Global Structural Behaviour of Ring Flange Joints

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Summary:

This report deals with FE-modelling of ring flange joints used for connections in towers of wind energy converters. The behaviour of these joints is often analysed by reducing the total ring-flange to a segment with only one bolt. For these segments analytical approximations exist in the literature to predict the relation between the force acting on the segment and the corresponding force in the bolt.

The advantages of modelling the total ring-flange-joint are demonstrated by worked examples. As a basis for the discretisation of the flange, the properties of the high-tensile bolts used for this kind of connections in Germany are pointed out and the influences on the modelling are discussed.

For reduction of the computing time the substructure-modelling technique is used. The advantages and restrictions of this technique in the worked examples are discussed.

Keywords:

Ring flange joint, high-tensile bolts, ductility, substructure modelling

1 Introduction

Towers of wind energy converters (WEC) are commonly constructed as steel tubes. At present the height of these towers is about 80-100 m. For a rapid installation the tubes are prefabricated in members of 25 to 30 m length which will be fitted to the tower using bolt connections. For these connections ring flange joints with pre-stressed high-tensile bolts are used.

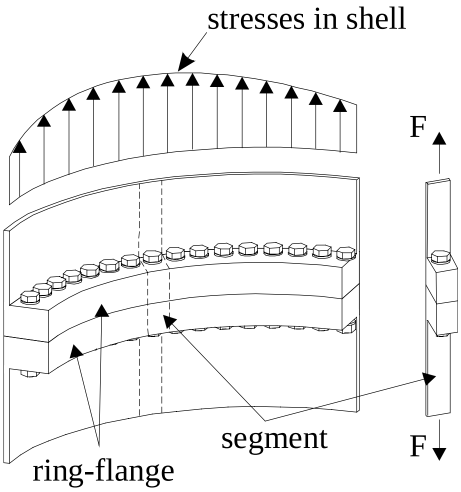


Fig. 1: Part of a ring-flange-joint and appropriate segment

The behaviour of these joints is generally analysed by reducing the whole ring flange to a segment with one bolt as shown in Fig. 1. The stresses in the shell can be transferred into a force acting on the segment. For these segments analytical approximations exist in the literature to predict the relation between the force acting on the segment and the correlating force in the bolt. The plastic load carrying capacity of the segment can be determined with mechanisms which represent different states of failure.
In the global ring flange joint the stresses can be reallocated because of plastic strains in the segments with maximum loading. This results in reduced stresses in the controlling segments. A more economical design of ring flange joints can be obtained by taking the overall joint into account instead of the segment. To define the failure of the joint within a FE-analysis the specific behaviour of the high-tensile bolts has to be taken into account especially the small ductility which is a result of the significant dimensions of the pre-stressed high-tensile bolts provided in the German standard for steel-constructions DIN 18800 called HV-bolts [2].

2 FE-Modelling of Bolts

The bolts have a significant influence on the carrying behaviour of the ring flange joint. Because of this the substantial properties of the bolts should be represented in the FE-model. These are on the one hand the elastic stiffness and on the other hand the behaviour when plastic strains occur. As mentioned above the specific dimensions of the HV-bolts result in a very small plastic elongation of the bolts which can determine the failure of the investigated joint. The presented FE-modelling of the bolts is founded on conclusions concerning this topic which are compiled by the authors in [5]. In the following chapters important topics for the bolt-modelling are pointed out.
The generation of the FE-Model has to be adjusted to the required results. The aim is to obtain the results of a certain exactness with acceptable computing time. For example, if only the elongation of a bolt has to be determined it is not necessary to include the thread of the bolt and the nut in the FE-model. But then it is not possible to evaluate the stress distribution within the thread.

