Design Optimization of an Articulated Boom for NET In-Vessel Handling Unit

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DESCRIPTION OF THE PROBLEM

In-vessel components of fusion reactors must be remotely replaceable. The necessary handling will be performed from inside the torus by means of work units. A major problem is to carry the work units inside the torus.

One concept to solve this problem is to use an in-vessel handling unit based on an articulated boom as shown in Fig. 1. It is supported outside the torus and enters the torus through an entry port. Additional supports are not available. Then the work unit (manipulator unit, divertor handling unit or antenna handling unit), attached to the end-frame of the boom, is able to reach any point inside the torus.

Therefore the boom consists of eleven links connected by yaw joints. Its streched (unfolded) length is about 25 m. Due to the scissor type of design, the boom can be folded such that the required area to store it is only 10.25 x 3.2 m. The cross-sections of the links (except those staying outside the torus) are 350 x 1350 mm. In order to allow easy repair and exchange, the drive mechanisms for the joints and the necessary cable are located above the links. The resulting overall dimensions are such that the boom may pass the entry port having an opening of 650 x 1900 mm.

The maximum load at the tip of the boom is about 3900 kg. It consists of the maximum payload of 1000 kg (which is the load of a divertor plate plus gripper) and the load of the divertor handling unit of 2900 kg.

The design of the boom such that the stresses and strains are within allowed limits turned out to be a difficult task. It led to a boom dead load of about 25000 kg which is 25-times the payload. In this paper the structural mechanics assessment to find an appropriate design is described.

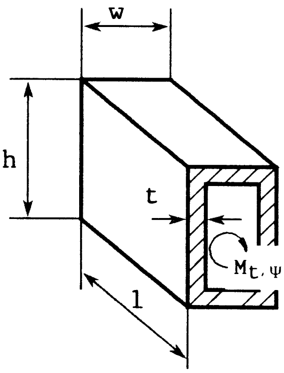
BOX TYPE OF LINKS OR LATTICE GIRDERS ?

Since under the given conditions the stresses reach rather high values, an investigation was carried out, whether lattice girders would be more favorable for the boom than the box type shown in Fig. 1. As far as beam bending is concerned the maximum stresses occur at the upper and lower fibers of the boom cross-sections. Therefore material accumulation at these fibers which can be provided by a lattice design would be advantageous.

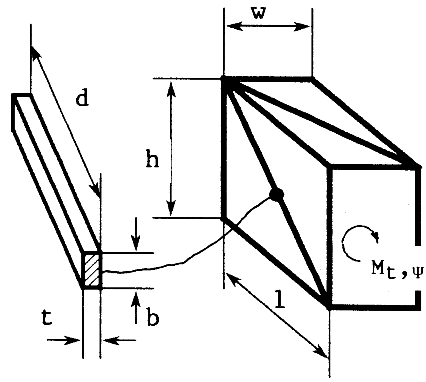
However, depending on the particular boom position, torsion of the boom cross-section may be superimposed causing high shear stresses, too. Since these stresses are inversely proportional to the local thickness of the wall surrounding the boom cross-section, for instance, material accumulations are not advantageous. This can be demonstrated by the formulas for the maximum stress σ and the torsion angle ψ caused by a torque M_t.

(E = Young modulus, G = shear modulus)

box type design



lattice girder



$$\sigma = \frac{M_t}{2hwt} \cdot 2 \qquad\qquad \sigma = \frac{M_t}{2hwt}\,\frac{d}{b}$$

$$\psi = \frac{M_t}{2h^2w^2t}\,\frac{(h+w)\ell}{G} \qquad\qquad \psi = \frac{M_t}{2h^2w^2t}\,\frac{\ell^3+d^3}{Eb}\cdot 2$$

Because

$$d \gg b$$

and

$$\frac{\ell^3+d^3}{Eb} \cdot 2 \gg \frac{\ell(h+w)}{G}$$

the stress σ and the torsion angle ψ for the lattice girder is considerably higher than for the box type of design. Therefore, with respect to torsion the boxes are more suitable.

Considering both, beam bending and torsion, the drawback of the boxes for the first type of loading is moderate, but the advantage in the case of torsion - as just described - is dominant.

