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STRUCTURAL ANALYSIS OF VACUUM VESSEL AND BLANKET SUPPORT SYSTEM FOR INTERNATIONAL THERMONUCLEAR EXPERIMENTAL REACTOR(ITER)

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Structural analyses of vacuum vessel and blanket support system have been performed to examine their integrated structural behavior under the design loads and to assess their structural feasibility, with two kinds of three-dimensional(3-D)FEM models;a detailed model with 18° sector region to investigate the detailed mechanical behaviors of the blanket and vessel components under the several symmetric loads, and a 180° torus model with relatively coarser meshes to assess the structural responses under the asymmetric VDE load. The analytical results obtained by both models were also compared for the several symmetric loads to check the equivalent mechanical stiffness of the 180° torus model.

As the results, most of the vessel and blanket components have sufficient mechanical integrities with the stress level below the allowable limit of the materials, while the lower parts of inboard/outboard back plate need to be reinforced by increasing the thickness and/or mounting a toroidal ring support at the lower edge of the back plate.

Two types of eigenvalue analyses were also conducted with the 180° torus model to investigate natural frequencies of the vessel torus support system and to assess the mechanical integrity of the elastic stability under the asymmetric VDE load.

This work was conducted as an ITER Design Task(Task No. D307).

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Analytical results show that the mechanical stiffness of the vessel gravity support should be higher in the view point of a seismic response, and that those of the blanket support structures should also be increased for the buckling strength against the VDE vertical force.

Keywords: ITER, Modular Blanket, Support System, Structural Analysis, Disruption Loads, Natural Frequency, Buckling Load Factor

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ITER真空容器・ブランケット支持構造体の構造解析評価

日本原子力研究所那珂研究所ITER開発室

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（1996年10月18日受理）

真空容器及びブランケット支持構造系の負荷時の機械的挙動の把握と各部の機械的健全性評価について，真空容器及びブランケット支持構造系の３次元FEM構造解析を以下の２種類のモデル；軸対称荷重下での機器細部の機械的健全性評価を目的にした18° セクタモデルおよび非軸対称荷重下での機器の機械的挙動の把握を目的にした180° トーラスモデル，で行った。

その結果，真空容器及びブランケット支持構造系の大部分の部位は使用材料の許容応力以下に収まり十分な機械的健全性を有するものの，後壁インボード／アウトボード下部は板厚（100mm）を増加させるか又はトロイダル補強リング構造による補強する必要がある。

又，180° トーラスモデルを用いて，トーラス支持系の固有振動数およびトーラス支持系の非軸対象VDE荷重に対する座屈強度について固有値解析を実施した。その結果，真空容器重力支持脚は自信荷重対策から，又，ブランケット支持多層板バネ構造物はVDE荷重に対する座屈強度の観点から，各々機械的剛性を増加する必要があることが分かった。

本作業は，国際熱核融合実験炉（International Thermonuclear Experimental Reactor）の工学設計活動として，1995年設計作業計画（Task No.D307）に基づいて実施した。

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Contents

目　　次

1. Introduction

A modular-type blanket structure has received consideration for use as a reference design concept of the ITER-EDA(Engineering Design Activities of the International Thermonuclear Experimental Reactor) for its easy maintenance scheme and structural reliability[1]. The blanket modules are welded to back plate of a cylindrical and thick shell structure in the toroidal direction, which is supported with inboard and outboard blanket support structures of multi-layered flexible plates from double-walled vacuum vessel. In addition to high heat loads during normal and off-normal operations, the large electromagnetic force acts on the blanket structure at the plasma disruption. Especially, VDE(Vertical Displacement Events) disruption load gives a large impact to mechanical integrity of in-vessel components, for its large magnitude and asymmetric load distribution in the toroidal direction. These loads acting on the blanket modules are supported with the gravity support legs of vacuum vessel through the blanket support structures, back plate and main parts of vacuum vessel, including the weight loads of their structural components. Thus, a knowledge of the integrated mechanical behaviors of vacuum vessel and blanket support system under the above loads is of great importance.

Then, structural analyses of vacuum vessel and blanket support system have been performed to examine their integrated structural behavior under the design loads and to assess their structural feasibility, with two kinds of three-dimensional (3-D) FEM models; a detailed model with 18° sector region to investigate the detailed mechanical behaviors of the blanket and vessel components under the several symmetric loads, and a 180° torus model with relatively coarser meshes to assess the structural responses on the displacements and stress values on the components under the asymmetric VDE loads. The analytical results on the displacements and stress values obtained by both models were compared to check the equivalent mechanical stiffness of the 3-D 180° torus model.

Two types of eigenvalue analyses were also conducted with the 180° torus model to investigate natural frequencies of the vessel torus support system and to assess the mechanical integrity of the elastic stability under the asymmetric VDE load.

2. Structural Analyses

2.1 A 18° Sector Model

To investigate the structural integrity of the blanket support system, an overall structural analysis has been conducted with 3-D detailed FEM 18° sector model including blanket modules, module attachment support legs, back plate, inboard/outboard blanket support structures and double-walled vacuum vessel.

2.1.1 Analytical Conditions

(1) FEM Modeling

1. Introduction

A modular-type blanket structure has received consideration for use as a reference design concept of the ITER-EDA(Engineering Design Activities of the International Thermonuclear Experimental Reactor) for its easy maintenance scheme and structural reliability[1]. The blanket modules are welded to back plate of a cylindrical and thick shell structure in the toroidal direction, which is supported with inboard and outboard blanket support structures of multi-layered flexible plates from double-walled vacuum vessel. In addition to high heat loads during normal and off-normal operations, the large electromagnetic force acts on the blanket structure at the plasma disruption. Especially, VDE(Vertical Displacement Events) disruption load gives a large impact to mechanical integrity of in-vessel components, for its large magnitude and asymmetric load distribution in the toroidal direction. These loads acting on the blanket modules are supported with the gravity support legs of vacuum vessel through the blanket support structures, back plate and main parts of vacuum vessel, including the weight loads of their structural components. Thus, a knowledge of the integrated mechanical behaviors of vacuum vessel and blanket support system under the above loads is of great importance.

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2. Structural Analyses

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2.1.1 Analytical Conditions

(1) FEM Modeling

All of structural components such as blanket modules, module attachment support legs, back plate, inboard/outboard blanket support structures and double-walled vacuum vessel are represented with shell elements. Table 2-1 shows materials and thickness of the components in the model[2-4]. Figure 2-1 indicates the FEM model with 18° sector in the toroidal direction. Both toroidal side edges of the vacuum vessel and back plate are under the cyclic symmetry conditions. Details of the blanket modules and back plate, and details of inboard and outboard blanket support structures in the FEM model are shown in Fig. 2-2 and Fig. 2-3, respectively.

The shield block of the modules was represented with box structure of 100 mm thickness. The back plate has locally 200 mm thickness around the blanket support structures. The configuration of the blanket support structures are applied to the model, with multi-layered flexible plates proposed by the VV group in Garching JCT[3]. They are a pair of 7-layered flexible plates with 20 mm thickness, 300 mm width and 1160 mm length for 18° sector inboard blanket support structure, and are formed from 4 sets of 20-layered flexible plates with 12 mm thickness, 600 mm width and 1340 mm length for outboard support structure, respectively, as shown in Fig. 2-3. Though the lengths of both the inboard and outboard blanket support structures in the FEM model are different from actual dimension, equivalent mechanical stiffnesses for the bending moment and normal and shear forces are used in the analysis.

The modified reinforcements on the vacuum vessel components are shown in Fig. 2-4 [4]. The details of the VV gravity support leg are shown in Fig. 2-5, consisting of 3 m long multi-layered flexible plates and 3 m long box-type rigid support leg.

Two types of gravity support leg configurations on multi-layered flexible plates were considered, including separate type support leg as the reference design and integrated type one as an alternative. The lower edge of the box-type gravity support leg was completely fixed in the model.

(2) Load Conditions

Following symmetric loads were applied for the analysis, including weight loads, electromagnetic loads and thermal load. The weight loads on the structural components were set to be 3.1 MN uniformly for 18° sector of the vacuum vessel, to be 0.6 MN and 1.9MN for 18° sector of the inboard and outboard blanket support system, respectively, and to be 0.38 MN and 0.38 MN concentratedly for 18° sector of the inboard and outboard divertor system at the lower vessel locations of R=6.4 m and 9.5 m, respectively, as shown in Table 2-2.

The centered disruption and VDE(Vertical Displacement Event) disruption loads were considered as the electromagnetic loads. The electromagnetic pressures on the first wall and shearing forces on both side walls of the blanket modules are shown in Fig. 2-6(a) as the centered disruption loads. The VDE disruption loads were applied to the lower parts of blanket modules and lower portions of the vacuum vessel corresponding to the divertor support rail locations. The VDE load has a load distribution with the averaged electromagnetic pressures on the first walls of #1, 2, 13, 14, and #15 blanket modules, as shown in Fig. 2-6(b)[5], except asymmetric load components.

The vertical loads of 2.27 MN and 0.93 MN were applied at the lower vacuum vessel locations of R=6.4 m and R=9.5 m as the VDE load on the inboard and outboard divertor sector, respectively.

In addition, overall thermal loads were also applied to the in-vessel components, assuming a uniform temperatures of 250°C for the blanket modules, 200°C for the back plate and 150°C for the vacuum vessel, respectively. Linear interpolations were considered from 200°C to 150°C for inboard and outboard blanket support structures and from 150°C to 20°C for the upper multi-layered plates of vessel gravity support.

2.1.2 Analytical Results under Symmetric Loads

Structural analyses of the blanket and vacuum vessel support system were performed under the several loads mentioned above, with a finite element structural code, NASTRAN[6].

(1) Dead Weight

The overall deformation of the blanket and vacuum vessel support system under the dead weight is shown in Fig. 2-7(a). It has a maximum deformation of ~6 mm at the lower edge of the outboard blanket(#15 blanket module), which are induced by the vacuum vessel deformations of 4.2 mm at the outboard lower region of the vacuum vessel and rotation displacement of the outboard blanket support. Figure 2-7(b) shows the overall Von Mises stress distribution on the blanket and vacuum vessel support system under the dead weight. The maximum stress of 54 MPa occurred at the lower edge on the vacuum vessel gravity support with the multi-layered flexible plates.

(2) Centered Disruption Load

The overall deformation of the blanket and vacuum vessel support system under the centered disruption load is shown in Fig. 2-8(a). The maximum deformation of ~7 mm appeared at the upper portion of the outboard blanket(#10 blanket module) in the plasma-side direction. Figure 2-8(b) indicates the overall Von Mises stress distribution on the blanket and vacuum vessel support system, which has a maximum stress of 129 MPa at the back plate around the inboard midplane blanket module(#4 module). A relatively large stress of ~110 MPa was observed at the module attachment support leg on the #4 blanket module.

(3) VDE Disruption Load (Symmetric Load Component)

The averaged electromagnetic pressure distribution on the lower blanket modules was applied as an in-plane load on the blanket modules. The overall deformation of the blanket and vacuum vessel support system under the symmetric VDE disruption load is shown in Fig. 2-9(a), which has a maximum deformation of ~29 mm at the lower edge of the outboard blanket(#15 module) in the lower direction. Figure 2-9(b) also illustrates the overall Von Mises stress distribution on the blanket and vacuum vessel support system, with a maximum stress of ~246 MPa at the lower portion on the outboard back plate. The toroidal stiffness of the back plate, therefore, needs to be further enhanced around the lower portions.

(4)Thermal Load

The overall deformation of the blanket and vacuum vessel support system and Von Mises stress distribution on their components under the thermal load, are shown in Fig. 2-10(a) and Fig.

2-10(b), respectively. The maximum deformation of ~52 mm appeared upward and radially at the top part of the outboard blanket(#9 module), and relative deformation between the blanket and vacuum vessel is estimated to be ~20 mm around the outboard top region. The maximum stress of 130 MPa occurred at the top part on the outboard back plate.

Table 2-3 summarizes the analytical results in terms of the maximum displacements and stresses on the blanket vacuum vessel support system under the respective symmetric loads.

