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Considerations on Features of the Stress Analysis Applied to Taper Pin-Box Shouldered Connections with Special Screw-Thread Profile

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Abstract

The taper shouldered connections with special screw-thread profile are practically used on such pin-box joints for large diameter drill stems and bring in some particularities: screw loading line is altered by the tapered shape; asymmetric profile affects the flexural rigidity and stress level; load distribution is obviously uneven.
The multiple start threads bring in a new parameter into account: the load unevenness over each "i" screw start.
This paper proposes the principles of strength calculation of contact, shearing or flexion analyze, for taper pin-box connections. The cases in analyze refer to special threads with asymmetrical screw-thread profile.

Key words: *large diameter drilling, taper shouldered connection, pin-box, special asymmetrical screw-thread profile, multiple start thread.*

Preliminary Considerations

In large diameter drilling the drill stem has a similar structure as in conventional oil and gas domain; the main differences refer to larger dimensions, also to the currently inverse mud circulation. Furthermore, the joint system for drilling pipes can use one of the solutions: threaded, flanged or bayonet. The case of medium or reduced stem diameter, it can be used the threaded pin-box connection.

The thread connections of shouldered type (Fig.1., *a*) are composed of the two ends of the pipes being joined into the dedicated design, namely pin-box, and they function as pre-load assemblies.

The conic shapes used in such joints are smaller than the ones of joints used in traditional oil and gas drilling. The large size claims for a combined structure of components (pin and box) to be separated part from the pipe body and their bolt joining.

The thread used herewith has got an asymmetrical triangular generating profile (Fig. 2, *d*). For technological reasons this profile has a top angle of 60^o (with $\alpha = 15^o$, $\gamma = 45^o$ and $\alpha + \gamma = 60^o$). The angle of the live flank having smaller value allows for taking over the thrust loads in the thread [2], [5].

Figure 1 presents – for the threaded connection – the separated parts, fitted stage and the main geometrical features for this asymmetrical screw shape.

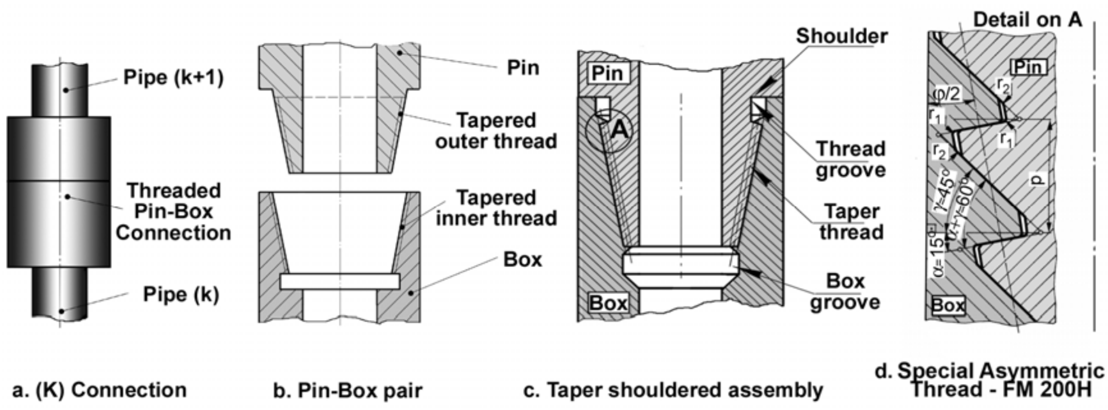

Fig. 1. Constructive and dimensional particularization of the threaded pin-box connection when used for large diameter drilling stem

Load and Strain in Thread Shouldered Joints

The threaded assemblies subject to our review refer to the pin-box pair through which drilling pipes are joined that composes the drill string. Loading in the string is so applied also to its joints. However not every such assembly is identically loaded: the higher loaded are those assemblies positioned towards the well hole, on which the thrust load given by the component weight cumulates.