OPTIMIZATION OF DESIGN DETAILS OF THE LINKS

Usually, a reduction of stresses can be easily obtained by increasing the wall thickness. However, for the boom, where the dead load is much higher than the payload, an overall increase of the wall thicknesses would result in a considerable increase of the loadings. So the net effect for the stress reduction would be marginal. Therefore the increase of the wall thickness has to be restricted to those regions where it is really necessary. Fig. 2a, for instance, shows the stress distribution caused by bending, Fig. 2b by torsion of a link. As can be seen, stress peaks occur where the hinge plates are attached to the link box. In order to reduce these peaks, the wall thicknesses are increased only in the end regions of the box. Furthermore, local stiffeners are introduced which allow a better load transfer from the hinge plates to the link box. Both improvements are illustrated in Fig. 3. The stress peaks are intensified at the bents of the links which are necessary to allow complete folding. Rounding of the transition region from the box to the hinge plates is not possible. However, a change of the angles is possible according to Fig. 4. This has two effects which reduce the stress peaks. The bent between the side wall of the box and the outer surface of the hinge plate is smaller and the width of the hinge plate base is larger. The analysis showed that the reduction of the stress peak is about 20 %.

Other regions of rather high stresses are the front plates of the link boxes. It is quite impressive that according to Fig. 2b the stresses in the front plate are as high as the stresses in the side walls although the thickness of the front wall is about 1.5 times the thickness of the side wall. The reason is that the torsion is caused by forces acting at the hinge plates which tend to deform the cross-section of the link box like a parallelogram. That means, the stresses in the front plate are a force application problem.

Considering these proposals a construction of the links was carried out utilizing the load carrying capacity of the material as far as possible (sum of membrane and wall bending stress $\leq$ 150 N/mm^2). The resulting overall wall thicknesses of the links are listed in Table I.

Now the deflections of the boom for a payload of 10 kN were calculated. The results are shown in Fig. 5. They do not include the deformations of the joints. For the streched boom where only bending occurs the deflection at the tip was 22 mm. For the boom in the 135° position where both bending and torsion occur the deflection was 27 mm. The deflection due to the dead load is not considered here, since it can be compensated.

DESIGN OF THE JOINTS

Basic considerations show that for the joints between the links a three-hinge plate construction according to Fig. 6a is more appropriate than a two-hinge plate construction, Fig. 6b. Nevertheless, first the two-hinge plate construction was favored because it would have advantages for the remote repair and exchange of failed components. For this construction lower forces occur when the bearings are integrated into the outer hinge plates and the pivot pins are clamped at the inner hinge plates.

However, the bending moment at the inner hinge plates turned out to be so strong that the plate width was not sufficient to carry the resulting stresses. An increase of the plate thickness would not have helped very much, since it would have increased not only the capacity to carry the bending moment but also the bending moment itself. Therefore, the three-hinge plate construction had to be chosen. Fig. 7 shows details of its design. Despite of its complexity a remote exchange of the bearing should be possible.

MATERIAL SELECTION

For an appropriate material selection different criteria were considered.

One criterion was the minimization of the certical deflection caused by a payload. For a given geometry of the link (length, width, height, wall thickness) the deflection is proportional to 1/E, where E is Youngs modulus. The lowest value results for the material with the highest Youngs modulus, which is steel.

Another criterion was the minimization of the vertical deflection caused by the dead weight. Now the deflection is proportional to ρ/E where ρ is the material density. Again, the lowest ratio is obtained for steel, but the ratio for titanium is only slightly (15 %) higher.

A third criterion was the minimization of the dead weight. It is proportional to ρ/σ_{all} where σ_{all} is the allowed stress. Here the lowest ratio is obtained for titanium.

Table II gives more detailed results of applying the above criteria.

The comparison shows that steel is the most favorable material. Only if a small value for the dead weight is required (small inertia forces) or if the loads of the bearings exceed the allowed limits, titanium or a combination of steel and titanium is a reasonable selection. Fig. 8 shows the deflection of the boom caused by its dead weight when the material is steel, titanium and combinations of steel and titanium.

CONCLUSIONS

It is shown that an articulated boom of cantilever type can be built for remote handling inside the NET torus. For the links a box type design is most appropriate. The special construction described here (scissor links) allows a narrow folding of the boom. However, care must be taken to reduce the stress peaks at the link bents and at the clamping of the hinge plates. The resulting deflections are still within reasonable limits.