2.1 Classification of FE-Bolt-Models

To define these differences in FE-modelling the following classification has been suggested by the authors in [5]:

Class 1: The bolt is modelled including the geometry of the thread of bolt and nut. To obtain realistic values of the stresses at the core of the thread a detailed discretisation of the FE-net is required (see Fig. 2). The amount of computing time can be immense, especially if the bolt is modelled three-dimensional. The model illustrated in Fig. 2 with axial symmetry comprises 18,000 degrees of freedom.

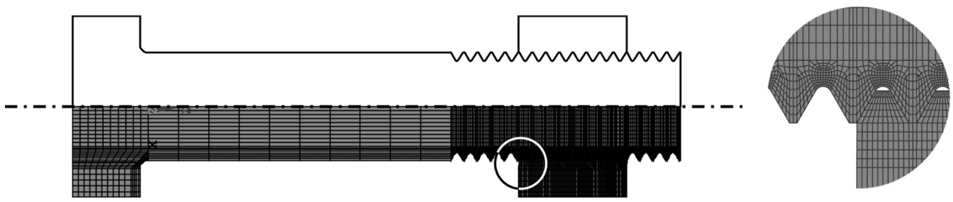


Fig. 2: 2D-model of a class 1-bolt and detail

Class 2: The bolt is assembled from cylindrical sections without concerning the geometry of the thread (see Fig. 3). Therefor the stresses in the thread cannot be determined, but the dimensions of the single cylindrical sections can be defined in that way, that the behaviour of the deformations in the elastic and in the plastic range of the loading can be obtained correctly. In comparison to the above described class 1-model the number of required degrees of freedom can be reduced to 10-15%. Compared to a three-dimensional class 1-model the reduction would be even greater.

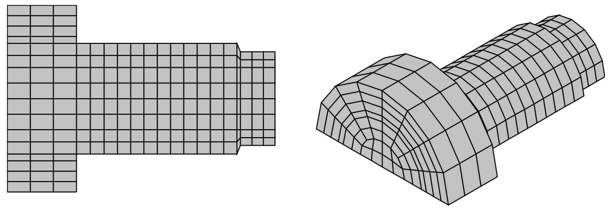


Fig. 3: 3D-model of a class 2-bolt

Class 3: In this class the bolt is described as a nonlinear spring. To determine the characteristics of the spring preliminary investigations with class 1 or class 2- models or otherwise test results are necessary.

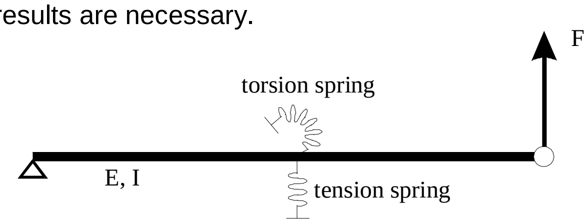


Fig. 4: Bolt as a spring (Class 3)

As described by the authors and explained with an example in [5] an appropriate carrying behaviour of bolts can be reached with class 2-bolt models in the range of elastic and plastic deformations. As an additional failure criterion a maximum elongation of the bolt has to be defined.

2.2 Definition of Bolt Failure

Despite the simplified discretisation of the bolt with a class 2-model the load-deformation-behaviour can be obtained correctly for the overall bolt. But the failure of the bolt is determined by the thread that is not modelled. Therefore the failure of the thread can not be described by the model and an additional criterion has to be defined. Significant for the failure of HV-bolts is, as Steurer has shown in [4], that this type of bolts has as a consequence of the special dimensions a very short elongation of the bolt for ultimate loading. This can mainly be traced back by the fact that these bolts only have short lengths of the free loaded thread (the part of the thread which is loaded but lies not within the nut) in which plastic deformation occurs.