The obtained individual results of the blanket and vessel support system were assessed by combination of the above-mentioned four basic load cases, as follows;

Case-1: Dead Weight + Centered Disruption Load
Case-2: Dead Weight + VDE Disruption Load
Case-3: Dead Weight + Centered Disruption Load + Thermal Load
Case-4: Dead Weight + VDE Disruption Load + Thermal Load

The stress evaluation based on the standard of ASME Sec. III [7], which has a design criteria on stress limitation, as shown below;

$$\text{Primary Stress, } Pm + Pb < 1.5*Sm \qquad (2\text{-}1)$$

$$\text{Primary + Secondary Stress, } Pm + Pb + Q < 3*Sm \qquad (2\text{-}2)$$

where, Pm, Pb and Q are a general primary-membrane stress, primary bending stress and secondary stress such as the thermal and peak stresses, respectively. Sm is an allowable stress limit for the general primary-membrane stress of Pm, and the values of SS316 at 150°C to 250°C are as follows;

$$\begin{aligned} Sm &= 142 \text{ MPa} \quad \text{at } 150^\circ C \\ &= 132 \text{ MPa} \quad \text{at } 200^\circ C \\ &= 125 \text{ MPa} \quad \text{at } 250^\circ C \end{aligned}$$

The overall deformations and maximum stress values on the vessel and blanket structural components are shown in Table 2-4 under the four combination load cases from Case-1 to Case-4. As the results, maximum stresses were enough below the allowable stress limit, except that in the load case of Case-2.

2.2. A 180° Torus Model

To investigate an effect of toroidally asymmetric VDE load distribution on the vacuum vessel and blanket support system, the overall structural analysis has been carried out with a FEM 180° torus model including a back plate, inboard/outboard blanket support structures, vacuum vessel and vessel gravity support. The analytical results were also compared with the results by 18° sector model to check the mechanical stiffness of the 180° torus model.

2.2.1 Analytical Conditions

(1) FEM Modeling

The vacuum vessel was represented simply with a single-walled shell elements having an mechanical stiffness equivalent to actual double-walled shell structure. All of the other components, such as the back plate and blanket support structures, were modeled with shell elements, except the vessel gravity support structures represented with beam elements. Figure 2-11 illustrates an overall FEM model with 180° torus region in the toroidal direction, consisting of vacuum vessel, back plate, blanket support structures and vessel gravity support structures. A 18° sector region of the 180° torus FEM model is also shown in Fig. 2-11. Both toroidal side edges of the vacuum vessel and back plate in the model were set to be under cyclic symmetry conditions.

(2) Load Conditions

Following symmetric loads were applied to the analysis with 180° torus model as well as those with 18° sector model, which include the weight load, VDE disruption load and thermal load, except centered disruption load for neglecting the blanket modules in the model. The VDE load has a load distribution with averaged electromagnetic pressures on the back plates which the location corresponds to the first walls of #1, 2, 13, 14 and #15 blanket modules, including asymmetric components with peaking factor of 1.5.

The EM pressures on the back plate locations corresponding to #1, 2, 13, 14 and #15 blanket modules have the distribution in the toroidal direction, given by the following equation.

$$P(\theta)=Pav.(1+0.5*\cos\theta) \qquad (2-3)$$

where, Pav. are the averaged EM pressures on #1, 2, 13, 14 and #15 modules(MPa).

θ is toroidal angle (deg.)

The EM pressure distributions on the F/W of #1, 2, 13, 14 and #15 blanket modules in the toroidal direction are shown in Fig.2-12[6]. These EM pressure distributions were applied directly to the back plate because of neglecting of the blanket modules in the 180° torus model.

2.2.2 Comparison between Results by 18° Sector Model and 180° Torus Model

To check the mechanical stiffness of the 180° torus model, analytical results of both 18° sector and 180° torus models were compared on the displacement and stress level under three basic symmetric loads of the dead weight, VDE disruption and thermal loads.

(1) Weight Load

The overall deformations on the vacuum vessel and blanket support system under the weight load are shown in Fig.2-13(a) for the results with 18° sector model, and also shown in Fig.2-13(b) for the results with 180° torus model. From the comparison of both results, maximum deformations of 6.3 mm for 18° sector model and of 5.8 mm for 180° torus model occurred at the same location of the lower edge on the outboard back plate. Maximum Mises stresses of 54 MPa for 18° sector model and 58 MPa for 180° torus model were similarly induced around the lower part on the inboard back plate.

A good agreement on both the displacement and stress level is obtained between analytical results with 18° sector and 180° torus models.

(2) VDE Disruption Load(Symmetric Load Component)

The overall deformations on the vacuum vessel and blanket support system under the VDE disruption load are shown in Fig.2-14(a) for the results with 18° sector model, and also shown in Fig.2-14(b) for the results with 180° torus model. From the comparison of both results, maximum deformations of 29.2 mm for 18° sector model and of 31.8 mm for 180° torus model were observed at the same location of the lower edge on outboard back plate. Maximum Mises stresses of 246 MPa for 18° sector model and 264 MPa for 180° torus model were similarly induced around the lower part on the inboard back plate.

A good agreement on both the displacement and stress level is obtained between analytical results with 18° sector and 180° torus models.

(3) Thermal Load

The overall deformations and Mises stress distribution on the vacuum vessel and blanket support system under the thermal load are shown in Fig.2-15(a) for the results with 18° sector model, and also shown in Fig.2-15(b) for the results with 180° torus model. From the comparison of both results, maximum deformations of 51.8 mm for 18° sector model and of 57.5 mm for 180° torus model appeared at the similar location of the outboard shoulder part on the back plate. Maximum Mises stresses of 130 MPa for 18° sector model and 35 MPa for 180° torus model were induced.

The overall deformation of the vacuum vessel and blanket support system has a relatively good agreement between the results with the both models, however, considerably large difference on the maximum stress of the structural components appears for both the stress locations and stress values. This seems a main reason because of neglecting the blanket module boxes in the 180° torus model.

Table 2-5 summarizes the comparison of the analytical results by the 18° sector and 180° torus models, for maximum displacements and stress values on the vessel and blanket support system under the respective symmetric loads.

From the above comparison of the analytical results of both the 18° sector and 180° torus models, the mechanical stiffness on the torus model is considered to be reasonable in comparison with that in the sector model, except for the thermal load.

2.3 Mechanical Behavior of Vessel and Blanket Support System under Asymmetric Disruption Load

The structural analysis of the vacuum vessel and blanket support system has been performed with the 180° torus model under an asymmetric VDE load including peaking factor of 1.5 and halo current of 40 % plasma current, shown in Fig.2-12. Figure 2-16 shows an overall deformation and stress distribution on the vacuum vessel and blanket support system under the asymmetric VDE disruption load. Maximum deformation of 32.4 mm occurred at the lower edge on the outboard back plate in the toroidal angle of $\theta = 0^\circ$ and max. stress of 343 MPa was induced around the lower edge on the inboard back plate between adjacent blanket support structures in the toroidal angle of $\theta \sim 60^\circ$, as shown in Fig.2-17. Then, analytical results by 180° torus model were higher by 11% for max. displacement and by 40% for max. stress, compared with those by 18° sector model under the symmetric VDE load with the averaged EM pressures. Inboard back plate has a large stress beyond the allowable stress limit of SS316 under the asymmetric VDE load, so that it needs to be reinforced by increasing the thickness on the lower parts of the back plate.

3. Eigenvalue Analysis

Two types of eigenvalue analyses were conducted with a 180° torus structural model to help the investigation of natural frequencies for the vacuum vessel and blanket torus support system against the seismic load, and to assess buckling load margin of the vacuum vessel and blanket torus support system against the VDE disruption load.

3.1 Natural Frequencies of Vacuum Vessel and Blanket Support System

The analytical FEM model consists of only vacuum vessel and vessel gravity support, while including the weight loads of the back plate and blanket modules, as shown in Fig.3-1. Each of natural frequencies for the vacuum vessel and blanket support system was analyzed for 2 types of vessel gravity support structures, as shown in Fig.2-5.

As the results, the natural frequencies of vacuum vessel torus system were obtained as shown in Table 3-1.

The first deformation mode on the natural frequency of the vacuum vessel torus support system is shown in Fig.3-2, which is the deformation mode so as that the torus system is swayed horizontally toward a same direction. Furthermore, 2nd to 5th deformation modes on the natural frequency of the vacuum vessel and torus support system are shown in Fig.3-3 to Fig.3-6, respectively.

From the results, it is found that mechanical stiffness of the vessel gravity support should be higher in the view point of a seismic response.

3.2 Buckling Evaluation

A buckling analysis of the vacuum vessel and blanket support system was performed with the 180° torus FEM model shown in Fig.2-11, to assess the mechanical integrity of the elastic stability under the load combination of their dead weight and asymmetric VDE load. As the results, all of the first, second and third buckling modes were induced at the outboard blanket support structures against the weight and asymmetric VDE loads and their buckling load factors(safety margin for the buckling load) were also estimated to be 4.11, 4.14 and 4.24, respectively, as shown in Table 3-2.

The blanket support structures, especially outboard support ones, have a marginal buckling back plate in the toroidal angle of $\theta = 0^\circ$ and max. stress of 343 MPa was induced around the lower edge on the inboard back plate between adjacent blanket support structures in the toroidal angle of $\theta \sim 60^\circ$, as shown in Fig.2-17. Then, analytical results by 180° torus model were higher by 11% for max. displacement and by 40% for max. stress, compared with those by 18° sector model under the symmetric VDE load with the averaged EM pressures. Inboard back plate has a large stress beyond the allowable stress limit of SS316 under the asymmetric VDE load, so that it needs to be reinforced by increasing the thickness on the lower parts of the back plate.

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The blanket support structures, especially outboard support ones, have a marginal buckling safety factor of ~4 against the weight and asymmetric VDE loads, so that their mechanical stiffness should be higher for the mechanical buckling strength against the VDE vertical force.

Figure 3-7 shows the first buckling deformation mode of the vacuum vessel torus support system with the separate type structure of the vessel gravity support.

4. Concluding Remarks

Three-dimensional structural analysis of vacuum vessel and blanket system has been performed to investigate their mechanical behaviors and to assess the structural feasibility. The analysis was conducted with two types of FEM models; 18° sector model to investigate the detailed mechanical behaviors of the vessel and blanket components for the several symmetric loads and 180° torus model for the asymmetric loads(in the toroidal direction). The eigenvalue analyses were also conducted with a 180° torus structural model to investigate natural frequencies of the vacuum vessel and blanket torus support system against the seismic load, and to assess buckling load margin of the vacuum vessel and blanket torus support system against the VDE disruption load.

From the studies, following conclusions were drawn;

(1) In the analysis with the 18° sector model, following symmetric loads were considered; weight load, centered disruption load, VDE disruption load and thermal load. Most of the vessel and blanket components have sufficient mechanical integrities within the allowable limits of stress and displacement against the above-mentioned loads, except that stress on the lower part of the outboard back plate exceeds the allowable limit at the VDE disruption load.

(2) The obtained individual results of the blanket and vessel support system were assessed by combination of the above-mentioned four basic load cases with the stress evaluation based on the standard of ASME Sec. III.;

Case-1: Dead Weight + Centered Disruption Load

Case-2: Dead Weight + VDE Disruption Load

Case-3: Dead Weight + Centered Disruption Load + Thermal Load

Case-4: Dead Weight + VDE Disruption Load + Thermal Load

The maximum stresses on the vessel and blanket structural components were enough below the allowable stress limit of SS316, except that around the lower portion on the back plate in the load case of Case-2.

(3) To check the mechanical stiffness of the 180° torus model, analytical results of both 18° sector and 180° torus models were compared on the displacement and stress level under above-mentioned basic symmetric loads. Good agreements are obtained with 18° sector and 180° torus models, except stress value due to thermal load, possibly due to the effects neglecting the modules in 180° torus model. In general, the mechanical stiffness on the

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(3) To check the mechanical stiffness of the 180° torus model, analytical results of both 18° sector and 180° torus models were compared on the displacement and stress level under above-mentioned basic symmetric loads. Good agreements are obtained with 18° sector and 180° torus models, except stress value due to thermal load, possibly due to the effects neglecting the modules in 180° torus model. In general, the mechanical stiffness on the

torus model is considered to be reasonable in comparison with that in the sector model.

