CALCULATION MODEL FOR PRE-STRESSED BOLTED JOINTS OF SLEWING BEARINGS

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Abstract: The presented paper describes a calculation model for pre-stressed bolted joints used to connect slewing bearings to the adjacent structures. A classical pre-stressed bolted joint calculation model is described and its insufficiencies for direct application on slewing bearings are pointed out. Finally, the suggested stress analysis is done by modelling the most loaded single bolt segment of the slewing bearing assembly, applying the preload and the working load on it. The stress determination in the bolt is done by finite element analysis.

Keywords: slewing bearing, bolted joint, calculation model, stress analysis

1 INTRODUCTION

Slewing bearings are bearings of large dimensions and they are used in different engineering applications (e.g. wind turbines, mobile cranes, communication systems, turning tables etc.). Their function is to connect two adjacent structures, allowing rotation and transmission of load between them. In contrast to the majority of other bearings, slewing bearings are mainly used for slow rotational speeds, often for intermittent and oscillatory movement. They consist from an inner and outer ring and at least one raceway with belonging rolling elements (Fig. 1.). With regard to application different construction variants are known and used. Thus the slewing bearings can be provided with or without inner or outer gearing and with rolling elements of different shape (ball, roller).

The connection between the individual ring of the slewing bearing and the rest of the structure is established by pre-stressed bolted joints (Fig. 1.). The presented paper presents and describes a calculation model for these pre-stressed bolted joints by combining conventional calculation methods and FEM analysis.

The installed and operating slewing bearings can be exposed to different external loads. For some applications it is a quite difficult task to exactly determine the loads over the entire lifetime. The loadings on slewing bearings installed in wind turbines are one of those examples. As a result of numerous studies it is possible to determine the load spectra which simulate the loading of the bearing over its entire lifetime [1]. However, the working load on the slewing bearing (and also on the bolted joints) is for the needs of calculation usually given in a simpler manner by defining the axial force \( F_{ax} \), the radial force \( F_r \) and the overturning moment \( M_T \) (Fig. 1.). The aim of this paper is to determine the axial stress distribution in the bolt during operation on the basis of an actual slewing bearing geometry and working load.
2 PRE-STRESSED BOLTED JOINT CALCULATION

A bolted joint in general is a detachable connection between at least two different parts, an element with external thread (bolt) and an element with internal thread (washer) (Fig. 2.a). When there is an axial tension force \( F_M \) in the bolt stud before the operation of the working load \( F_A \) the bolted joint is called pre-stressed. Pre-stressed and dynamically loaded bolted connections are frequently used in different constructions and they are often representing the most critical part. Despite of the fact that there is no universally accepted calculation model for bolted joints the most often used recommendation in industrial calculations is the VDI 2230 [2, 3, 4].

When the working load \( F_A \) acts on the pre-stressed bolted joint in traction manner the tension force in the bolt \( F_S \) increases in comparison to assembly preload \( F_M \), on the other hand when the working load acts in compression direction the tension force in the pre-stressed bolt decreases (Fig. 2.b). The dependence between the working load \( F_A \) and the bolt force \( F_S \) is usually presumed to be linear [3].

**Fig. 1.** External loads acting on the slewing bearing

**Fig. 2.** Concentric loading of a single bolted joint (a) and \( F_A - F_S \) diagram (b)
In this cases the additional axial bolt load \((F_{SA})\) can be calculated as:

\[
F_{SA} = \Phi \cdot F_A = n \cdot \frac{\delta_p}{\delta_S + \delta_p} \cdot F_A
\]  

(1)

where \(\Phi\) stands for the relative resilience factor and \(n\) for the load introduction factor. The whole bolt load \((F_S)\) consists of the assembly preload \((F_M)\) and the additional axial bolt load \((F_{SA})\):

\[
F_S = F_M + F_{SA}
\]

(2)

It is evident from Eq. (1) that for a quantitative analysis of pre-stressed bolted joint it is necessary to manage both the bolt \((\delta_S)\) and member resilience \((\delta_P)\). The bolt resilience can be easily determined with analytical methods [3]. On the basis of recent researches, which used numerical analyses for verification, an even more detailed bolt resilience calculation was presented. This approach takes into account various geometrical parameters of the bolt (head, thread, nut etc.), its material and friction properties [5]. On the other hand the determination of member resilience is a more complex task. There are different models for describing the effective volume of the member that is subjected to compressive stress and as a consequence the member resilience. Those models were mainly confirmed with finite element analyses and experiments [6].