In calculation practice – for the specific case of large diameter drilling – there are two reference cases to be noticed as for drill string assemblies that are deemed to be the critical cases for the loading of drill string and its joints [2], [5], [9]:

- *Case I* – in handling the drill string (when trip in and trip out) the hanged string is exclusively axially stressed by the component weights the maximum load corresponds to *n* rank assembly at completion of the string, wherever the maximum depth is reached in the well hole;
- *Case II* – while drilling – the string is compositely strained at tensile given by axial resultant $(G_{tot}–W)_n$ and torsion with the drilling M_d torque.

Figure 2 shows in scheme such cases of loading and the corresponding sectional load diagrams. *Case II* shows in addition the diagram M_b, referring to any possible supplementary bending stresses (accidental loads that are not generally considered).

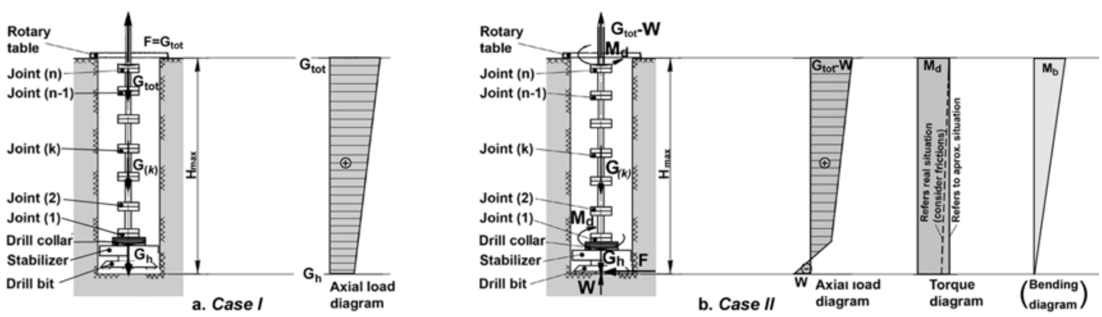

Fig. 2. The load and stress diagrams for the drill stem in both specific loading cases

Principles in Stress Calculation of Threaded Joints

The principles for calculation of pin-box threaded joint components go towards two directions:
- calculate the pin and box (bodies) on tension load, calculation used for sizing;
- verification specific to the thread of both components on contact, shear and bending.

There are two necessary comments referring to the first aspect:
- Dimensioning of pin and box applies differently in sections released by the configuration required to get the thread; for the pin, the critical section is in the closeness of the shoulder provided with coming out or releasing; for the box the area of critical section is the same end section of the thread (or releasing);
- The maximum load for dimensioning is done at depends on several elements:
  - total weight of the string, depending on the drilling depth, the string typo dimension, the type of bottom assembly, degree of the well hole filling, etc.;
  - the bit thrust (WOB) in use (usually recommended at 30-50% of the weight of bottom assembly); the smaller the average WOB, the bigger the axial component of stress composed of tension-torsion;
  - the drill torque, M_d, - importance of this torsion compared to the axial load resulted while drilling, $(G_{tot}–W)$.

To highlight it better, dimensioning of pin may be attained in design practice by starting from getting its dimensions in the area of releasing:

- consider the inside diameter of the releasing of pin D_i – required by the size of drill pipes;
- consider the outside diameter of the pin in the area of releasing, $D_e=\lambda\cdot D_i$;
- for dimensioning in *Case I*, starting from the condition of tensile resistance the outside diameter, results in:

$$D_e = \sqrt{\frac{4F_{tot}^{t}}{\pi \cdot \sigma_{a,t}} - D_i^2} \tag{1}$$

- for proper dimensioning in *Case II*, when stress is composed by tension with torsion, starting from the same inside diameter, D_i,, being required, the outside one may be determined with the relation:

$$D_e = \sqrt{\frac{4 \cdot k_s \cdot \left(F_{tot}^{t,t} - W\right)}{\pi \cdot \sigma_{a,t}} - D_i^2}\ , \tag{2}$$

where factor k_s observes that load is composed (k_s=1.25...1.35) [1], [8], [11]..