(4) Analytical results by 180° torus model under the asymmetric VDE load were higher by 11% for the max. displacement and by 40% for max. stress compared with those by 18° sector model under the symmetric VDE load. Maximum deformation of 32.4 mm occurred at the lower edge on the outboard back plate in the toroidal angle of $\theta = 0°$ and max. stress of 343 MPa was induced around the lower edge on the inboard back plate between adjacent blanket support structures in the toroidal angle of $\theta = \sim 60°$, which is fairly beyond the allowable stress limit of SS316, 198 MPa.

(5) Then, reinforcement of the lower parts of the inboard and outboard back plate or reconsideration of the load conditions for the asymmetric VDE load is needed in the future. The stress level on the support region of the blanket modules to the back plate was enough within the allowable level with the present model, though further detailed analysis is needed to evaluate local peaked stress.

(6) The first deformation mode on the natural frequency of the vessel and blanket support system corresponds to the bending mode of the global torus gravity supports horizontally in the same direction, with a frequency of 1.8 Hz for the separate type gravity support structure and 2.6 Hz for the integrated one. It is found that mechanical stiffness of the vessel gravity support should be higher in the view point of a seismic response.

(7) The first, second and third buckling modes against the weight and asymmetric VDE loads were induced at the outboard blanket support structures with their buckling load factors of 4.11, 4.14 and 4.24, respectively. The blanket support structures, especially outboard support ones, have a marginal buckling safety factor of ~4 against the weight and asymmetric VDE loads, so that their mechanical stiffness should be ihigher for the buckling strength against the VDE vertical force.

Acknowledgment

The authors would like to express their sincere appreciation to Drs. S.Shimamoto, S.Matsuda, M.Seki and T.Tsunematsu for their continuous guidance and encouragement. They also would like to acknowledge Dr. M.Araki for his useful advices and Mr. M.Komatsuzaki of Kanazawa Computer Service for his useful support of FEM analysis.

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(5) Then, reinforcement of the lower parts of the inboard and outboard back plate or reconsideration of the load conditions for the asymmetric VDE load is needed in the future. The stress level on the support region of the blanket modules to the back plate was enough within the allowable level with the present model, though further detailed analysis is needed to evaluate local peaked stress.

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[6] MSC/NASTRAN Ver. 66A User's Manual, The MacNeal-Schwendler Corporation, Nov. 1989.

[7] ASME Boiler and Pressure Vessel Code, Sec. III.

Table 2-1 Materials and Thicknesses of Structural Components

Components	Thickness	Material
Vacuum Vessel		SS316
Inner Skin	40 mm	
Outer Skin	40 mm	
Poloidal Ribs	40 mm	
Upper Port	80 mm(H= 80 mm)	
Mid Port	80 mm(H=200 mm)	
Divertor Port	80 mm(H=200 mm)	
Support Base Plates	100 mm	
Flexible Support Leg	5x100 mm	
Box Support Leg	100 mm	
Blanket Module		SS316
First Wall	100 mm	
Side Wall	100 mm	
End Wall	100 mm	
Top/Botm. Plates	100 mm	
Support Leg	70 mm	
Back Plate		SS316
Inboard	100 mm	
Outboard	100 mm	
Blanket Support Structures		SS316
Inboard	2- 7 Layers-20 mm	
Outboard	2-20 Layers-12 mm	

Table 2-2 Dead Weights of Structural Components

Vacuum Vessel	3.1	MN/ 18° Sector,Uniformly.
Blanket System	0.6	MN/ Inboard Sector and
(Modules & B.P)	1.9	MN/ Outboard Sector,Uniformly.
Divertor	0.38	MN/ Inboard Sector at R=6.4 m and
	0.38	MN/ Outboard Sector at R=9.5 m, Concentratedly.

Table 2-3 Results of Max. Displacements and Stresses due to Symmetric Loads conducted by 18° Sector Model

Load	Max. Disp.(mm)/Location	Max.Stress(MPa)/Location
Dead Weight	6.3 mm /Lower Edge of Outb. BLK	54 MPa /VV Grav. Suppt.
Centered Disruption	7.3 mm /Outb. BLK Shoulder	129 MPa /Inb. Back Plate
VDE Disruption	29.2 mm /Lower Edge of Outb. BLK	246 MPa /Lower on Outb. Back Plate
Thermal Load	51.8 mm /Top of Outb. BLK	130 MPa /Top on Outb. Back Plate

Table 2-4 Results of Max. Displacements and Stresses due to Combination Loads

Load Case	Disp.(mm)/Location	Stress(MPa)/Location
Case-1	12.3 mm /Lower on Outb. BLK	131 MPa /Inb. Back Plate
Case-2	35.2 mm /Lower Edge on Outb. BLK	262 MPa /Lower on Outb. Back Plate
Case-3	46.9 mm /Top on Inb. BLK	193 MPa /Attach. Leg on Inb. BLK
Case-4	43.7 mm / Lower Edge on Outb. BLK	323 MPa /Lower Edge on Inb. Back Plate

Notes; Load Combinations considered in the Analysis are as follows;

Case-1:Dead Weight + Centered Disruption
Case-2:Dead Weight + VDE Disruption
Case-3:Dead Weight + Centered Disruption + Thermal Load
Case-4:Dead Weight + VDE Disruption + Thermal Load

Table 2-5 Comparison of Results on Max. Displacement and Max. Stress by 18° Sector and 180° Torus Models

Symmetric Load	18° Sector Model Max.Disp./Max.Stress	180° Torus Model Max.Disp./Max.Stress
Dead Weight	6.3mm/54MPa	5.8mm/58MPa
Centered Disruption	7.3mm/129MPa	- / -
VDE Disruption (Averaged Press.)	29.2mm/246MPa	31.8mm/264MPa
Thermal Load	51.8mm/130MPa	57.5mm/ 35MPa

Table 3-1 Natural Frequencies of VV Torus Support System

VV GravitySupport	Separate Type	Integrated Type
1st Frequency	F=1.83 Hz	F=2.64Hz
2nd	8.74	9.10
3rd	11.86	11.99
4th	17.70	17.95
5th	18.53	18.53

Table 3-2 Buckling Load Factor, Fb, of Torus Support System

VV Gravity Support	Separate Type
1st Mode	Fb=4.11
2nd	4.14
3rd	4.24

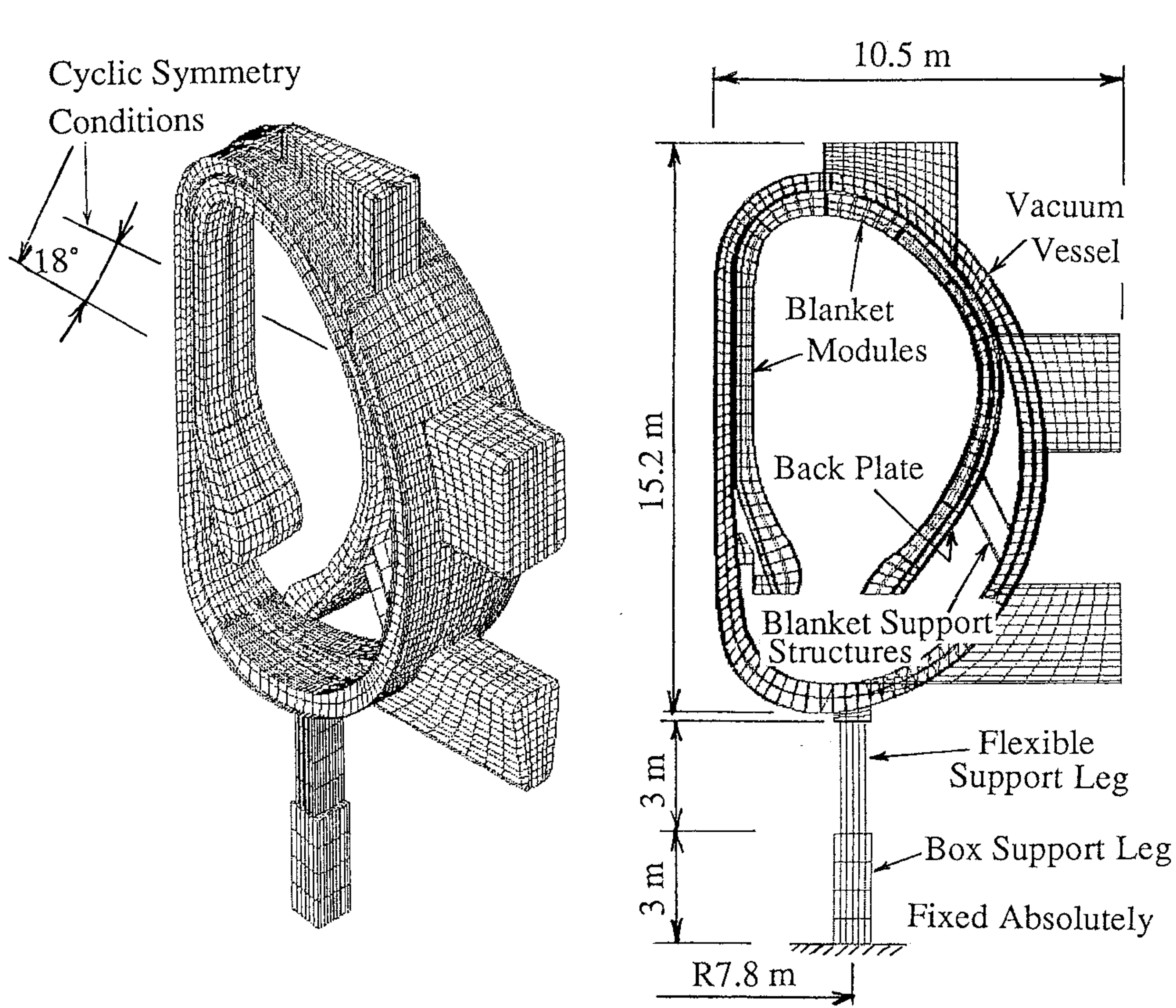
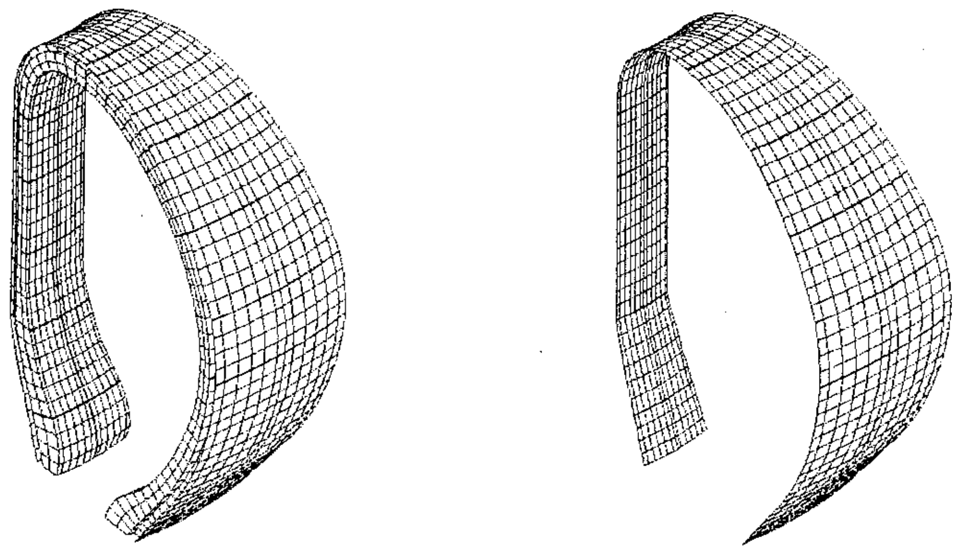
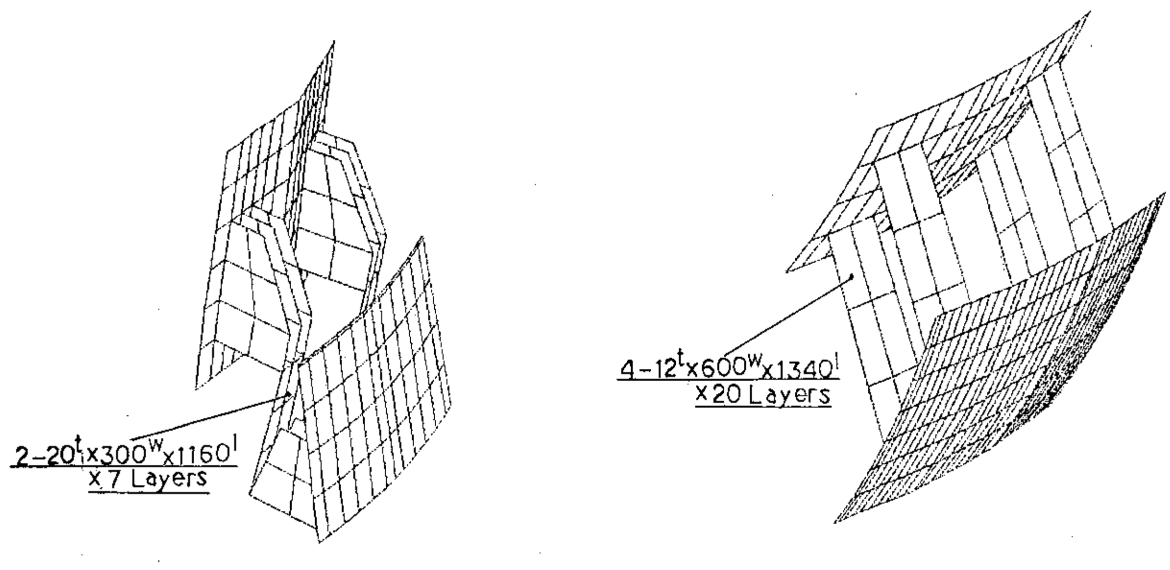


Fig.2-1 A 18° Sector Model of Vacuum Vessel and Blanket System.