The direct use of those calculation models is however rather difficult for pre-stressed bolted joints of slewing bearings. The reasons for that are the following:

- **Analytical models [3,4,5]** are mainly valid for single bolted joints (cylindrical or prismatic bodies) with concentric clamping and loading. Due to calculation restrictions only small eccentricity is allowed. This means that the vertical axis of the working load \((F_A)\) must be relatively close to the axis of the bolt. On the contrary, the cross section of the slewing bearing ring has a complex geometry where the eccentricity of the clamping and loading is quite considerable.

- **The load introduction factor \(n\) (Eq. (1)) is decisive for the additional bolt load determination.** For some basic geometrical configurations and loading conditions load introduction factors are available as the results of extensive parametrical studies [3]. Yet for many cases, also for the bolted joints of the slewing bearings, there are no reliable load introduction factors available. As a consequence this can also be a significant source of uncertainty in bolt load determination.

- **Some authors already pointed out that as a result of eccentricity partial opening of the bolted joint is often present during the operation and loading of the slewing bearings [1,7,8].** For dealing with partial opening of the pre-stressed bolted joints only some approximate solutions exist [3]. Partial opening of the joint is also one of the reasons for non-linear dependence between the working load and the bolt loading (Fig. 3.b).

- **The existing calculation models are mainly developed for single bolted joints. Generally they do not give assistance for working load \((F_A)\) determination.** In the case of slewing bearings, especially when there is an overturning moment, every bolt can be differently loaded. Therefore the working load determination requires additional attention.
As a possible method for dealing with the pre-stressed bolted joint of the slewing bearings some authors presented a numerical solution in which a particular finite element was developed. This complex element is built in a way that it behaves like a ring except in the axial direction. This was achieved with careful combination of several simple finite elements which allows simulating contact, bending etc. By fine tuning of this element it is possible to take into account partial opening, slight eccentricity of loading and more realistic load factor. Numerous finite element simulations were performed for results confirmation [8].

3 CALCULATION MODEL

The logical consequence of inability of direct use of classical pre-stressed bolted joint calculation methods is the application of finite element method. In general, there are two basic ways for analyzing ring flange connections with multiple bolts (e.g. slewing bearing). One method requires the modeling of the whole ring (Fig. 4.a), the second takes into consideration an equivalent sector of the most loaded bolt (Fig. 4.b). Both methods have their pros and contras.

Fig. 3. Eccentric loading of eccentrically clamped bolted joint (a) and the belonging non-linear $F_A - F_S$ diagram (b)

Fig. 4. Model of the whole slewing bearing and flange assembly (a) and the equivalent segment of one bolt (b)
3.1 Determination of working load \((F_A)\)

Modeling and analyzing the whole bearing ring with the belonging substructure enables a more accurate determination of bolt operating forces at critical joints by taking into account the joined structure elasticity in the direction of the bolt axis \([9,10]\). This is particularly important when the substructure is complex and it cannot be considered as ideally rigid or its rigidity is variable around the circumference of the ring. Analyses where the stochastic ring geometry imperfections are taken into consideration also demand the modeling of the whole ring \([11]\).

On the other hand, when the bolts are equally arranged around the ring, the support is rigid or tubular and no stiff points are present or they are eliminated with the help of so called “mounting tubes” \([2,12]\), the modeling of the most loaded sector can be used. This method usually overestimates the loading in the maximum loaded segment since it cannot cover any stress redistributions when the partial opening occurs \([1,2,7,12,13]\). In these cases the following equation can be used for determination of the working load \(F_A\) on the analyzed (most loaded) segment \([1,2]\):

\[
F_A = \frac{1}{\cos \beta} \left( \frac{2 \cdot M_T}{\pi \cdot R} \cdot \sin \frac{\pi}{z} + \frac{F_{ax}}{z} \right) \quad (3)
\]

where \(\beta\) is the angle between the working load \((F_A)\) and the axis of the bearing (Fig. 6), \(z\) denotes the number of equally distributed bolts around the circumference and \(R\) is the radius of the analysed bolt (Fig. 1.).

3.2 Finite element model of the ring segment

Due to geometrical symmetry only the half segment of the most loaded bolted joint was modelled. For this purpose the ABAQUS/CAE finite element software \([14]\) was used. The model of the analyzed segment consists of the slewing bearing ring (inner or outer), the belonging support (flange) and the bolt (Fig. 5.).

There are different approaches for modeling the bolt and pre-stressed bolted joint connections in FE analyzes. In dependence of the desired results and their accuracy the bolt can be modeled as one dimensional beam element, a solid body or the combination of both \([15]\). When the bolt is modeled as a solid body different studies showed that acceptable results can be attained even without detailed modeling of the bolt thread \([13,16]\). Because of that and according to the other similar analyses \([2,12,13]\) in presented model the bolt was modeled as a solid body consisting of three cylinders (head, bolt stud and nut).