It is important to notice on the other hand that design axial calculations presented in relations (1) and (2) do not stand for actual loads in weights (G_{tot}) and respectively (G_{tot}–W). These assume a preliminary analytical determination, to be done separately, depending on pre-load forces of each assembly. This step is achieved based on the well known theory based on force-deformation diagrams associated to each assembly 1-*n*. The calculated force system [2], resulted after the pre-load application, for the 14 $^3/_8$ inch stem size, are shown here in table 1

Table 1. The force system resulted after drill stem pre-loading, for the range of depth 100-650 m

Conditions / Axial loads	Drill conditions for depth cases, H_{max}, [m]						
	100	200	300	400	500	600	650
Axial exterior forces, by weights, F_e, [x10^3N]	1930	2290	2630	2970	3310	3650	3860
Total axial forces, after pre-load, F_{tot}, [x10^3N]	2316	2748	3156	3564	3972	4380	4632
The pre-load tension force, F_o, [x10^3N]	1477	1752	2012	2272	2532	2792	2953

Particularities of Verifying Calculation Applied to Threaded Joints

The verifying calculation for the pin and box threads approaches the following consecrate directions [1], [7], [8], [10], [11]:

- Thread shearing verification;
- Thread contact pressure verification;
- Thread bending verification.

Towards acknowledged formula used in usual verifying calculation, when tapered threads with asymmetrical profile have to be approach, there are some customizations to be done. For relevance, the calculus schemes are shown in figure 3.

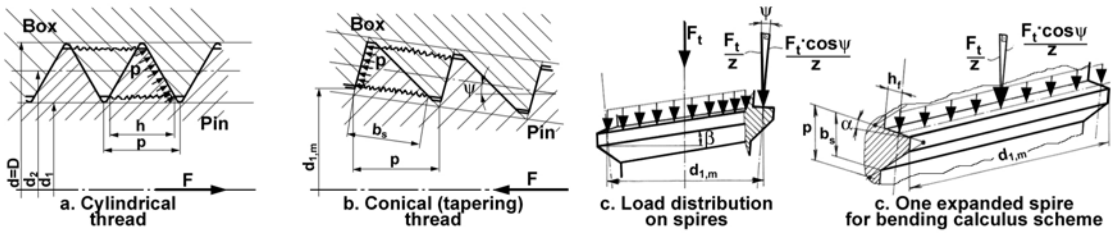

Fig. 3. The necessary calculus schemes for pin and box thread verifications

Based on fig. 3, *a* scheme, the strength shearing condition for cylindrical thread is:

$$\tau_{ef,sh} = \frac{F_{tot}}{A_{sh}} = \frac{F_{tot}}{k_s \cdot k_u \cdot \pi \cdot d_1 \cdot m} \leq \tau_{a,sh} \tag{3}$$

where:

- k_s – shape factor, concerns shape influence of the generator profile of the thread; this factor values are determinate for usual thread profiles;
- k_u – unevenness factor, concerns the unevenness of load distribution on the z number of screw spires included in thread length ($m=z\cdot p$); a possible determination of its value is provided by the empiric formula $k_u = 5p/d$;
- d_1 is the bottom diameter of the thread..

The special case of tapered threads, when generator profile is asymmetric (fig. 3., *b*), requires a separately determining for these influence factors.

Initially, the shape factor, k_s, means a geometrical determination of the bottom thread width-thread pich ratio. The dedicated value was separately determined in [2]; for this shape of special thread, it is:
k_s=0,756.

For the unevenness factor, k_u, the trend determination suggested above converges to a value field of k_u=0.2…0.3. Referring to multiple starts threads, these values reach over 0.6. On the other hand, the studies achieved in work [2] shows an obvious unevenness over the z spire of pin-box pair. Certain cases, the first 2-3 spires take the most of the load so, the unevenness factor could be taken, these cases:
k_u=0.18…0.25.

Finally, the special cases of multiple start threads, has to introduce another factor, considering the unevenness of stress distribution among separate screw starts. The technological indexing variation can often determine a severe imbalance of load distribution between i thread starts. The studies [2] suggest significant increases on the most loaded start, at 40…60%, that means to consider:
K_i=0.60…0.75.