Steurer gives in [6] a bilinear relation to determine the load-deformation-behaviour of high-tensile bolts which is based upon tests of bolts with diameters between M12 to M27. The ultimate elongation of a high-tensile bolt is described by the following formular :

$$\delta_{bolt,u} = \frac{F_u}{E} \cdot \left(\frac{0,4 \cdot d + \ell_{shaft}}{A_{shaft}} + \frac{\ell_{shaft-thread-transition}}{0,5 \cdot \left(A_{shaft} + A_{kern} \right)} \right) + \frac{F_y}{E} \cdot \frac{\ell_{thread} + 0,6 \cdot \ell_{nut}}{A_{kern}} + \frac{F_u - F_y}{\alpha \cdot E} \cdot \frac{\ell_{thread} + 0,6 \cdot \ell_{nut}}{A_{kern}}$$

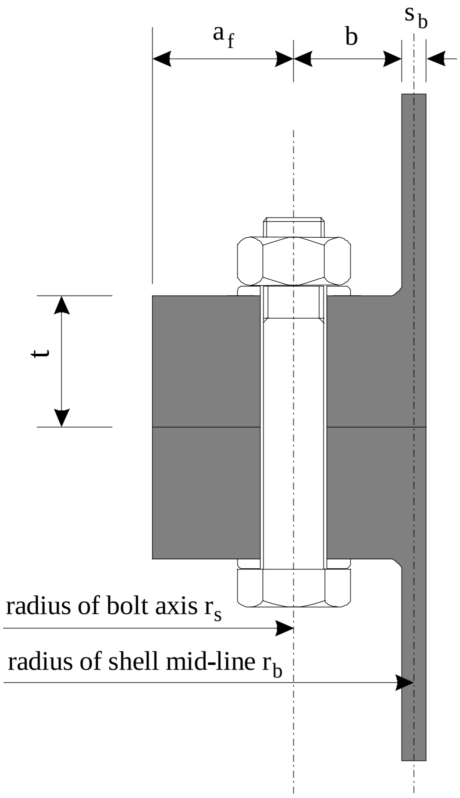
With the modelling of the bolt described above it can be guaranteed that the model reaches the ultimate bolt force which is determined in the standard exactly at this elongation. For a HV-bolt M30x175 (as used in flange 2, see Tab. 1) the maximum elongation has to be limited to 1,5 mm.

Beside the HV-bolts Steurer analysed bolts with identical properties but with continuos threads. At these bolts, so called HVN-bolts as mentioned by Steurer, plastic deformations occur at the total loaded bolt length. This results in a significant higher plastic elongation. If in flange 2 HVN-bolts M30x175 instead of the HV-bolts are used the maximum elongation has to be limited to 5,95 mm and exceeds the HV-bolt elongation about four times.

2.3 Class 2 Bolt Model in this Investigation

The class 2 bolt model is assembled for cylindrical sections. (Fig. 3). The model contains three-dimensional 8-node-elements which have three degrees of freedom per node (SOLID 45). Because of the symmetry of the ring flange only one half of the flange is considered. Therefore the flexibility of the bolt has to be reduced to the half of the bolt. The way how a bolt model can be adjusted to a realistic elastic and plastic behaviour is described in detail in [5]. Here only the crucial points are presented:

- Elastic flexibility in axial direction
  The different sections of the bolt which have to be considered to calculate the elastic flexibility (see VDI 2230 [3]) can be distinguished in parts with a direct counterpart in the model (these are for HV-bolts according to DIN 6914 [1] shaft, head and free loaded thread) and parts for which special consideration have to be made (core of the thread within nut and nut) and which should not be neglected. The consideration can be obtained by adjusting the lengths of cylindrical sections.
- Plastic deformations in axial direction
  Investigations of class 1 models of HV-bolts have shown that mostly the first and the last notch of the free loaded thread determine the plastic strain distribution in the thread. These significant notches can be modelled in class 2-bolts by using short lengths for the transition between shaft and thread (see Fig. 3). In a first step for the thread a section with tensile stress area and length of the free loaded thread should be modelled. By adjusting the length of the shaft the elastic flexibility of the bolt can be obtained by using the analytic relations of VDI 2230.
- Combined loading
  If significant parts of bending moments are existing in the bolts, the bending flexibility should be taken into account to receive realistic overall stresses. With the analytic relations of VDI 2230 the elastic flexibility for axial and bending loading can be kept in one class 2-model. This is important for the calculation of fatigue stresses in the bolts. But it is in most cases not possible to reach additionally the correct plastic behaviour. If the plastic behaviour is important for the investigation, a mistake in the bending flexibility can be accepted. This will have less influence on the global structural behaviour of the flange than a wrong elastic axial flexibility. Additionally plastic deformations will reduce the bending moments in the bolt in the range of the ultimate loading.



Ring Flange		**1**	**2**
a_f	[mm]	80	70
b	[mm]	57	53,5
t	[mm]	80	65
s_b	[mm]	18	12
r_s	[mm]	1380,5	1229
r_b	[mm]	1446,5	1288,5
Number of bolts	[-]	96	66
Bolt		M36x220	M30x175
Material of bolts		FK 10.9	FK 10.9
Material of flange		S 355	S 355
Material of plate		S 355	S 235

Tab. 1: Dimensions of the ring flanges

3 Analysis of Ring Flanges

The use of a single segment as shown in Fig. 1 to represent the whole flange leads to an overestimation of the loading in the maximum loaded segments because any rearrangements of the stresses can not be covered by the numerical model. If the bolt with the highest loads reaches yielding point, the stiffness reduces. For further increasing external loads the additional stresses will be taken over by the adjacent bolts. The dimensions of the investigated flanges are assembled in Tab. 1 with their significant parameters.

3.1 FE-Modelling of Ring Flanges

Bending moments and axial forces lead to symmetric stresses in ring flanges. Therefore the system can be reduced to the half. To represent the loading situation even a quarter of the joint would be sufficient. But this would prevent the rearrangement of the stresses to the side of the compression stresses like it is significant assumed by Ebert in [7].
To permit this rearrangements in the half joint the shell has to be modelled with a sufficient length. In these calculations a length of about two times the diameter of the ring flange is used. For the discretisation of the flange and the bolt a compromise has to be found. On the one hand a very fine discretisation would prevent an economic solution of the investigations because of enormous computing time on the other hand could a rough FE-net leads to stress distribution which do not represent the reality good enough. For the overall carrying behaviour single stress peaks are less important. Therefore a smaller number of elements seems to be acceptable for the calculations presented here. It is much more important to consider the parts of the joint which have a great influence on the overall behaviour. For the rearrangement of the stresses the bolts are determining. Therefore the bolts are modelled with the method described above. In comparison with finer FE-meshs it could be shown, that the discretisation shown in Fig. 5 can represent the overall behaviour of the flange.

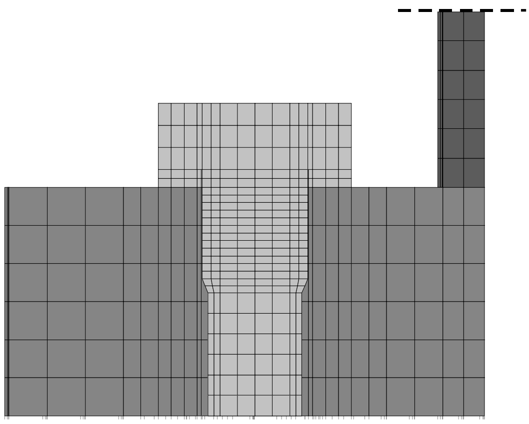


Fig. 5: FE-mesh for calculations

3.1.1 Substructure Modelling

If ring flange joints with such a high number of bolts as in the examples discussed here are analysed by using the FE-method a large amount of computing time has to be assumed. The software Package ANSYS provides the possibility to substitute parts of the FE-model (substructures) by so called “superelements”. Applying this method the stiffness matrix of the substructure is calculated in a first step. In the next step the overall model has to be built up by inserting the superelements at the appropriate places. The pre-calculated stiffness matrix of the superelement will be filled in the overall stiffness matrix. An advantage of this method is that the required working memory of the used computer can be significantly smaller. If the stiffness matrix has to be re-calculated during the calculation process as it is typical for calculations with plastic deformations the computing time can be reduced because the stiffness matrices of the superelements will not change. This shows on the other hand the restrictions of this technique: if the stiffness matrices of the superelements will not change during the calculation process in this areas nonlinear effects can not be considered. Therefore only parts of the joint can be substituted by superelements in which plastic strains do not have to be assumed. For ring flange joints this will be the case on the compression side of the bending moment. On this side the loads can be distributed on a significant part of the bottom side of the flange. On the tension side the loads are collected in the bolts and on the inner edge of the flange for ultimate loading.

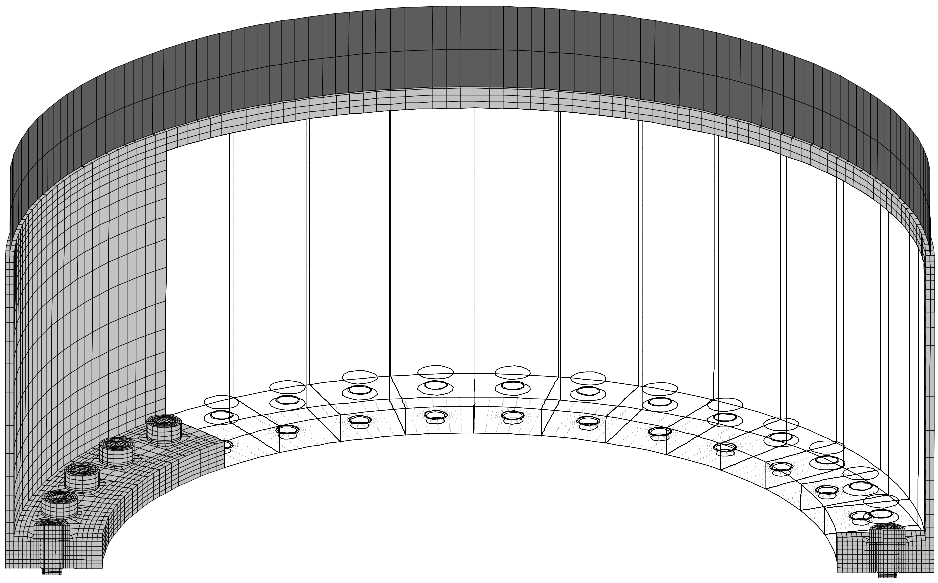


Fig. 6: Example for a FE-model of a ring flange joint with superelements

3.1.2 Check of the Linear Range

To estimate the area of the ring flange in which superelements can be applied, the calculation of a segment of flange 2 with material nonlinearities is compared with a calculation with linear-elastic behaviour (see Fig. 7). In this case the curves fit very well together up to a load in the shell of about 250 kN or a bolt force of about 465 kN. This means that up to these loads the reaction of the system can be described with superelements.

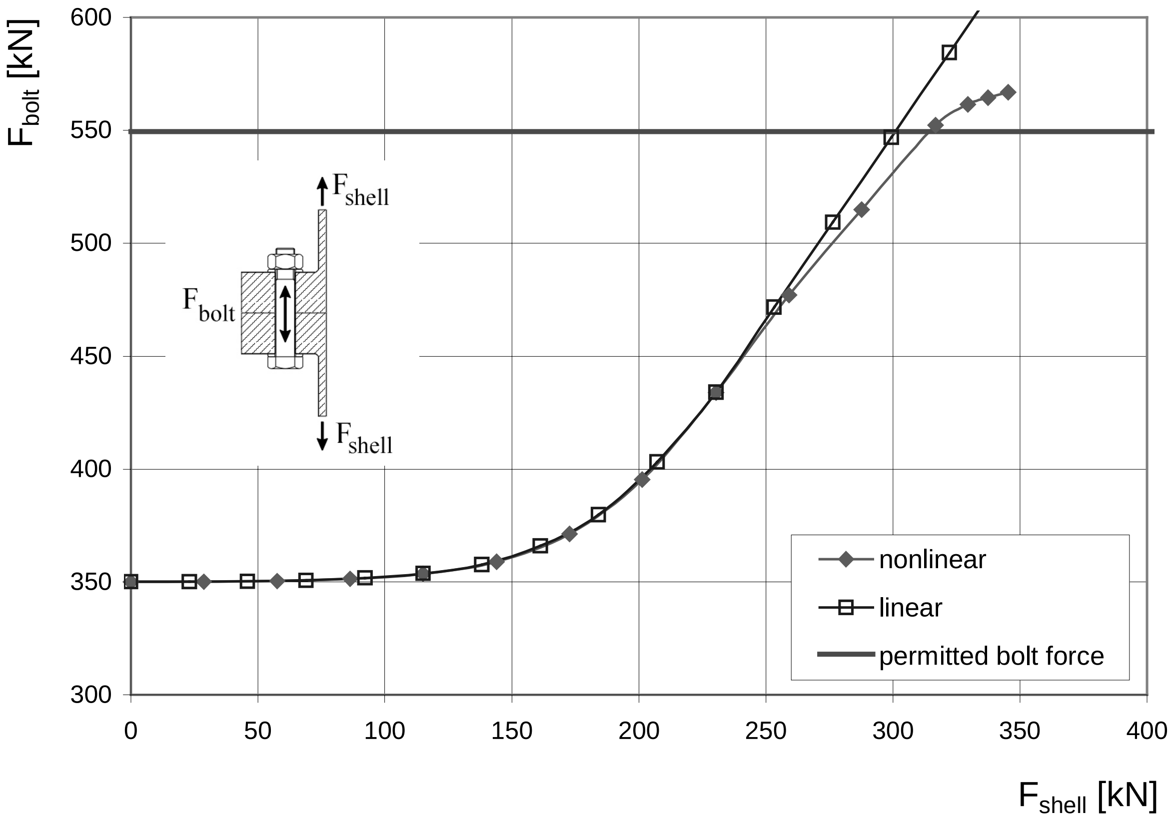


Fig. 7: Segment model of flange 2 with different material properties

3.1.3 Check of the Adapted Model

The area of the ring flange in which superelements can be applied depends on the loading of the joint and the number of bolts used in the flange because this parameters determine the force which have to be carried by the single segments. If there is only a bending moment, more than half the number of the bolts can be described with superelements. On the tension side the forces in the segments around the midline can be expected to be smaller than the critical value while on the compression side plastic deformations are hardly to be expected. The utilised discretisation has to be reviewed after the calculations if the assumed number of superelements was not to great. This can be done by comparing the forces in the bolt for all segments. As an example in Fig. 8 these forces are displayed for ring flange 2 for two different external bending moments. For the smaller moment the bolt force in the last superelement seems to be slightly to high and the results can be accepted while for the higher bending moment the bolt forces in the last two or three superelements are to high that for this loading the results can not be accepted and the number of superelements has to be reduced.

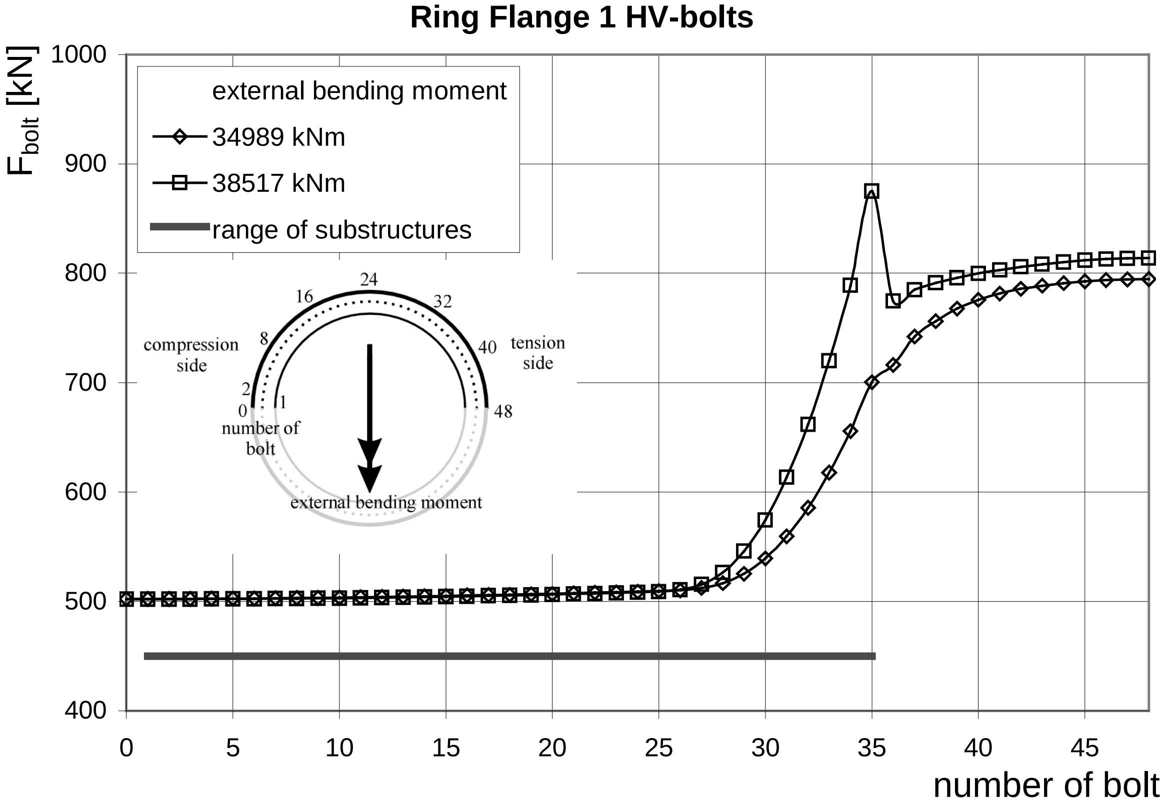


Fig. 8: Control of the results of the substructure-modelling

3.2 Results of Calculations

With FE-models which are validated with the described method ring flange joints with the dimensions of Tab. 1 were calculated and the results were compared with the extrapolation of results of segment calculations. In addition to the HV-bolts with the typical small ductility, the HVN-bolts as mentioned above were taken into account. The results for flange 1 are shown in Fig. 9 as a typical example. By considering the behaviour of the overall ring flange a significant increase of the ultimate loading is possible particular for the bolts with the enlarged plastic deformability. The increases are about 11 to 24%. The failure of the joint has been determined by the limited elongation of the bolts. Therefore the adaptation of the HVN-bolts lead to higher carrying capabilities, as the possible stress rearrangements are significant higher. These higher rearrangements reduce the advantages of the substructure technique because plastic deformation occur in a greater region of the ring flange and the possible number of substructures is lower and the calculation amount raises up.

Not only the stresses for ultimate loading are reduced but also for fatigue loading. The difference between the segment and the overall flange calculations for the same bolt are moderate and significant only for higher loads. The advantages of the utilisation of HVN-bolts instead of HV-bolts are mainly to increase the ultimate load.

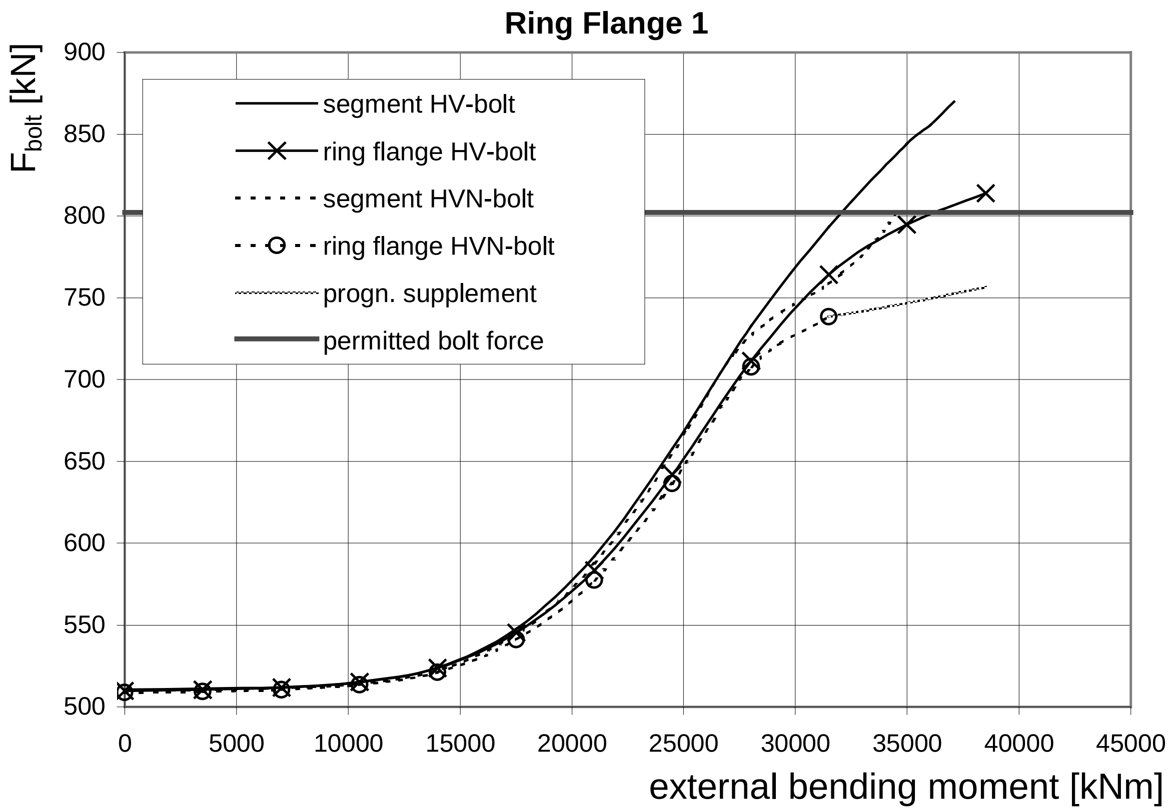


Fig. 9: Development of the bolt forces in the bolt with highest stresses, comparison segment model – overall model; HV-bolt- HVN-bolt

4 Conclusions

The modelling of a ring flange with the commercial FE-software-package ANSYS is presented in this report. Especially the differences between calculations of a segment and an overall ring flange reduced only by symmetry are pointed out. To obtain realistic loading-deformation relations the modelling of the bolts is shown in detail including a specific term of failure. To reduce the required computing time for the calculations substructure modelling has been applied. The advantages and restrictions of this technique for this example are described.

The global structural behaviour of a ring flange joint has significantly higher carrying capacities in comparison to the extrapolated segment calculations. These are limited in the investigated examples by the specific dimensions of the HV-bolts according to the German standard DIN 18800 [2], which have to be used for flange joints under dynamic loading typical for wind energy converters. The limitation can be explained by the short length of the free loaded thread where the plastic deformations occur. The utilisation of bolts with a higher ductility could lead to a higher safety in the ring flange joints not only for ultimate loading but also for fatigue loading.

With the presented modelling techniques the required computing time for the calculations can be reduced significantly. The deformation behaviour of the bolts for elastic and plastic loading has been calibrated. By using substructure modelling results with high exactness can be obtained. Restrictions for the advantages of the substructure modelling exist, when plastic deformations occur in great parts of the ring flange.

5 References

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