The used FE software already has a built-in tool for simulating the bolt preload (“Bolt load”). Therefore the preloading and the subsequent external loading of the joint can be easily and realistically simulated. Working load was introduced as a uniform pressure on the raceway of the ring under \(\beta = 45^\circ\) \([7]\). For more precise simulation, contacts with normal and tangential (coefficient of traction 0.3) behavior were defined between (Fig. 5.a):

- bolt head and the ring
- ring and the flange (to allow separation under loading)
- nut and the flange

In the realized FE analyses a linear-elastic material model was applied \((E = 210\ \text{GPa} \text{ and } \nu = 0.3)\). For the model meshing 8-node linear brick elements with reduced integration were used (Fig. 5.c). The approximate size of the used elements was 1 mm.
4 PRACTICAL EXAMPLE

The proposed computational model has been used to determine the load capacity of a real bolted joint. In this case both the outer and inner ring of an existent single-row four point contact bearing was analyzed. The geometrical properties of the slewing bearing used for the FE model were taken from a slewing bearing manufacturer’s catalogue [17] and they are shown in Tab. 1. Because of 24 bolts a 7.5° half segment of the assembly was modeled. The defined bolt size was M16, the required strength grade 10.9 (yield strength of the bolt’s material is 900 MPa).

During analysis the studied slewing bearing was subjected to overturning moment $M_T = 250$ kNm and to compressive axial force $F_{ax} = 300$ kN. Those values were chosen on the basis of the critical loading curve for the bolts shown for this particular bearing in the manufacturer’s catalogue [17]. With regard to the Eq. (3) this external loading translates to a working load of approx. $F_A = 58$ kN at the most loaded segment of one bolt. The analyzed slewing bearing has a geared outer ring but as a simplification this was not considered in the FE simulation.

<table>
<thead>
<tr>
<th>Outer ring / Inner ring</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Ring height [mm]</td>
<td>54 mm</td>
</tr>
<tr>
<td>Ring width [mm]</td>
<td>45 mm / 46.5 mm</td>
</tr>
<tr>
<td>Ball track diameter [mm]</td>
<td>764 mm</td>
</tr>
<tr>
<td>Bolt circle diameter [mm]</td>
<td>823 mm / 706 mm</td>
</tr>
<tr>
<td>Number of bolts</td>
<td>24</td>
</tr>
<tr>
<td>Bolt holes diameter [mm]</td>
<td>17.5 mm</td>
</tr>
<tr>
<td>Overall height of the bearing [mm]</td>
<td>63 mm</td>
</tr>
</tbody>
</table>

Tab. 1. Main geometrical parameters of the slewing bearing used in the FE model
During analysis axial stresses in bolts were observed. This was done in three steps:
- pretension of the bolt (Fig. 6.a)
- tensile loading of the bolted joint (Fig. 6.b)
- compressive loading of the bolted joint (Fig. 6.c)

Other authors [2] already pointed out in their works that higher preloading has a positive effect on bolted joints because it reduces alternate stresses in the bolt, reduces the risk of loosening of the joint and lag the beginning of slipping at transverse loading. To confirm that two different magnitudes of pretension (σM) were simulated in this practical example: preloading to σM = 0.7·Rp0,2 (yield strength) of the bolt’s material (630 MPa) and preloading to σM = 0.9·Rp0,2 of the bolt’s material (810 MPa).

5 COMPUTATIONAL RESULTS

The numerical results of the bolted joint (outer and inner ring) are presented as axial stress distributions along the bolt (Figs. 8-16.) The stress distribution was monitored on both sides of the bolt along the whole height of the bolted joint with exception of the beginning and end of the bolt stud, where the rapid changes of bolt geometry take place (Fig. 7).

From the gained stress distributions both the highest axial working stress σs as also the highest alternating stress σa can be determined. The highest working stress σs on both sides of the bolt can be obtained directly from the presented results (Figs. 8-16.) by locating the maximum value of axial stress, taking into account both sides of the bolt (A-side & B-side, Fig. 7.). The highest working stress σs in the bolt should be lower than the yield strength of the given bolt material. In this case, the bolt strength grade is 10.9, that means Rp0,2 = 900 MPa.

