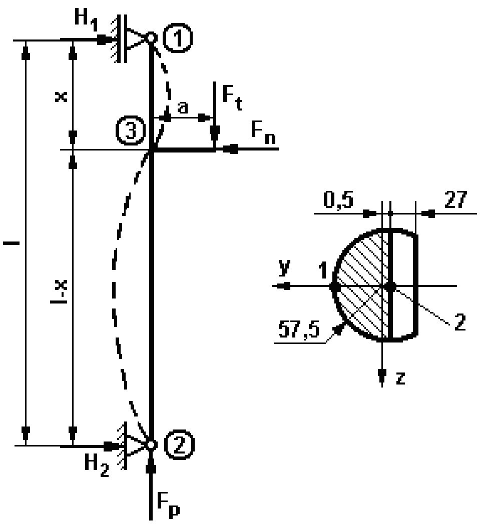


Fig. 3. Calculation pattern for the rack

The maximum bending moment in the rack shall be accomplished for the abscissa x given by the equation:

$$x = \frac{l}{2}\left(1 - \frac{F_t}{F_r} \cdot \frac{a}{l}\right) = 320 \text{ mm} \tag{1}$$

where: $F_t = \pi D^2 p/4 = 12272p$; $F_r = F_t \mathrm{tg}\alpha_0 = 4466p$; $l = 747$ mm - length of the rack; a = 39 mm - distance between the gearing point and the geometrical axis of the rack; α_0 = 20° - the gearing angle; p - the working pressure in the hydraulic cylinders; D = 125 mm - the diameter of the piston.

The equation of the bending moment in the rack, in the current section x is:

$$M_i = F_r x\left(1+\frac{x}{l}\right)+F_t a\left(1-\frac{x}{l}\right) \tag{2}$$

By taking into account the maximum bending moment and the compression force in the rack, there shall be obtained the maximum stretching and compression stresses in points 1 and 2 of the most stressed area, in relation to the working pressure:

$$\sigma_{t\,max} = \frac{M_{i\,max}\cdot y_1}{I_z} - \frac{F_t}{A} = 60{,}25\,p;$$

$$\sigma_{c\,max} = \frac{M_{i\,max}\cdot y_2}{I_z} - \frac{F_t}{A} = 52\,p \tag{3}$$

where: I_z = 1,2^6 mm^4 is the minimum inertia momentum of the rack cross section; A = 2654 mm^2 the rack minimum cross section area.

The material from which the rack is made of is 40Cr10, with the resistance to flow of σ = 780 N/mm^2, and the stress displayed at the contact pressure is of σ_H = 1250 N/mm^2.

According to the equation (3), the stress σ_{tmax} exceeds the material resistance to flow for values of the working pressure which are higher than 15 MPa.

The maximum arrow of the rack is acquired in the gearing point for $x = l_1$ = 107 mm and can be acquired with the help of the following equation:

$$u_3 = \frac{l_1(l_1 - l)}{3\cdot l\cdot E\cdot I_z}\left[F\cdot a(l-2l_2)+F\cdot l_2(l-l_2)\right] = 0{,}0679\,p$$

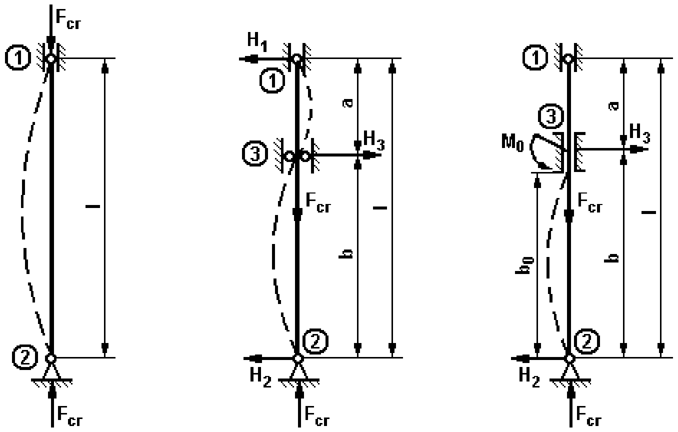
where for p = 15 MPa, there results an arrow of u_3 = 1 mm, fact that would lead to a totally inappropriate operation of the driving gear.

This is the reason why two measures have turned out as necessary: the introduction of a sliding bearing 5 (Fig. 2) which prevents the deformation that might occur when the rack is bent within the gearing area and the introduction of a safety valve in the supply hydraulic circuit for avoiding pressures higher than 13 MPa that might occur due to some handling errors of the human operator.

As the rack is submitted to a stress between the gearing point and the active piston and is part of elastic bars, there has been performed a study on its static stability.

Accordingly, the rack has been assimilated to a bar which is considered in three bearing positions: (Fig. 4), i.e.: a double joint bar (Fig. 4.a); butt joint bar and a bearing joint in an intermediary point, in point 3 (bearing 5 in Fig. 2), when a clearance between it and the rack appear due to the wearing process (Fig. 4.b); the butt joint bar and an intermediary bearing in point 3 which don't allow rotation in this point (bearing 5 in Fig. 1) when this bearing is not worn out (Fig. 4.c).

There has to be underlined the fact that, by construction, the bearing 5 (Fig. 2) is permanently lubricated.



a) b) c)

Fig. 4. Calculation patterns for the rack

For the first bearing manner (Fig. 4.a), the buckling takes place in the plastic domain, as the elasticity ratio $\lambda = l_f/i_{min} = 640/21{,}2 = 30 < \lambda_0 = 50$; accordingly, it results a safety ratio to buckling of:

$$c = \frac{\sigma_f}{\sigma} = \frac{461 - 2{,}26\lambda}{4{,}6p} = \frac{85{,}47}{p} \tag{4}$$

where $\sigma = F/A = 4{,}6p$.

According to equation (4), it results that for a compulsory safety ratio of $c = 6$, the working pressure cannot exceed the value of $p = 14$ MPa, smaller than the nominal pressure $p_N = 15$ MPa.

For the second bearing manner (Fig. 4.b), the critical force shall be determined with the help of the following equation:

$$\frac{1}{(kb)^2}\left(1 - \frac{kb}{tgkb}\right) + \frac{1}{3}\frac{a}{b} = 0 \tag{5}$$

where: $k^2 = F_{cr}/EI_z$; F_{cr} - buckling critical force; $E = 21\times10^4$N/mm^2 - modulus of elasticity; $a = 107$ mm; $b = 640$ mm (for the most adverse situation of the rack bearing).

After resolving the equation (5), there shall be obtained k=0,005; this one helps for acquiring the minimum value of the critical force $F_{cr} = 6303\times10^3$ N and the safety ratio $c = F_{cr}/F = 513{,}6/p$ whose value is 6 has been reached for a pressure $p = 85.6$ MPa, much higher than the nominal pressure $p_N = 15$ MPa.

For the third bearing manner (Fig. 4 c), when the load reaches the critical value, the rack changes its shape only within the area 2 - 3 (i.e. the compressed part). In this situation, the most adverse situation shall be attained for b = 540 mm.

As $\lambda = l_f/i_{min}$=60,18 > 50, the buckling occurs in the elastic domain and the safety ratio to buckling can be obtained with the help of the following equation:

$$c = \frac{F_{cr}}{F} = \frac{\pi^2 EI_z}{(2b)^2} \cdot \frac{1}{12272\,p} = \frac{174}{p}$$

consequently, it results that for c = 6, the value of the pressure p = 29 MPa, i.e. twice higer than the nominal pressure.

3. CONCLUSIONS

The safety valve in the hydraulic circuit of the rotation system that is part of the handling facility shall limit the pressure maximum value to 15 MPa, value that can be reached during its operation. This pressure attained and as a result of the stress to bending, a strain near the resistance to flowing is reached in the rack without any intermediary bearing; this aspect has made necessary the introduction of a supplementary bearing. In this situation only compression stress shall occur in the gearing area with the rack at a zero value of the arrow (close gearing between the rack and the toothed wheel).

The introduction of an intermediary bearing has also been necessary for proving a suitable safety ratio to buckling. All through the operation period of the mechanism made up by the rack and the toothed wheel, there were not any interventions necessary for repairing or for replacing any component parts.

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