REFERENCES

Suppan, A. Hübener, J. Investigations of In-Vessel Handling Concepts for NET. IAEA Technical Committee Meeting on Robotics and Remote Maintenance Concepts for Fusion Machines, Karlsruhe, Feb. 1988

Tab. I : Wall thicknesses and weights of the boom

link or joint number	overall wall thickness (mm)	reinforced wall thickness (mm)	weight of the link (kg)	weight of the joint (kg)	weight of the drive + cable (kg)
1	6	6	478	216	113
2	6	6	336	251	213
3	6	12	513	267	224
4	10	16	807	308	256
5	10	16	844	370	280
6	12	20	1000	468	304
7	16	25	1280	500	368
8	25	30	1375	698	360
9	40	40	2436	1053	425
10	40	40	2400	1740	430
11	40	40	2138	2314	400
total	-	-	13607	8185	3373

Tab. II : Characteristic properties of different materials which are candidates for the articulated boom

	$\frac{1}{E}(10^{-11}\frac{m^2}{N})$	$\frac{\rho}{E}(10^{-8}\frac{kg}{Nm})$	$\frac{\rho}{\sigma_{all}}(10^{-5}\frac{kg}{Nm})$
steel	0.47619	3.714	4.
titanium	0.95057	4.315	1.
aluminium	1.42857	4.	2.

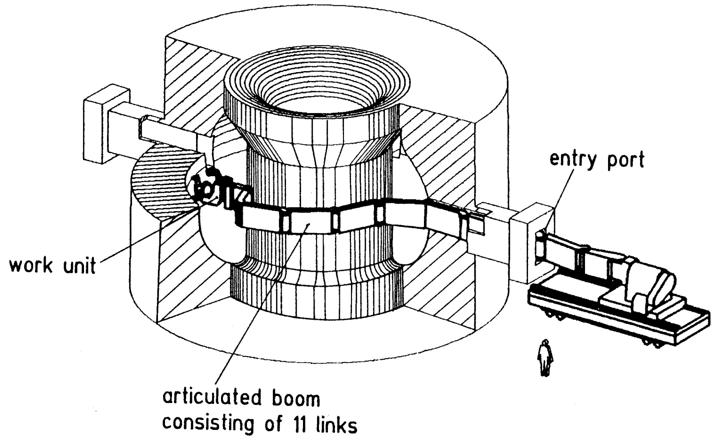
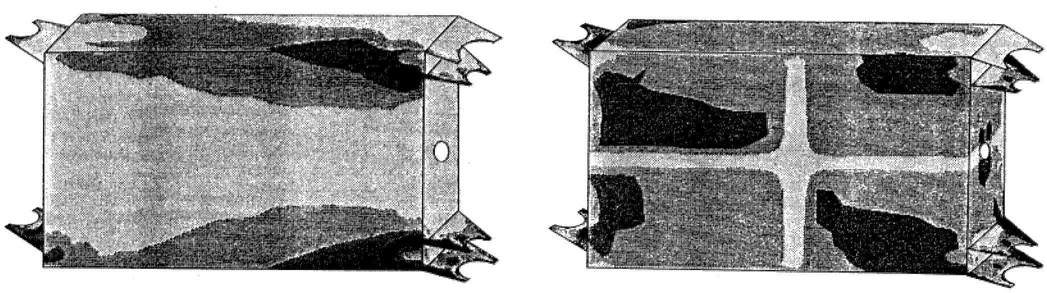


Fig. 1 : In–vessel handling unit



a) stress distribution caused by bending　　*b) stress distribution caused by torsion*

Fig. 2 : Von Mises stress distribution in a link box and the attached hinges

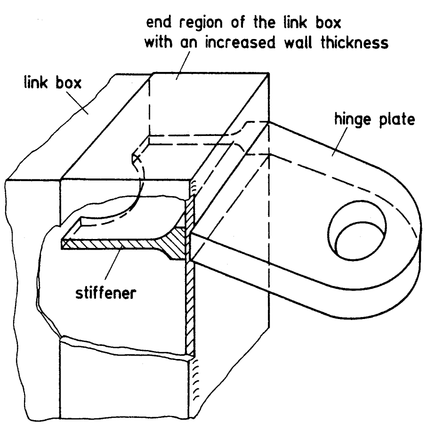


Fig. 3 : Structural improvements in order to reduce stress peaks

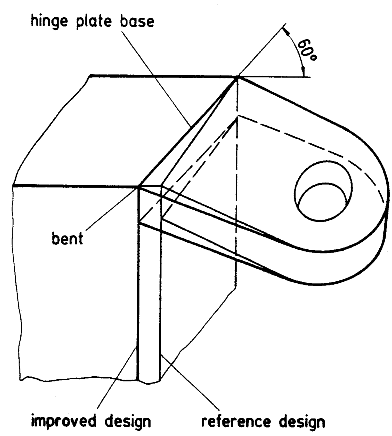


Fig. 4 : Change of angles in order to reduce stress peaks

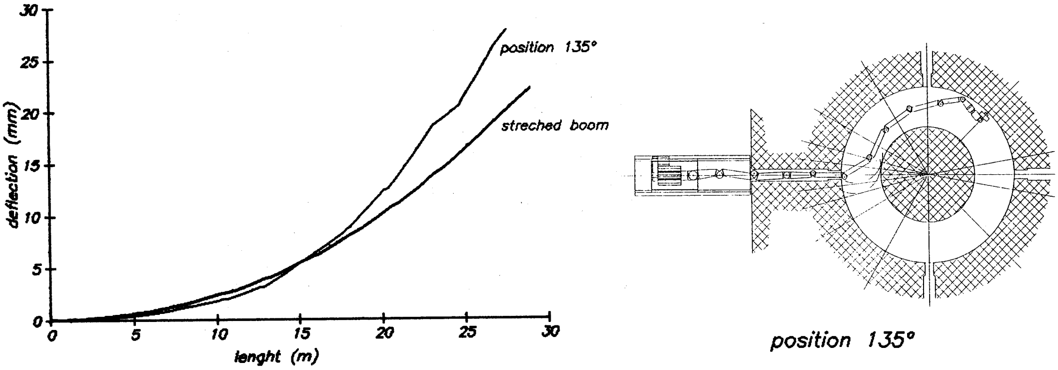


Fig.5 : Deflection of the boom for a payload of 10 kN

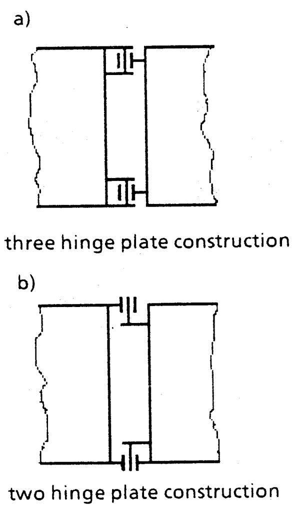


Fig.6 : Types of joints

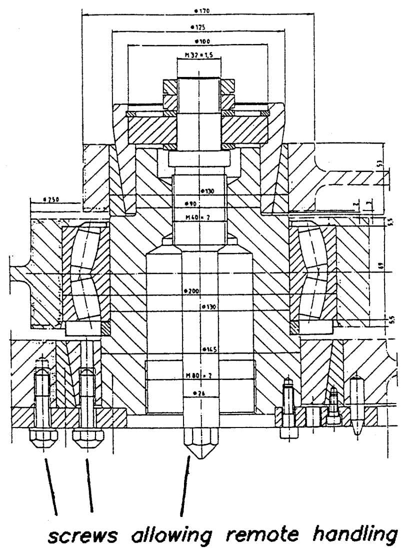


Fig.7 : Details of the three hinge plate construction

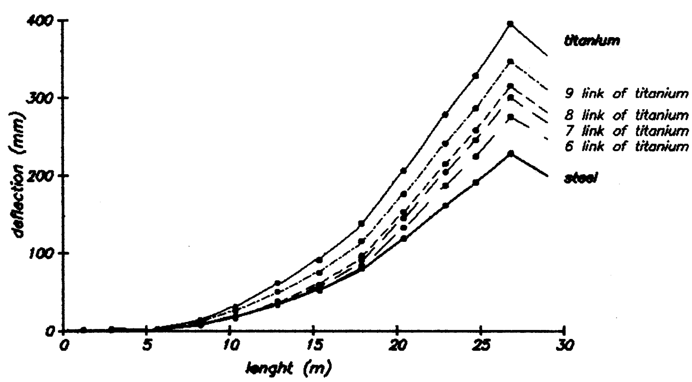


Fig.8 : Deflection of the boom for steel, titanium and different combinations of steel and titanium