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Some Considerations over Determining the Torque Bearing Capacity of a Conical Yielding Squeeze Ring Assembly

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Abstract

This paper presents some theoretical and experimental research over the conical yielding squeeze ring assembly. We studied the general algorithm for the load system determination – meaning axial and radial forces – and the calculus of the axial load reduction factor. Thus, it becomes possible to determine the torque carrying capacity and to verify it experimentally.

Key words: *assembly, yield squeeze, tapered rings, torque bearing capacity.*

General Considerations

The conical yielding squeeze ring assemblies are generally used for shaft-hub connections. Each conical ring pair has two components: a conical inner ring and a conical outer one, which have a direct contact on the conical area (fig.1) and a cylindrical contact area by the shaft and hub components (fig. 2).

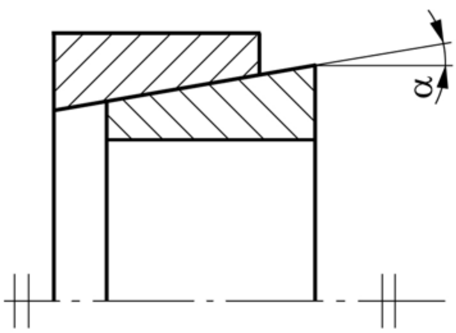

Fig. 1. Tapered ring pair

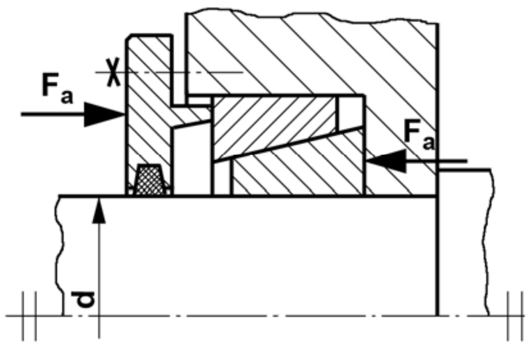

Fig. 2. Tapered yielding squeeze ring assembly with axial hub shoulder

In the squeeze fixing stage, rings get mutual contact on the conical surface. The tapered inner ring gets compressed, fixing the shaft, and the tapered outer ring gets stretched, fixing the hub side of the assembly.

The main advantages of tapered yielding squeeze ring assemblies are the following:

- High torque bearing capacity;
- Both shaft-hub components get excellent centring;
- Recurrent fitting and sliding off may not cause deterioration of the active surface;
- In overload conditions a temporary skidding happens, preventing any component deterioration;

Some of the main disadvantages are:

- The assembly build up requires additional coupling components;
- The low strain level of the tapered rings require high precision manufacturing (execution);
- In overload cycle conditions, skidding phenomena determines intensive heating.

The Load System and the Torque Bearing Capacity for the Tapered Ring Pair Assembly

The assembly fitting requires a mounting system using the axial displacement of the tapered ring pair. This leads to radial deformation, causing squeeze contact pressure on each shaft-inner ring and hub-outer ring surface.

The assemblies get torque bearing capacity caused by frictional load on tapered active surface of the ring pair and on double cylindrical areas.

The torque bearing capacity determination requires the squeeze radial forces, Q, depending on the axial pre-load force, F_a. The relation between the Q and F_a variables can be determined by imposing the equilibrium condition over the assembly axis, for each inner and outer tapered ring, considering the load system (fig. 3, *a*, *b* and *c*).

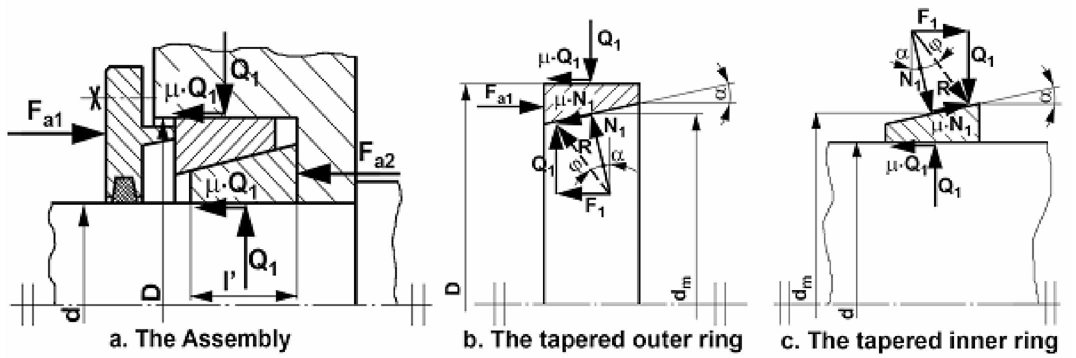

Fig. 3. Geometrical characteristics and the loading system of the assembly – *a*; loading system over the taper outer ring – *b*; loading system over the taper inner ring – *c*.