The magnitude of the alternating stress σa influences the fatigue life of the bolt and it must be lower than the fatigue limit of the bolt σAS to reach whole fatigue life.
(number of alternating cycles greater than \(2 \cdot 10^6\)). The alternating stress \((\sigma_a)\) can be determined as [3]:

\[
\sigma_a = \frac{\sigma_{S_{\text{max}}} - \sigma_{S_{\text{min}}}}{2}
\]

(4)

In this equation \(\sigma_{S_{\text{max}}}\) stands for the highest axial working stress while \(\sigma_{S_{\text{min}}}\) represents the lowest axial working stress in the bolt. Both the \(\sigma_{S_{\text{max}}}\) and the \(\sigma_{S_{\text{min}}}\) are observed in each node of the bolt model separately. As a result, local alternating stresses are obtained in every node along the bolt axis (on both sides). The highest value of the alternating stress on whichever side of the given bolt is considered deciding.

According to VDI 2230 [3] the fatigue limit of the high-strength bolts rolled before heat treatment \((\sigma_{ASV})\) can be calculated as:

\[
\sigma_{ASV} = 0.85 \left( \frac{150}{d} + 45 \right)
\]

(5)

This means that the fatigue limit \((\sigma_{AS})\) of the used bolt size (M16) is 46.2 MPa. The alternating stresses in the bolt should be under this limit.

Fig. 7. Definition of the bolt sides for stress distribution monitoring: inner (a) and outer ring (b)

The highest axial working stresses \((\sigma_S)\) in the bolt of the outer ring are 732 MPa \((\sigma_M = 0.7 \cdot R_{p0.2})\) and 854 MPa \((\sigma_M = 0.9 \cdot R_{p0.2})\). For the inner ring the highest axial working stresses in the bolt are 694 MPa \((\sigma_M = 0.7 \cdot R_{p0.2})\) and 852 MPa \((\sigma_M = 0.9 \cdot R_{p0.2})\). According to these results, the axial working stresses \((\sigma_S)\) are in all cases under the admissible value of 900 MPa on the monitored path along the bolt.

Meantime, the maximum alternating stress \((\sigma_a)\) during loading (pretensioning to \(0.7 \cdot R_{p0.2}\)) is 56.3 MPa in the bolts of the outer ring and 30.4 MPa in the bolts of the inner ring. By pretensioning to \(0.9 \cdot R_{p0.2}\) alternating stress \((\sigma_a)\) in the bolts of both rings falls, in outer ring to 23.8 MPa, in inner ring to 17.4 MPa. The alternating stress control confirmed the assumption that higher pretension positively affects on the magnitude of the alternating stress. With regard to the results of this analysis, when the
bolts are pretensioned to $0.7 \cdot R_{p0.2}$, the alternating stress ($\sigma_a$) in the bolt of the outer ring is higher that the fatigue limit of the M16 bolt $\sigma_{ASV}$ (46.2 MPa).

### 5.1 Outer ring

**Fig. 8.** Stress distribution along the A-side of the bolt ($\sigma_m = 0.7 \cdot R_{p0.2}$); max. $\sigma_a = 56.3$ MPa

**Fig. 9.** Stress distribution along the B-side of the bolt ($\sigma_m = 0.7 \cdot R_{p0.2}$); max. $\sigma_a = 19.6$ MPa

**Fig. 10.** Stress distribution along the A-side of the bolt ($\sigma_m = 0.9 \cdot R_{p0.2}$); max. $\sigma_a = 23.8$ MPa
5.2 Inner ring

Fig. 11. Stress distribution along the B-side of the bolt ($\sigma_M = 0.9 \cdot R_{p0,2}$); max. $\sigma_a = 14.3$ MPa

Fig. 12. Stress distribution along the A-side of the bolt ($\sigma_M = 0.7 \cdot R_{p0,2}$); max. $\sigma_a = 7.6$ MPa

Fig. 13. Stress distribution along the B-side of the bolt ($\sigma_M = 0.7 \cdot R_{p0,2}$); max. $\sigma_a = 30.4$ MPa
6 CONCLUSION

A calculation model for pre-stressed bolted joints of slewing bearings is presented. Because of the specific clamping and loading conditions of the slewing bearing rings it is difficult to accurately verify the stress conditions in the connecting bolts with the help of the usual pre-stressed bolted joints calculation methods. The presented calculation model uses a finite element analysis to obtain the axial stress distribution along the bolt axis during loading. From these results the axial working stress $\sigma_S$ and the alternating stress $\sigma_a$ can be determined. Both stresses serve as a basis for strength verification of the used pre-stressed bolted joints.

In the further researches more attention should be paid to the determination of the working load ($F_A$) as the main goal is to determine the dependence between working load ($F_A$) and the belonging bolt load ($F_S$). As many authors already pointed out, in some cases it is a oversimplification to consider the supporting structure as ideally rigid. Beside that an influence of more realistic material, contact and geometrical
properties of the bolt should be investigated. This would probably significantly improve the applicability of results.

References: