Load carrying capacity of canister type blanket elements for NET under accident conditions

E.Wehner, R.Krieg & B.Dolensky
Kernforschungszentrum Karlsruhe GmbH, Institut für Reaktorentwicklung, FR Germany

ABSTRACT

For a NET blanket, consisting of canister type elements, the rupture of the coolant pipes in the elements is a postulated accident to be analysed. In this paper the structural response of an element has been calculated. Provided the ductility of the material of the element including the necessary weldings is sufficient, the pressure in the coolant pipes (60 bar) may be also carried by the elements.

1. DESCRIPTION OF THE PROBLEM

One possible solution for the tritium breeding blanket of NET consists of using canister type elements with solid breeder and helium cooling. The detailed design as decribed in /1/ is shown in Fig. 1. The cross-section of the elements is almost rectangular, about 350 x 450 mm^2, the length is approximately 1500 mm. The thickness of the side walls is 6 mm.

In order to reduce the stresses and the deformations of the side walls, the elements are subdivided by stiffening plates having a thickness of 2 mm. The distance between two stiffening plates is approximately 170 mm.

As material 15MnNi63-steel is assumed with a tensile strength of about 500 N/mm^2.

The heat generated inside the elements will be removed by a system of cooling pipes inside the elements. The coolant flowing through these pipes is helium with a pressure of 60 bar.

A postulated accident is the rupture of such a cooling pipe leading to an overpressurization and a corresponding deformation of the element. In this paper different internal overpressures have been assumed and the resulting element deformations have been calclulated. Of special interest was the pressure of 60 bar which corresponds with the pressure in the coolant pipes and the maximum pressure before failure which describes the safety margin.

2. COMPUTATIONAL PROCEDURE

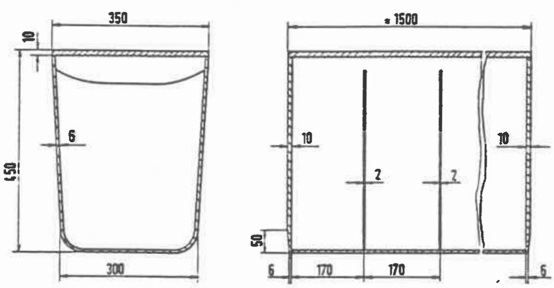
From elastic calculations carried out elsewhere /2/ it turned out that a pressurization of 6 bar already causes relatively high stresses in the order of 300 N/mm^2. Therefore, it was concluded that for a 60 bar pressurization large plastic strains and large geometry changes have to be expected. Considering these conditions the cross-sections of the elements are allowed to approach to a more circular shape increasing the load carrying capacity of the elements considerably. Therefore, the calculations have been carried out with the finite element program ABAQUS /3/ qualified to include both, large plastic strains and large geometry changes. Since right now shell elements do not meet the above requirements, three-dimensional solid elements had to be used, which are based on a logarithmic strain / Kirchhoff stress formulation (the "updated Lagrangian method"). The strains were calculated from the differences of the displacements of the nodes in a post-procress. Owing to the character of the elements (second order isoparametric elements with twenty nodes each), a coarse grid could be applied, as shown in Fig. 2. Therefore the computational effort was within reasonable limits.

3. RESULTS

Fig. 3 shows the results for a quarter section between two stiffening plates. At the complete upper boundary in Fig. 3 rigid clamping was assumed. At a pressure of 66 bar the maximum displacement was 1.5 mm and the maximum strain was about 1 %. At a pressure of 520 bar the maximum displacement was 86 mm and the maximum strain was about 112 %. For pressures exceeding this value no solution could be found. From the strong strain increases and geometry changes versus pressure it was concluded that plastic instability has occured.

The results of this calculation were used to optimize the canister element design. The thickness of the side walls was reduced from 6 mm to 5 mm and the thickness of the stiffening plates was increased from 2 mm to 3 mm (the 2 mm stiffening plates would not have been able to carry the required supporting forces).

Figure 4 shows the results for a quarter section between two stiffening plates of the optimized design. The stiffening plates were assumed to be rigid, but this time it was taken into account, that the stiffening plates do not cover the hole cross section, but leave a part of the side walls unsupported (s. Fig. 1). At a pressure of 62 bar the maximum displacement was 7 mm and the maximum strain was about 3 %. At a pressure of 186 bar the maximum displacement was 31 mm and the maximum strain was about 38 %. For pressures exceeding this value no solution could be found, because of local numerical problem at point A where the boundary conditions change.



Cross–section *Longitudinal–section*

Fig. 1: Geometry of canister element

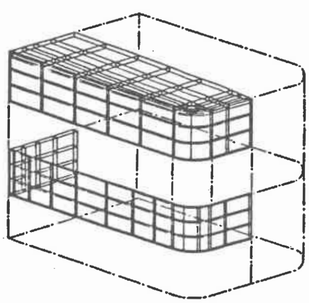


Fig.2: Finite element model

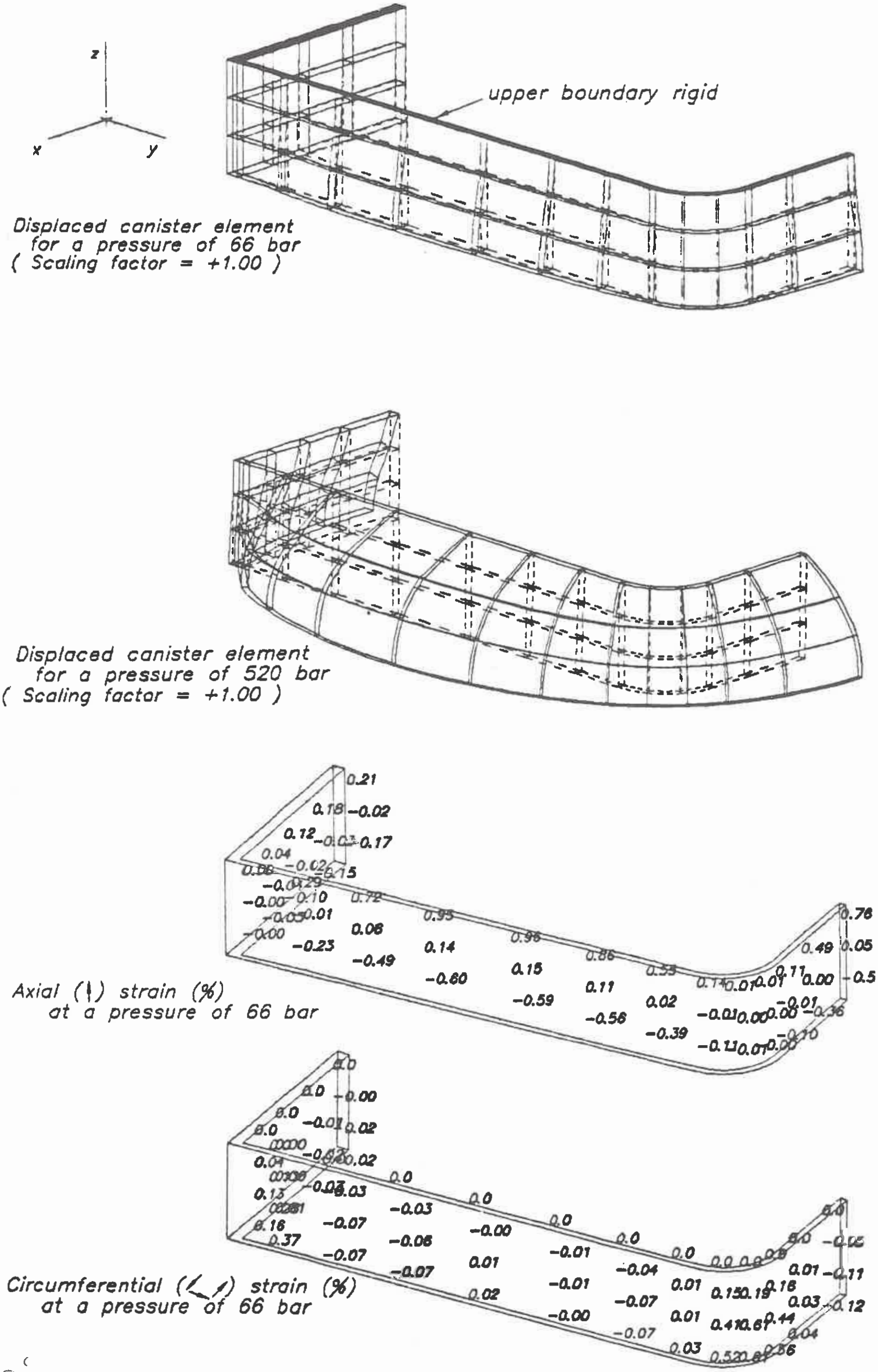


Fig.3: Results for a qarter section between two stiffening plates with the initial configuration

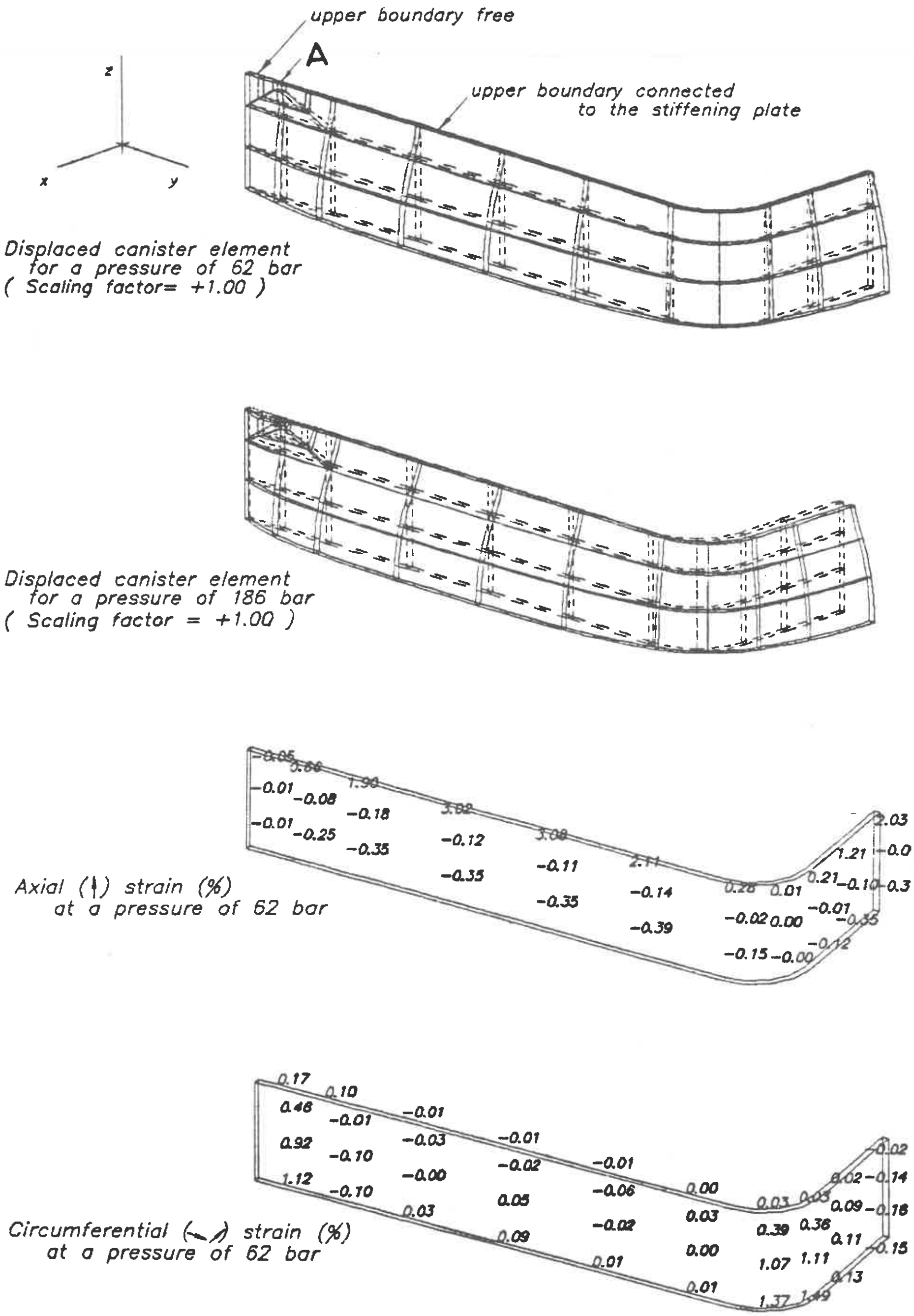
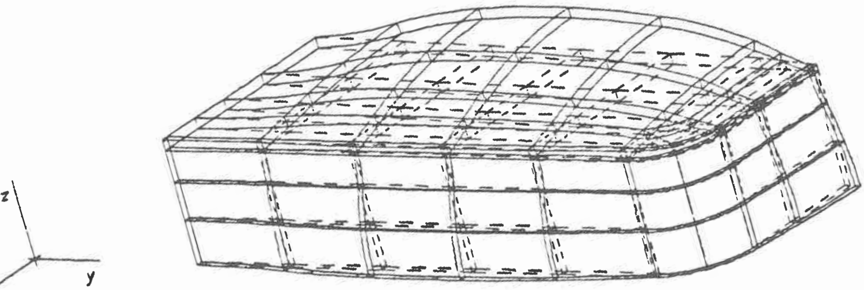
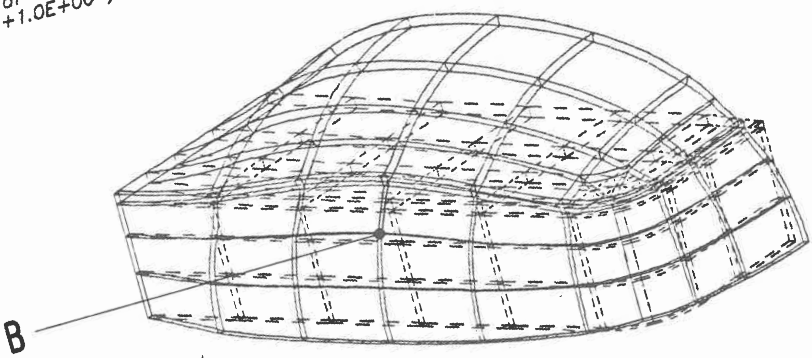


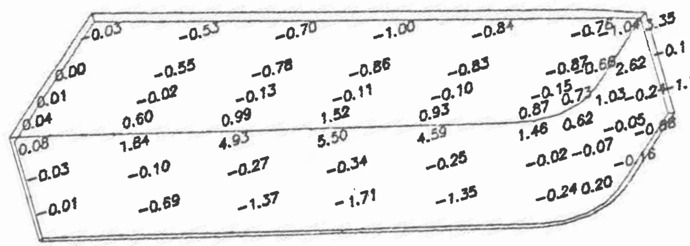
Fig.4: Results for a qarter section between two stiffening plates with the optimized configuration



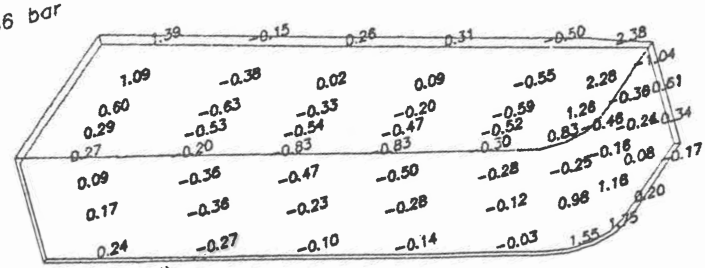
splaced canister element
for a pressure of 66 bar
Scaling factor = +1.0E+00)



Displaced canister element
for a pressure of 250 bar
(Scaling factor = +1.0E+00)



Axial (↕) strain (%)
at a pressure of 66 bar



Circumferential (⇄) strain (%)
at a pressure of 66 bar

Fig.5: Results for a qarter section including the end plates
with the optimized configuration

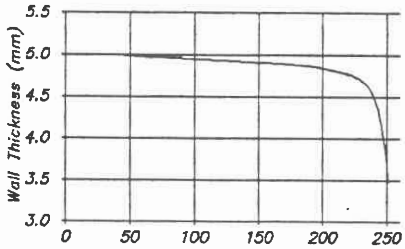


Fig.6: Geometry changes

Figure 5 shows the results for the quarter section including the end plate. At a pressure of 66 bar the maximum displacement was 29 mm and the maximum strain was about 9 %. At a pressure of 250 bar the maximum displacement was 70 mm and the maximum strain was about 65 %. For pressures exceeding this value no solution could be found. As in the first case, the strong strain increases and geometry changes versus pressure indicate that plastic instability has occured. Fig. 6 shows the thickness decrease versus pressure at location B, for example.

4. CONCLUSIONS

For low pressures in the order of 5 bar the stresses already reach rather high values close to the yield stress. Nevertheless, it turns out that even much higher pressures can be carried by the canister elements. In this case the deformations increase such that the cross-section of the element is slowly approaching a more circular shape. As a consequence, membrane stresses occur which cause an essential increase of the load carrying capacity. In other words, the large deformations stiffen the canister elements. Provided the ductility of the material including the necessary weldings is sufficient, even the coolant pressure of 60 bar can be carried. Assuming unlimited ductility plastic instability is reached for a pressure of 520 bar in the first case (canister element between stiffening plates with a thickness of the side wall of 6 mm) and 250 bar in the third case (canister element including the end plate with a thickness of the side wall of 5 mm).

Since calculations of this type represent a field with poor experience, an experimental investigation of the deformations and the load carrying capacity of the canister type element is felt to be necessary in order to check the results.

REFERENCES

/1/ M. Dalle Donne, U. Fischer, E. Bojarsky, H. Reiser, P. Norajitra, E. Bogusch, Pebble Bed Canister: a Ceramic Breeder Blanket with Helium Cooling for NET, 14th SOFT, Avignon, Sept. 1986

/2/ E. Bogusch, Weiterführende Untersuchungen zum Blanketkonzept mit lobularen Elementen für NET, unpublished

/3/ ABAQUS - A General Purpose Linear and Nonlinear Finite Element Code, Update to Version 4.5, Hibbit, Karlsson and Sorensen, Inc.

/4/ FEMGEN - A General Finite Element Mesh Generator, Update to Version 8.5, IKO-Software Service GmbH, Albstadtweg 10, 7000 Stuttgart 80

/5/ FEMVIEW - A General Finite Element Analysis Post-processor, Update to Version 3.5, IKO-Software Service GmbH, Albstadtweg 10, 7000 Stuttgart 80