Considering the same friction coefficient for both contact surfaces ($\mu_1 = \mu_2 = \mu$), the equilibrium equation for the outer ring is:

$$F_{a1} - \mu \cdot Q_1 - F_1 = 0 \tag{1}$$

where:

$$F_1 = Q_1 \cdot \mathrm{tg}(\alpha + \varphi) \tag{2}$$

Using equation (1) and (2) follows:

$$F_{a1} = Q_1 [\mathrm{tg}\,\varphi + \mathrm{tg}(\alpha + \varphi)] \tag{1′}$$

In mentioned relation (1′) we can consider $\mu = \mathrm{tg}\varphi$ and equation becomes:

$$Q_1 = \frac{F_{a1}}{\mathrm{tg}\varphi + \mathrm{tg}(\alpha + \varphi)} \tag{1''}$$

Writing the same equilibrium condition of axial projections for the inner ring (fig. 3.,c):

$$F_{a2} + \mu \cdot Q_1 - F_1 = 0 \tag{3}$$

Considering both (3) and (1″) equations the result is:

$$\begin{aligned} F_{a2} &= F_1 - \mu \cdot Q_1 = Q_1 \cdot \mathrm{tg}(\alpha + \varphi) - \mu \cdot Q_1 = \\ &= Q_1 \left[\mathrm{tg}(\alpha + \varphi) - \mathrm{tg}\varphi\right] = \frac{\mathrm{tg}(\alpha + \varphi) - \mathrm{tg}\varphi}{\mathrm{tg}(\alpha + \varphi) + \mathrm{tg}\varphi} \cdot \mathrm{F}_{a1} \end{aligned} \tag{3'}$$

Using the specific notation:

$$k = \frac{\mathrm{tg}(\alpha + \varphi) - \mathrm{tg}\varphi}{\mathrm{tg}(\alpha + \varphi) + \mathrm{tg}\varphi} = \frac{1 - \dfrac{\mathrm{tg}\varphi}{\mathrm{tg}(\alpha + \varphi)}}{1 + \dfrac{\mathrm{tg}\varphi}{\mathrm{tg}(\alpha + \varphi)}} < 1 \tag{4}$$

which can be designated as "*reducing axial force factor*" *(RAFF)* – correlative for axial forces on frontal surfaces of both taper rings. Some of the usual values for RAFF are presented in diagrams shown in fig. 4. These were determined for different tapered α values, for some specific (usual) frictional cases – $\mu = \mathrm{tg}\varphi$.

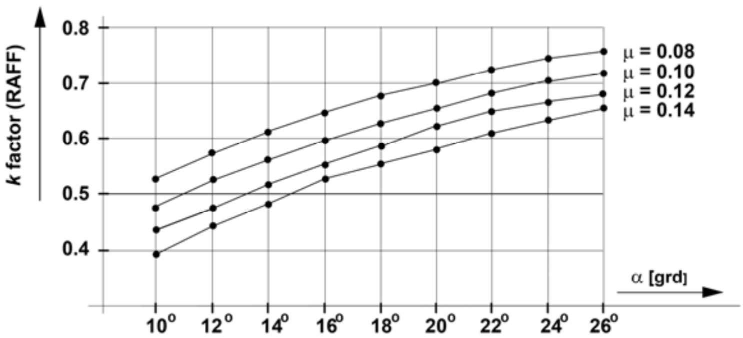

Fig. 4. RAFF (k factor) diagrams, determined for usual tapered (α) and frictional (μ) cases

Consequently, the RAFF (k factor) can diminish relation (3′) to the expression:

$$F_{a2} = k \cdot F_{a1} \tag{3''}$$

The passing over torque load of the tapered ring assembly gets the lower level at the shaft peripheral zone; at that place, a temporary skidding may happens. This frictional torque load level can be expressed by the relation:

$$M_{f1} = \mu_c \cdot Q_1 \cdot \frac{d}{2} = \mu_c \cdot \frac{F_{a1}}{\mathrm{tg}\varphi + \mathrm{tg}(\alpha + \varphi)} \cdot \frac{d}{2} \tag{5}$$

This torque level must be higher than exterior applied moment:

$$M_{f1} > M_t^c = \beta \cdot M_t \tag{6}$$

The β factor adds the significance of a safety coefficient for skidding situation. Equation (5) also includes the own friction coefficient for the cylindrical contact between shaft and inner ring, μ_c.

Experimental Equipment and Control Instrumentation

By the purpose to corroborate theoretical data by experimental results, The Machine Elements Laboratory was endowed by a dedicated test installation. This equipment was designed and manufactured for experimental tests of specific yielding squeeze assemblies; its structure is:

- The main conical yielding squeeze assembly, using a tapered ring pair;
- The stand for assembly torque testing;
- Control apparatus: dynamometer, dynamometric-hand key, vernier calliper.

The tapered yielding squeeze assembly (see fig. 5), is composed by the shaft (1), having a cylindrical extremity (squeeze zone) and the opposite one with external-hexagonal zone, for stand mounting. The two tapered rings (2, 3) have over side contact with a flanged hub (4) and a clamping bush (5), pressed by a single row tapered roller bearing (6). The axial pre-load system is composed by the axel cap (7), hexagonal bolt (8) and nut (9).

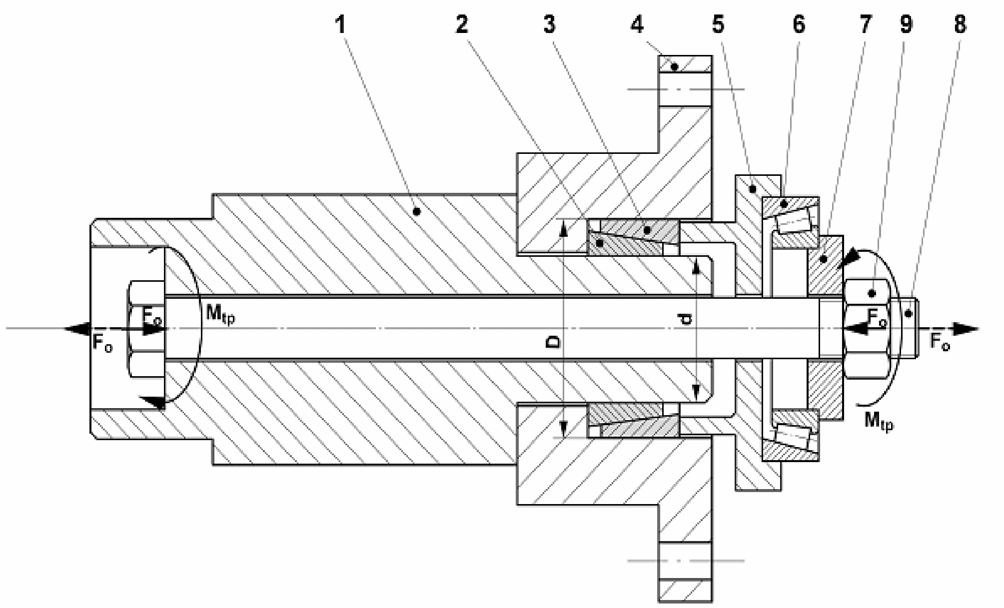

Fig. 5. Complete structure of the tapered yielding squeeze assembly, used for the experiment.

Friction factors for the active zones of the squeeze assembly – considered for fine turning – are recommended in the following range:

- For the bolt and nut threaded zone:
  $\mu = 0.10 \ldots 0.15$;
- For the shaft-inner ring contact:
  $\mu_c = 0.10 \ldots 0.15$.

The axial pre-load can be achieved by the nut screw, using the dynamometric-hand key. By consequence, it involves the load (force) system developing, shown in the fig. 4.

The Axial Pre-load Force and Torque Bearing Capacity Calculus of the Assembly

When the pre-load torque, M_{tp}, is applied on hexagonal nut by dynamometric key, the axial squeeze force (F_o) stretch the bolt core.
If neglecting the frictional moment of the row tapered roller bearing (a negligible quantity), the squeeze force (F_o) could be expressed as a torque moment (M_{tp}), by the formula:

$$F_o = \frac{M_{tp}}{\frac{d_2}{2} \cdot \mathrm{tg}(\alpha_m + \varphi')} = k_1 \cdot M_{tp} = F_{a1} \tag{7}$$

where d_2 is the thread medium diameter and φ' is the well-known partitioned angle of repose, which can be determined using equation:

$$\varphi' = \mathrm{arctg}\frac{\mu}{\cos\frac{\beta'}{2}} \tag{8}$$

In the (7) relation, k_1 factor has an obvious significance, as a geometrical constant. Similar to (7), the equation (1″), can be put in another concise form:

$$Q_1 = \frac{F_{a1}}{\mathrm{tg}\varphi + \mathrm{tg}(\alpha + \varphi)} = k_2 \cdot F_{a1} \tag{1''}$$

At least, the torque moment expression (5) becomes:

$$M_{tc} = M_{f1} = \mu_c \cdot Q_1 \cdot \frac{d}{2} = k_3 \cdot Q_1 \tag{9}$$

Torque Bearing Capacity Determination by Experiment

The torque bearing capacity of the tapered ring assembly can be also determined by the experimental procedure, using the stand for assembly torque testing (fig. 6). There are detailed the whole stand structure (fig. 6, *a*), the kinematical diagram of the torque loading mechanism (fig. 6, *b*) and a general view of the conical yielding squeeze assembly (fig. 6, *c*).

The experimental torque moment can easily be applied by using a motion screw linkage mechanism (fig. 6, *b*), which loads a lever lengthwise R = 200 mm. The specific force (F) developed by this screw mechanism is continually measured by the dynamometer. This makes expressible the experimental torque moment:

$$M_{tm(exp)} = F \cdot R \cdot \eta_r \tag{10}$$

where η_r is mechanical efficiency of the motion screw linkage mechanism.

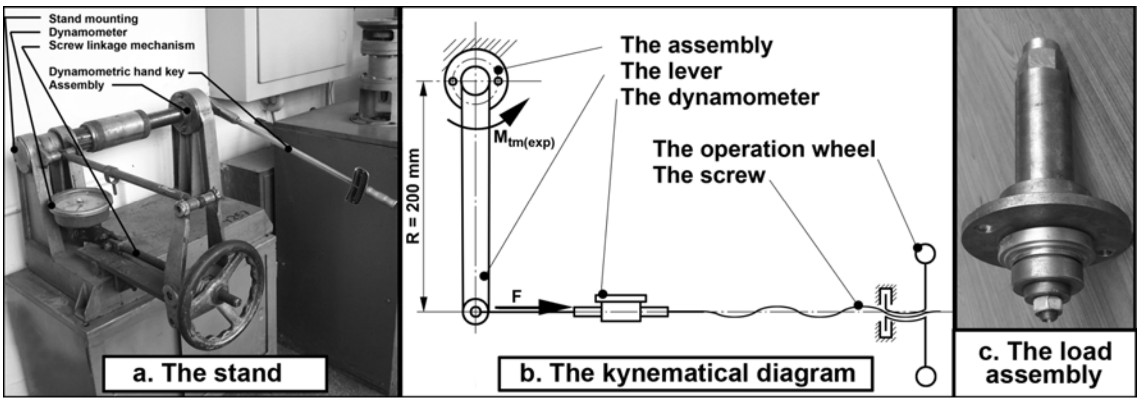

Fig. 6. The experimental stand structure; general view – *a*, kinematical diagram – *b*, pre-loaded assembly – *c*.

The torque bearing capacity determination of the conical yielding squeeze ring assembly requires the following work stages:

- The screw fitting of the nut, applying a torsion moment ($M_{tp(1)}$), whose value can be read at the dynamometrical key. According to this torsion moment value, we can determine the axial squeeze force (F_o) – relation (7), the normal reaction Q_1 – relation (1‴) and the bearing torsion moment of the assembly (M_{tc}) – relation (9).
- The fitted assembly (fig. 6, *c*) is mounted on the testing stand (fig. 6, *a*) and it follows be loaded by torque moment. The value of the experimental F force is read on the dynamometer scale disc (fig. 6, *b*), when the couple shaft-inner ring gets skidding; this way, the torque bearing moment – relation (10) – can easily be found.
- The trial of the fitted assembly may be repeated several times, for other higher values of the screwing moment of the bolt-nut connection: $M_{tp(2)}$, $M_{tp(3)}$, … $M_{tp(k)}$. Than, the acquired results must be specified in table 1 and graphically plotted.

Table 1. The main geometrical and mechanical parameters, calculated data and experimental results

No.	$M_{tp(i)}$	μ	μ_c	α_m	φ’	α	k_1	$F_o=F_{a1}$	k_2	Q_1	k_3	M_{tc}	R	F	$M_{tm(exp)}$
1	$M_{tp(1)}$							$F_{o(1)}$							
2	$M_{tp(2)}$							$F_{o(2)}$							
3	$M_{tp(3)}$							$F_{o(3)}$							

Conclusions

This work presents the determination of the torque bearing capacity for a tapered yielding squeeze pair ring assembly.

The possible incongruence between the calculated bearing moment (M_{tc}) and the experimental determined moment ($M_{tm(exp)}$) of the assembly can be explained through the initial theoretic appreciation of the frictional coefficient (or its possible working modifications) and also by the reading errors at the dynamometrical hand-key, stand dynamometer etc.

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Unele consideraţii asupra determinării capacităţii portante a asamblărilor prin strângere elastică cu inele tronconice

Rezumat

Lucrarea prezintă unele cercetări privind asamblarea cu inele tronconice. Se prezintă componentele sistemului de forţe dezvoltate în asamblare, relaţiile dintre acestea şi se determină capacitatea de transmitere a momentelor de torsiune.