Consequently, the strength shearing condition for taper special profile threads, in the most general case, can be the following form:

$$\tau_{ef,sh} = \frac{F}{A_{sh}} = \frac{F_{tot} \cdot \cos\psi}{k_s \cdot k_u \cdot k_i \cdot \pi \cdot d_1 \cdot z \cdot p} \le \tau_{a,sh} \tag{4}$$

An other important thread verification refers to contact pressure. The helical surface of both pin and box helical flanks are powerful compressed. The excessive load appears on the live flank but considering pin and box radial strains it extends over the passive flank too.

Taking over the strength contact condition, recommended for the cylindrical thread for number (*z*) of spire determination [11]:

$$z = \frac{4 \cdot F}{\pi \cdot k_u \cdot \left(d^2 - d_1^2\right) \cdot p_a} \tag{5}$$

The equation can be turn into an enlarged state, to use for special tapered threads:

$$p_{ef} = \frac{4 \cdot F_{tot} \cdot \cos\psi}{\pi \cdot k_u \cdot k_i \cdot \left(d^2 - d_1^2\right) \cdot z} \le p_a \tag{6}$$

Both (4) and (6) relation considers the diminishing load, caused of inclined direction of the cone generatrix towards the joint axis:

$$F_n = F_t \cdot \cos\psi \tag{7}$$

Most times, the bending strength verification for threads is negligible. This is because, in threads, the flexural stress is always much lower than shearing or contact stress. The calculus procedure of verification the bending tensile stress approximates the thread spire with a fixed beam (fig. 3., *c*, *d*) after imaginary rolled off. The bending moment is done by force component $F_t \cdot \cos\psi$, positioned on the live flank centre.

This situation, the bending strength condition – written for the tapered thread – can be formed such as:

$$\sigma_b = \frac{M_b}{W_z} = \frac{\dfrac{F_t \cdot \cos\psi}{z} \cdot h_f}{\dfrac{1}{6} \cdot \pi \cdot d_{1,m} \cdot b_s^2} = \frac{6 \cdot F_t \cdot \cos\psi \cdot h_f}{\pi \cdot z \cdot d_{1,m} \cdot (0.756 \cdot p)^2 \cdot k_u \cdot k_i} \le \sigma_{a,b} \tag{8}$$

In equation (8), moment arm h_f considers the thread clearance, c:

$$h_f = \frac{h}{2} + c \tag{9}$$

Conclusion

The here above review results in the following conclusions:

- The large diameter drilling required for shouldered thread joints the profile of an asymmetrical thread (with α=15°, γ=45° and $\alpha + \gamma$ =60°), with tapers generally smaller than at traditional drill strings.
- In case of threaded special pre-load joint assemblies of such is necessary when mounting the string; in the calculus of dimensioning of the pin and box (for Cases *I* and *II*) the total thrust load per each assembly (F_{tot}) is significantly fostered against loads determined out of the components' weights (G_{tot}) – by about 20%.

- The asymmetrical profile of the thread – with an angle of the live flank smaller than the passive one – contributes to a more convenient positioning of the loading versus the surface of this flank; this way, the contact strain on flanks is diminished and the actual pressure contact reaches smaller values.
- The asymmetrical shape of the thread leads to an increase in the wire stiffness thus determining a diminution of shearing tensions at basis of such and also leads to bending as a consequence of the same effect.
- Under the circumstances of making use of multiple start threads the calculation relations used in verifications of the thread include the effect of fostering of the tension level by considering the uneven distribution (k_e) in these i thread starts.

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Consideraţii privind particularităţile calculului filetelor asamblărilor cu umăr mufă-cep conice, cu profil generator asimetric

Rezumat

Aceste filete sunt utilizate în construcţia asamblărilor mufă-cep de la garniturile de foraj de diametre mari şi prezintă câteva particularităţi: direcţia de încărcare a spirelor este afectată de conicitate; forma asimetrică influenţează rigiditatea spirei şi nivelul tensiunilor; repartizarea sarcinii pe spire este accentuat neuniformă.
În plus, cazul particular al filetelor cu mai multe începuturi introduce un nou parametru: neuniformitatea repartiţiei sarcinii între începuturi.
Lucrarea prezintă principiile calculului filetelor la solicitările de contact, forfecare şi încovoiere, adaptat pentru asamblările conice cu umăr, de tip mufă-cep. Cazurile analizate se referă la filete de construcţie specială, care au profil generator asimetric.