Blanket Modules Back Plate

Fig. 2-2 : Details of Blanket Modules and Back Plate.



Inboard Blanket Support Outboard Blanket Support

Fig. 2-3 : Details of Inboard and Outboard Blanket Support Structure Models, with Multi Layered Flexible Plates Proposed by VV Gr in Garching JCT.

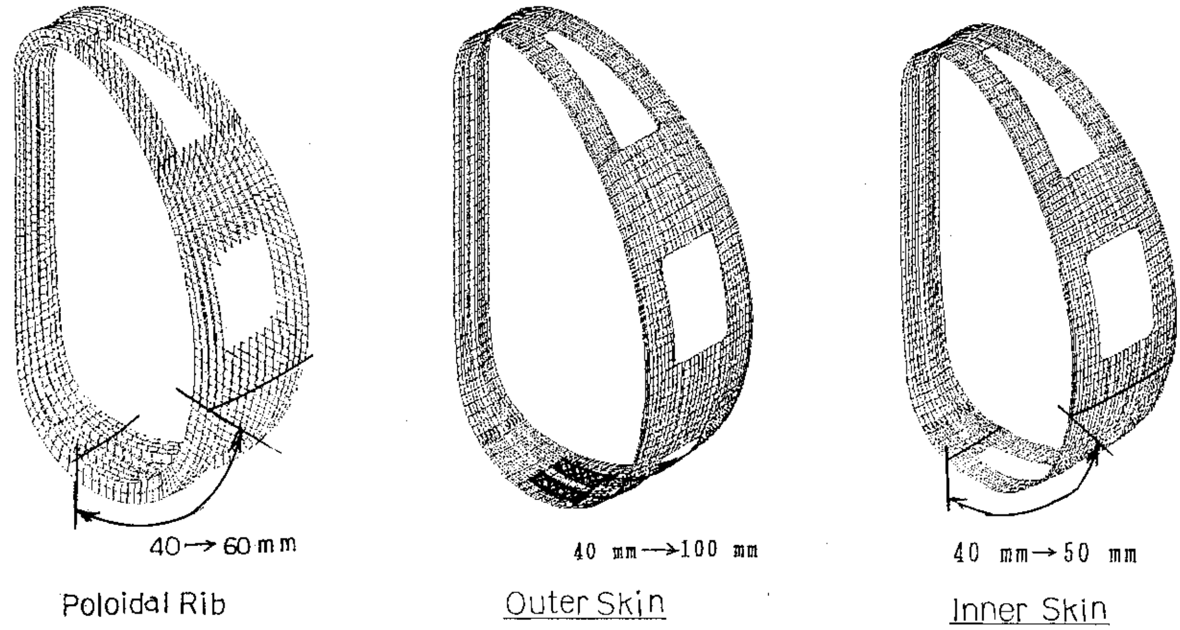


Fig.2-4 Modified Reinforcements on VV Structural Components

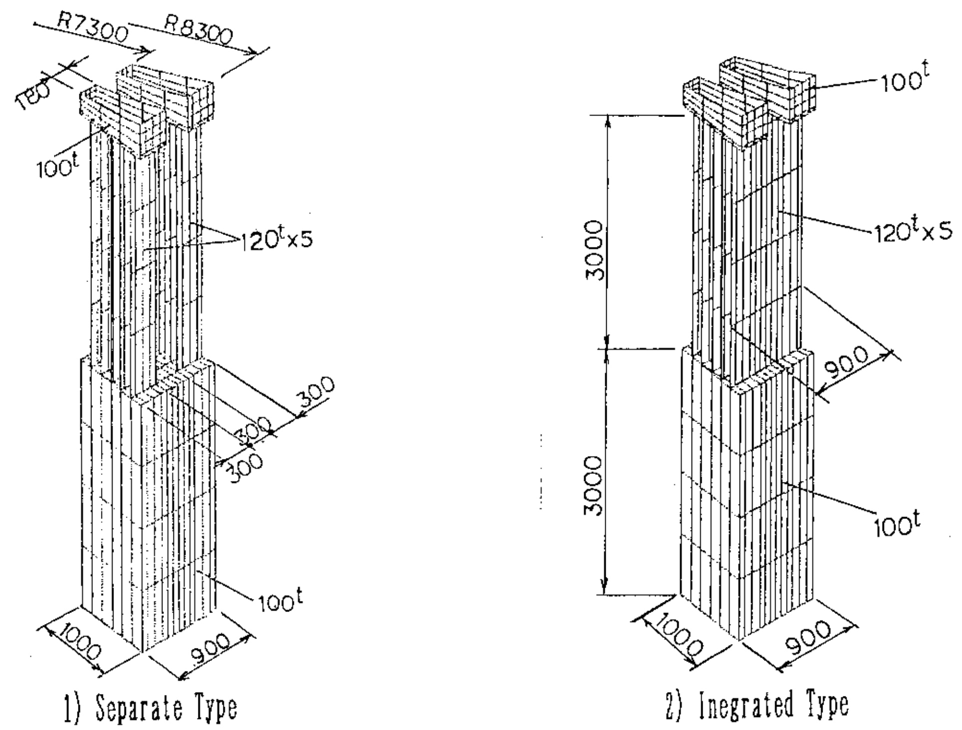
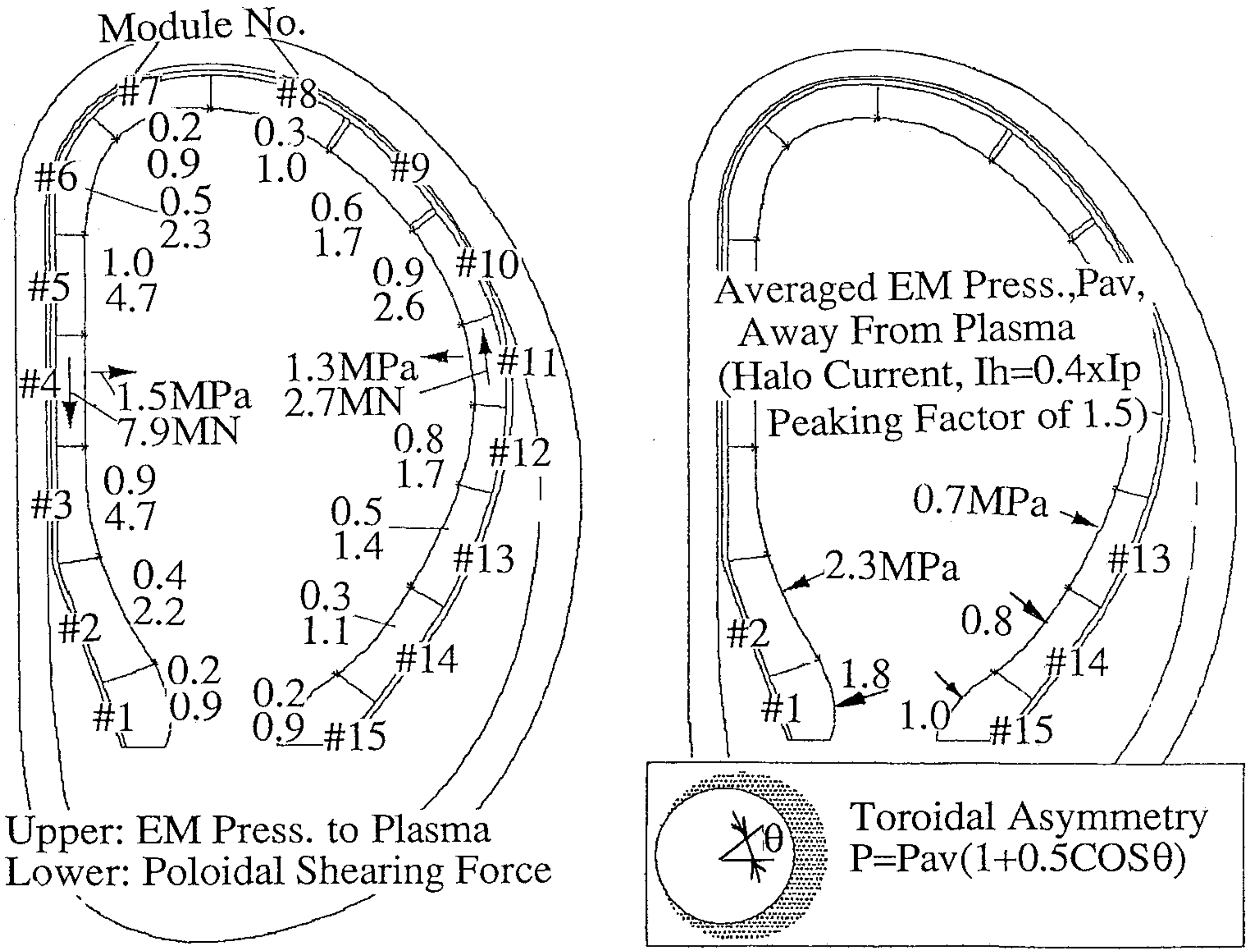


Fig.2-5 : Details of VV Support Leg,
with Multi Layered Flexible Plates of 3 m Length,
and Box-Type Rigid Support Leg of 3 m Length.



a) Centered Disruption Load

b) VDE Disruption Load

Fig. 2-6 Electromagnetic Forces on the Blanket Modules.

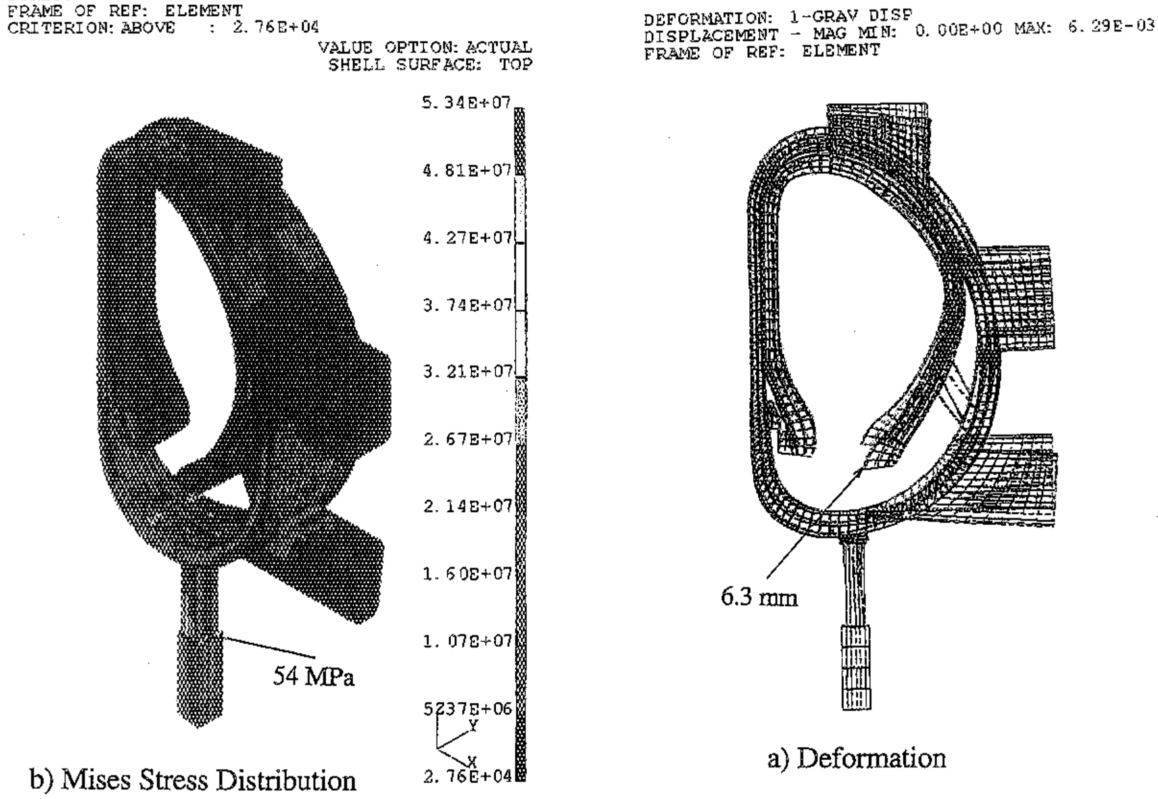


Fig. 2-7 Overall Deformations and Mises Stress Distribution on VV and Blanket under Dead weight.

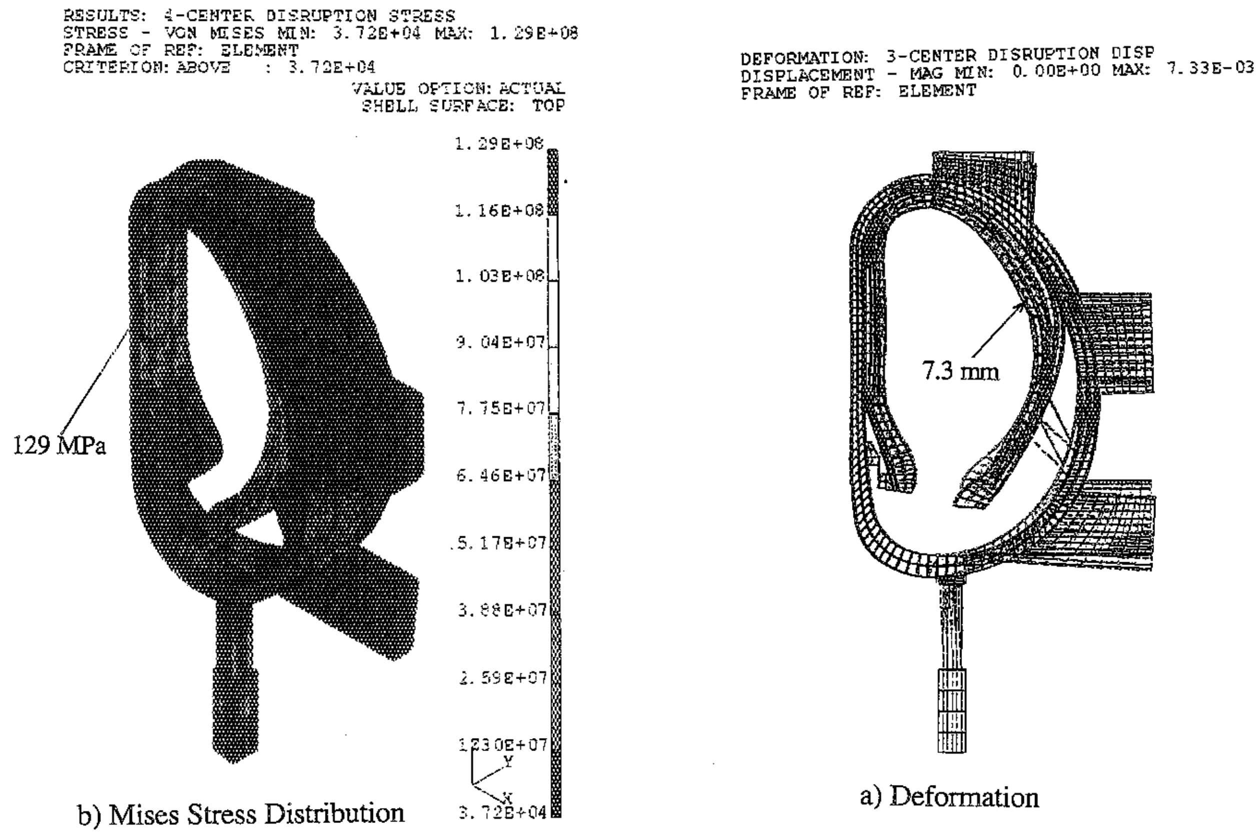
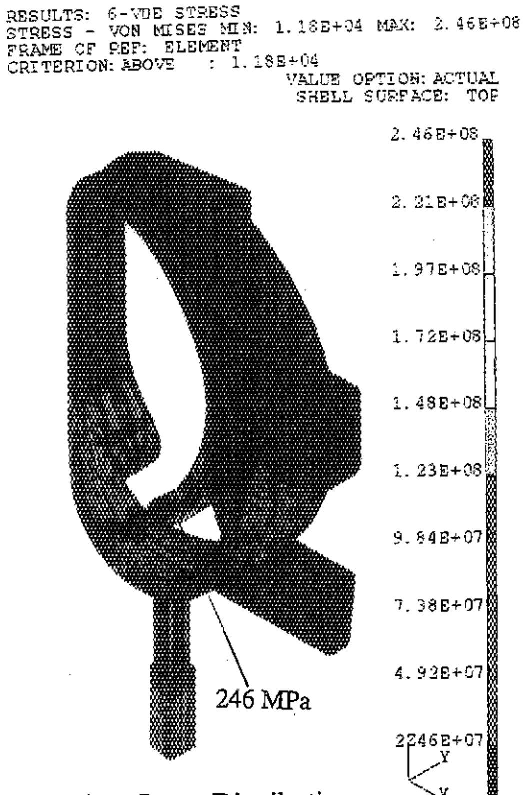
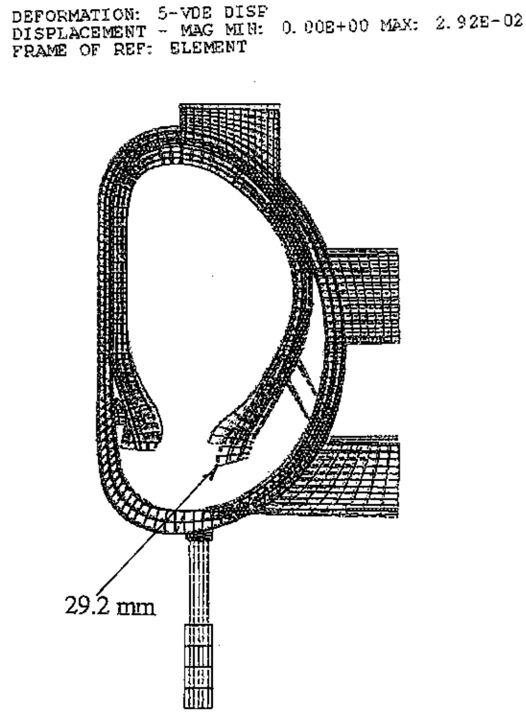


Fig. 2-8 Overall Deformations and Mises Stress Distribution on VV and Blanket under Centered Disruption Load.

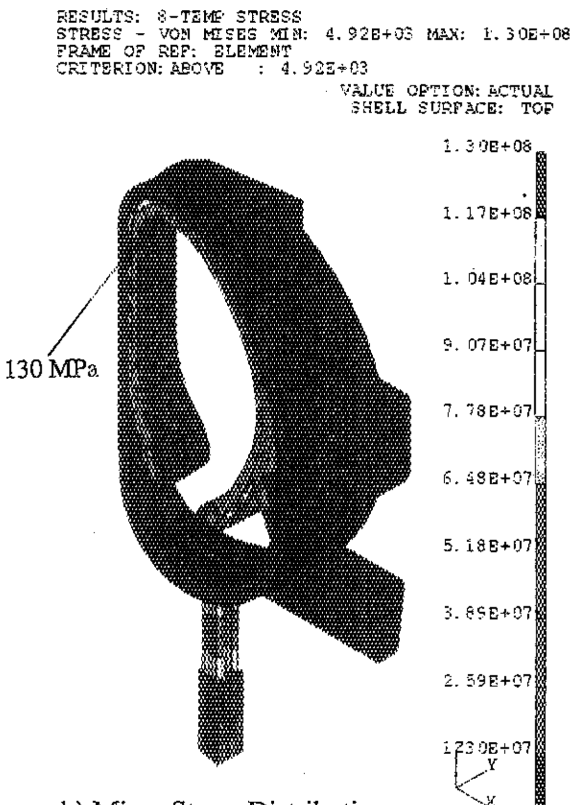


b) Mises Stress Distribution

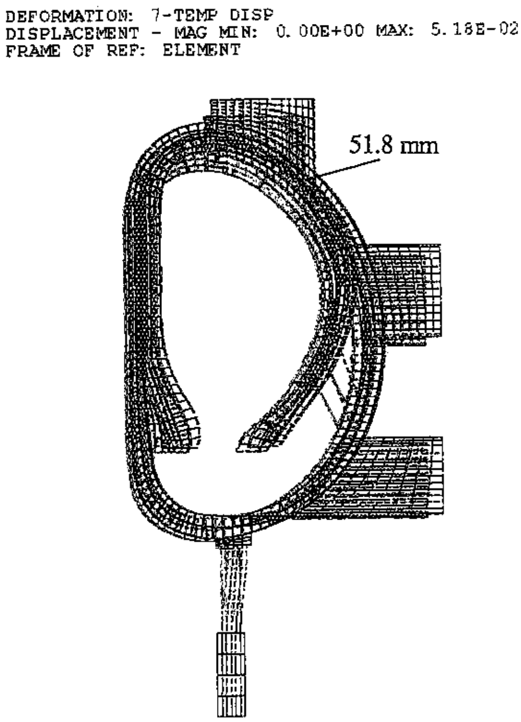


a) Deformation

Fig. 2-9 Overall Deformations and Mises Stress Distribution on VV and Blanket under VDE Disruption Load.



b) Mises Stress Distribution



a) Deformation

Fig. 2-10 Overall Deformations and Mises Stress Distribution on VV and Blanket under Thermal Load.

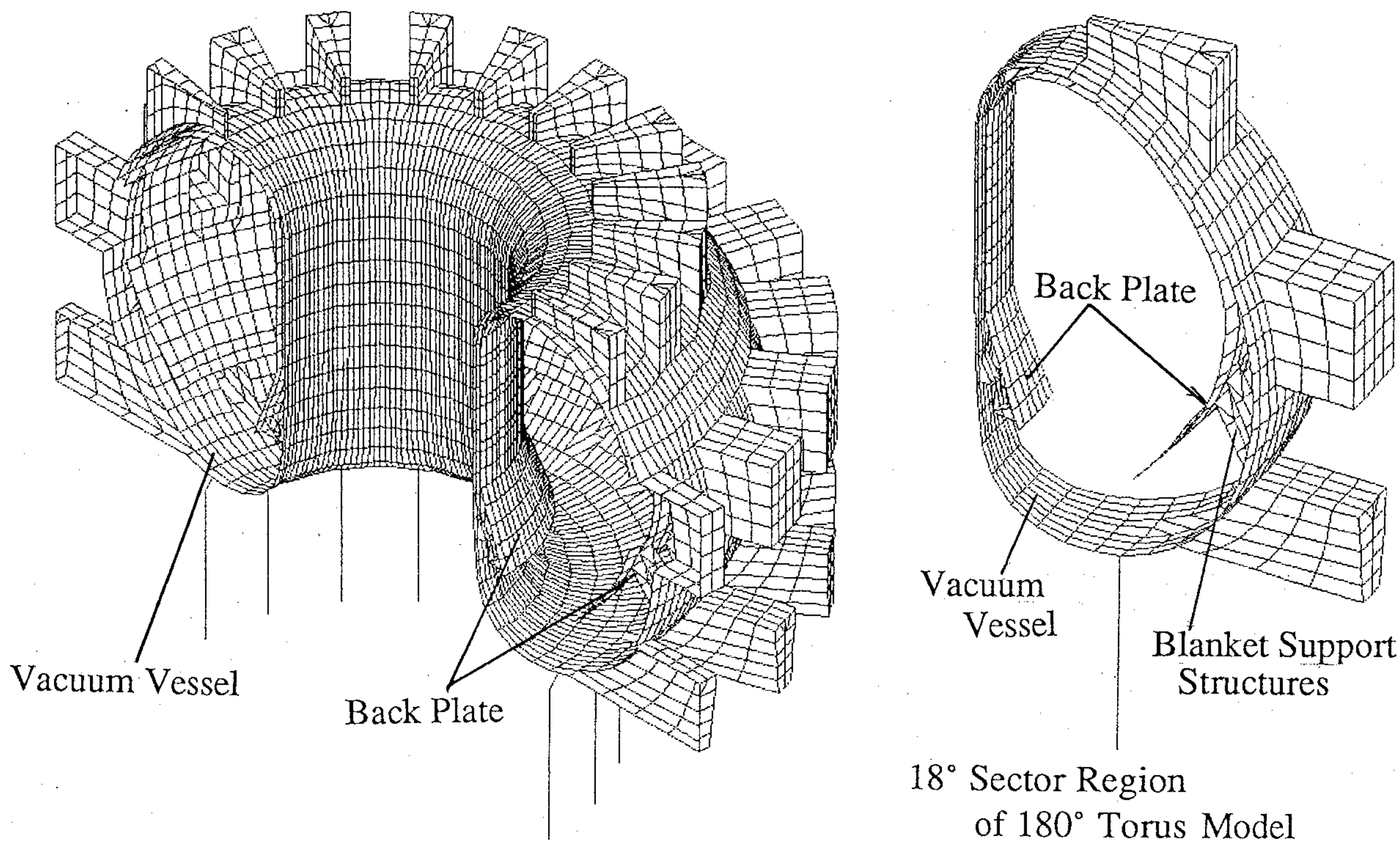


Fig.2-11A 180° Torus Model of Vacuum Vessel and Blanket System.

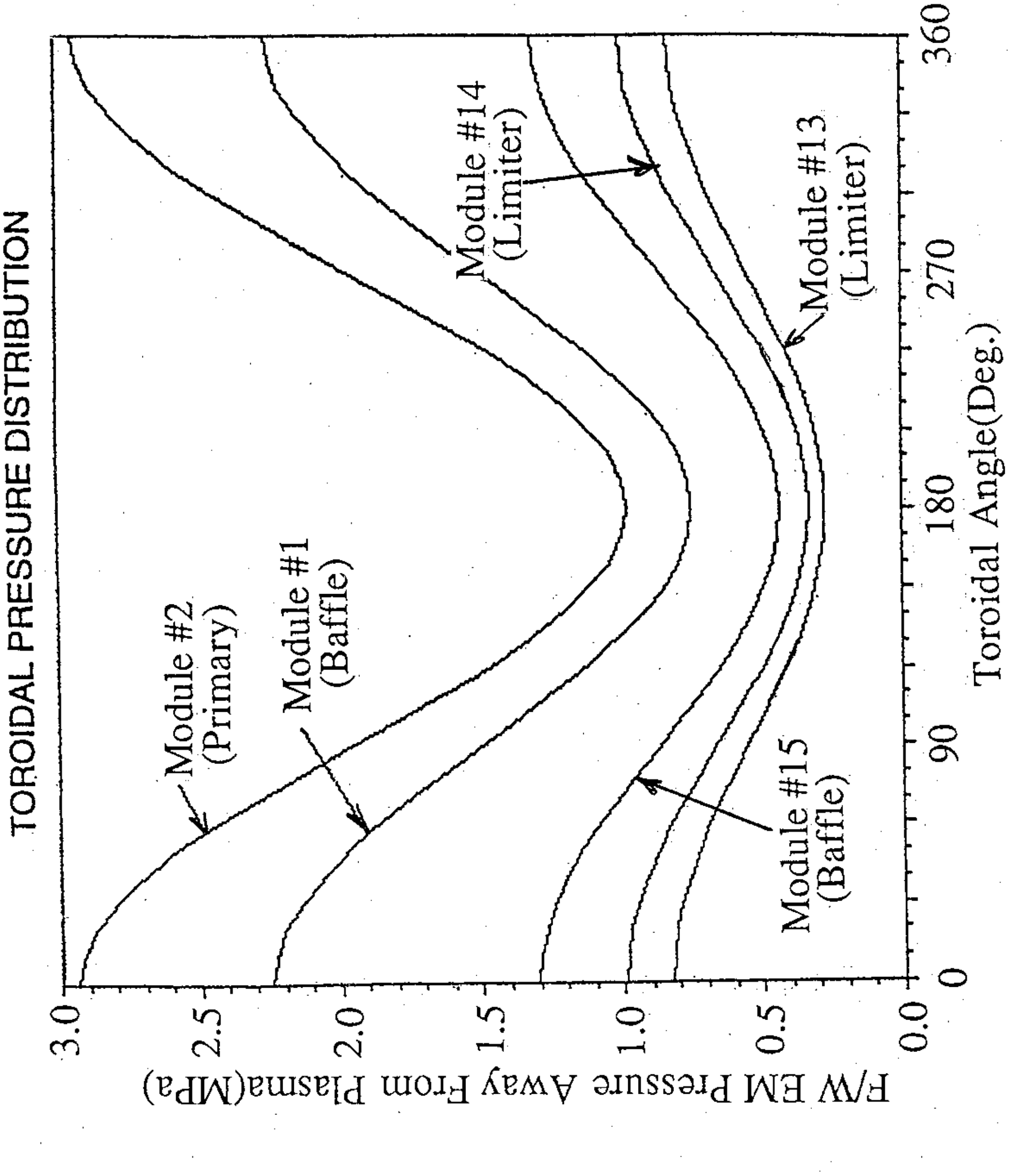
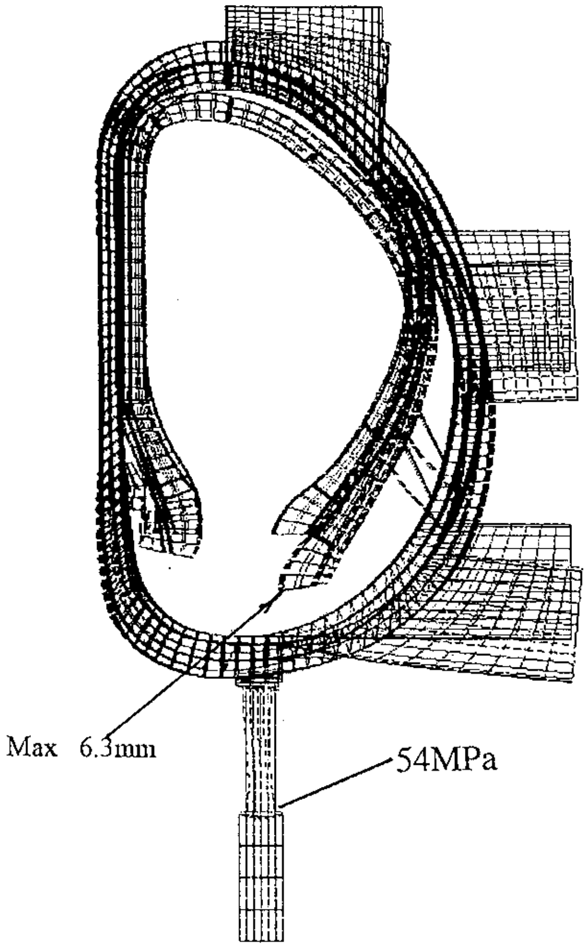
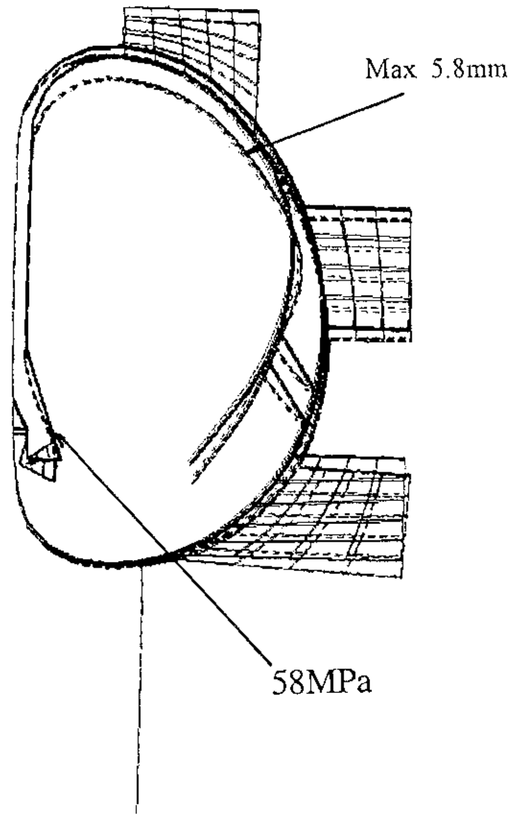


Fig.2-12 Toroidal Distribution of VDE Load on Blanket.

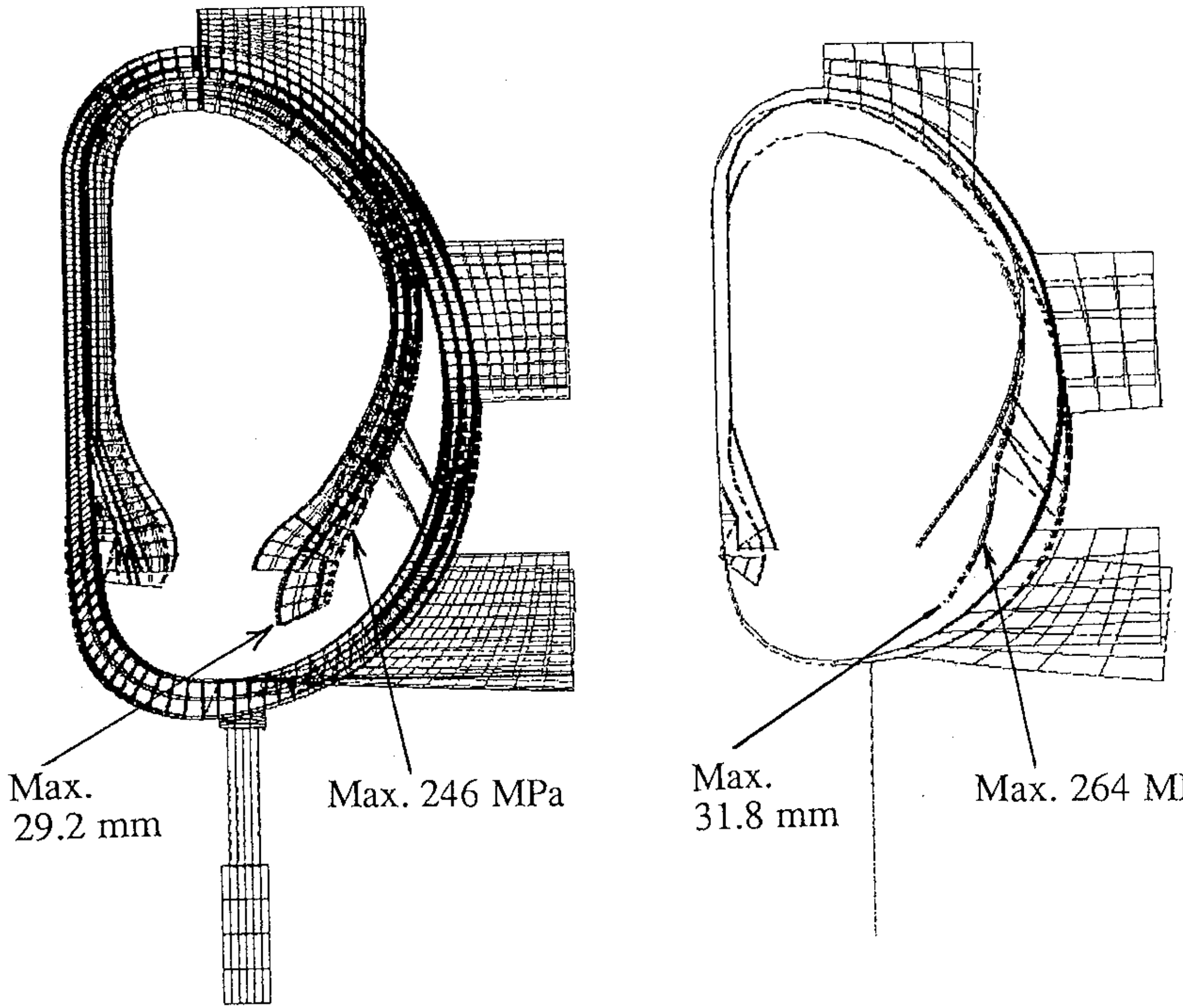


a) by 18° Sector Model



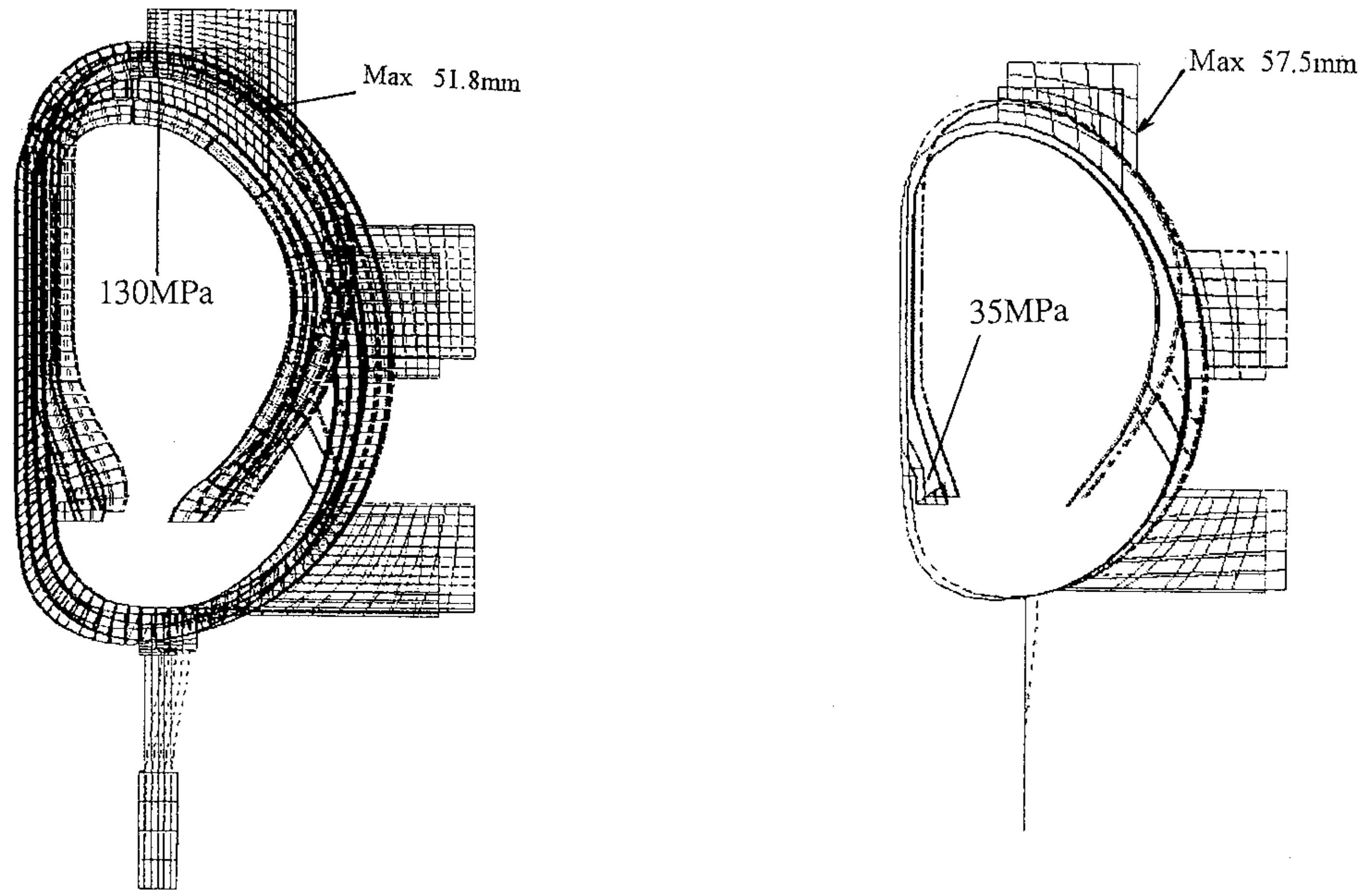
b) by 180° Torus Model

Fig.2-13 Comparison of Analytical Results by 18° Sector and 180° Torus Models due to Dead Weight.



a) by 18° Sector Model b) 180° Torus Model

Fig.2-14 Comparison of Analytical Results by 18° Sector and 180° Torus Models due to Symmetric VDE Load.



a) by 18° Sector Model

b) by 180° Torus Model

Fig.2-15 Comparison of Analytical Results by 18° Sector and 180° Torus Models due to Thermal Load.

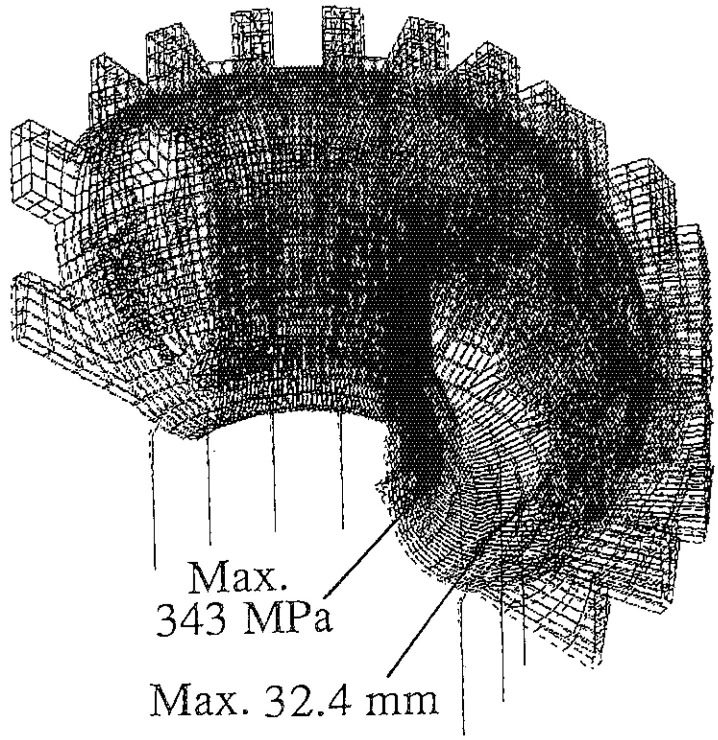
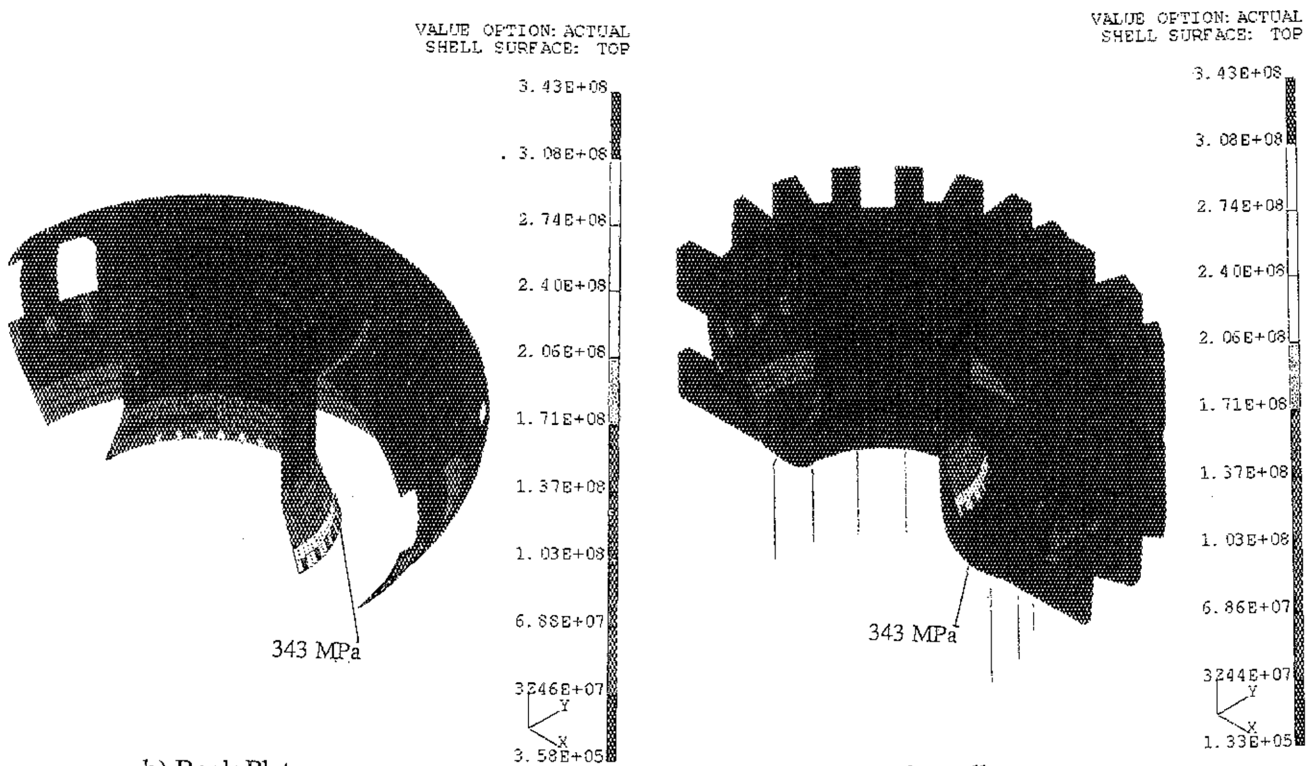


Fig.2-16 Overall Deformations of VV and Blanket due to Asymmetric VDE Load with 180° Torus Model.



b) Back Plate

a) Overall

Fig.2-17 Mises Stress Distribution on VV and Blanket due to Asymmetric VDE Load with 180° Torus Model.

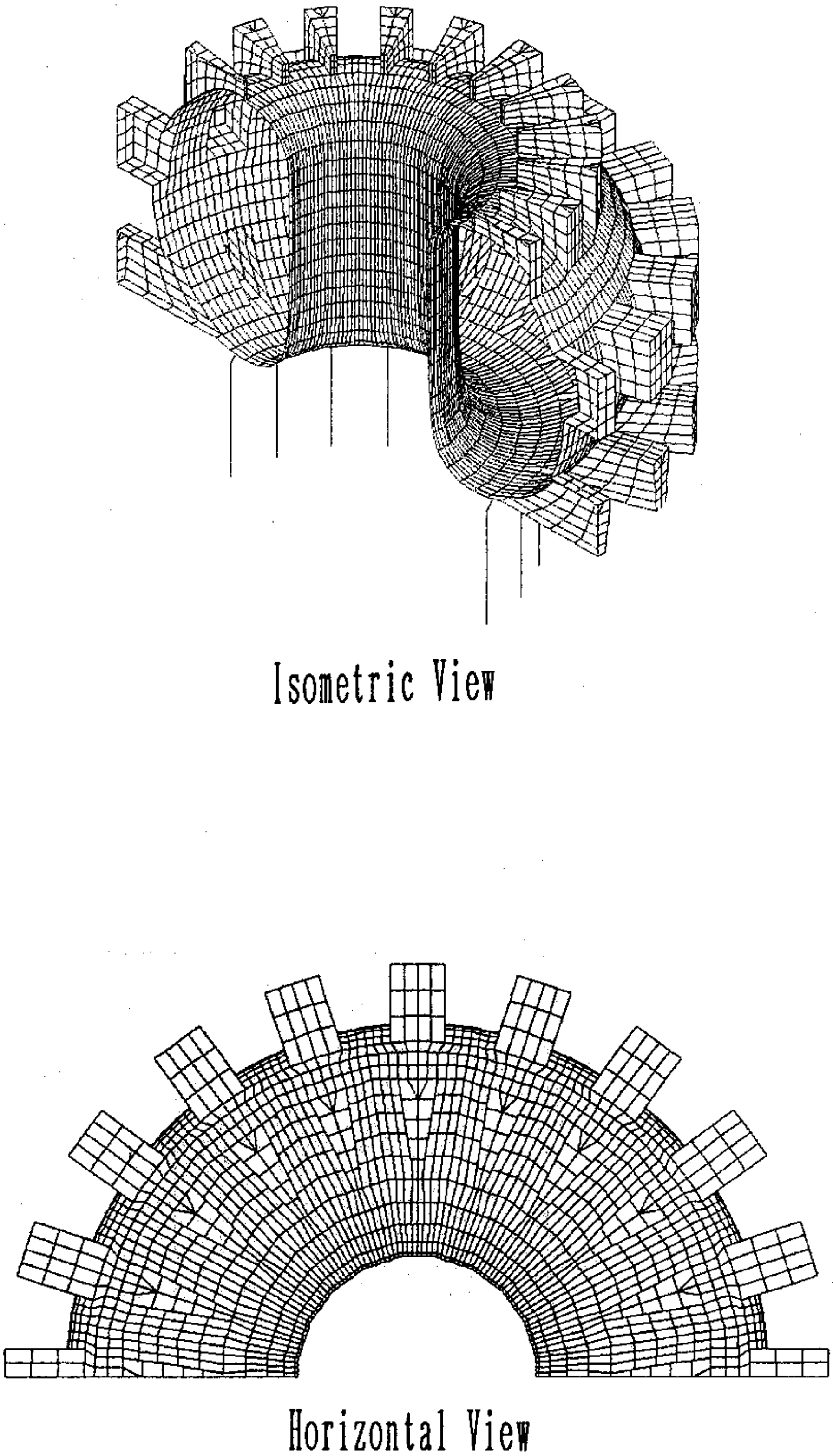
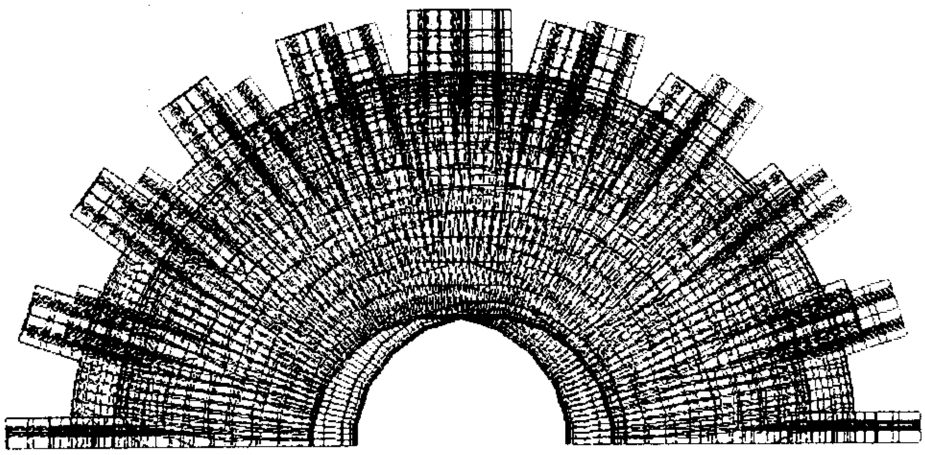


Fig. 3-1 A 180° Torus VV Model including Weight Loads of Back Plate and Blanket Modules for Natural Frequency Analysis

ITER 180 MODEL
DEFORMATION: 1-B.C. 0, MODE 1, DISPLACEMENT_1
MODE: 1 FREQ: 1.83264
DISPLACEMENT - MAG MIN: 0.00E+00 MAX: 5.51E-04
FRAME OF REF: PART



ITER 180 MODEL
DEFORMATION: 1-B.C. 0, MODE 1, DISPLACEMENT_1
MODE: 1 FREQ: 1.83264
DISPLACEMENT - MAG MIN: 0.00E+00 MAX: 5.51E-04
FRAME OF REF: PART

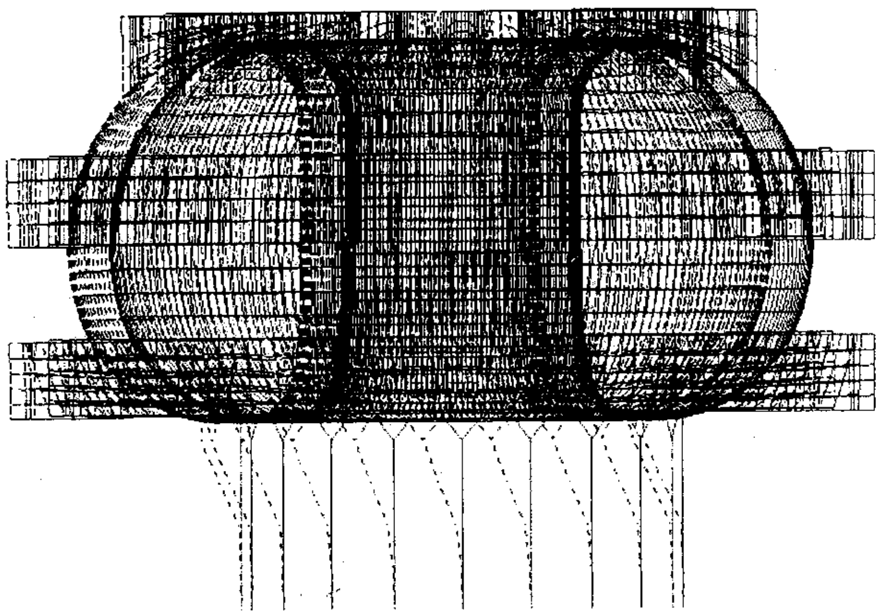
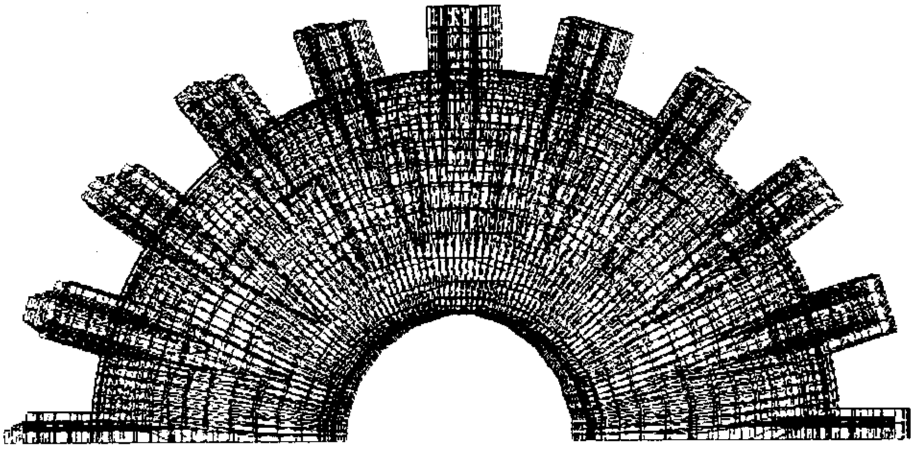


Fig. 3-2 1st Mode Natural Frequency of VV Torus Support Sysytem with Separate Type Structure of VV Gravity Support.

ITER 180 MODEL
DEFORMATION: 2-B.C. 0, MODE 2, DISPLACEMENT_2
MODE: 2 FREQ: 8.74445
DISPLACEMENT - MAG MIN: 0.00E+00 MAX: 1.17E-03
FRAME OF REF: PART



ITER 180 MODEL
DEFORMATION: 2-B.C. 0, MODE 2, DISPLACEMENT_2
MODE: 2 FREQ: 8.74445
DISPLACEMENT - MAG MIN: 0.00E+00 MAX: 1.17E-03
FRAME OF REF: PART

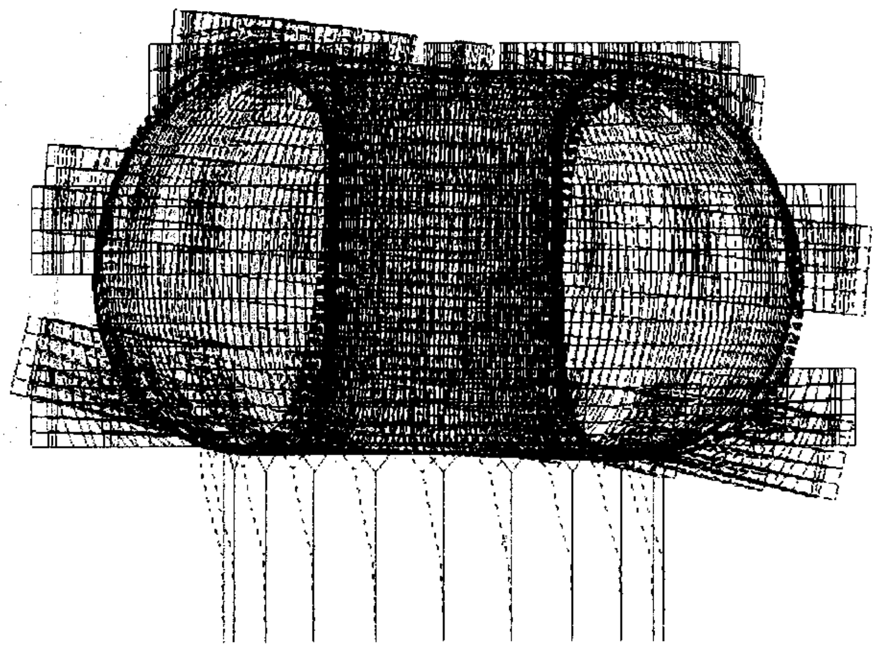
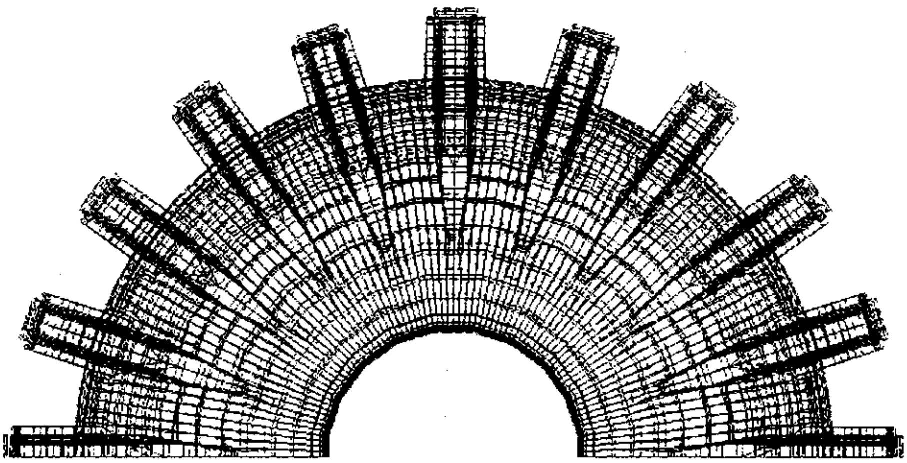


Fig.3-3 : 2nd Mode Natural Frequency of VV Torus Support Sysytem with Separate Type Structure of VV Gravity Support.

ITER 180 MODEL
DEFORMATION: 3-B.C. 0,MODE 3, DISPLACEMENT_3
MODE: 3 FREQ: 11.8579
DISPLACEMENT - MAG MIN: 0.00E+00 MAX: 9.53E-04
FRAME OF REF: PART



ITER 180 MODEL
DEFORMATION: 3-B.C. 0,MODE 3, DISPLACEMENT_3
MODE: 3 FREQ: 11.8579
DISPLACEMENT - MAG MIN: 0.00E+00 MAX: 9.53E-04
FRAME OF REF: PART

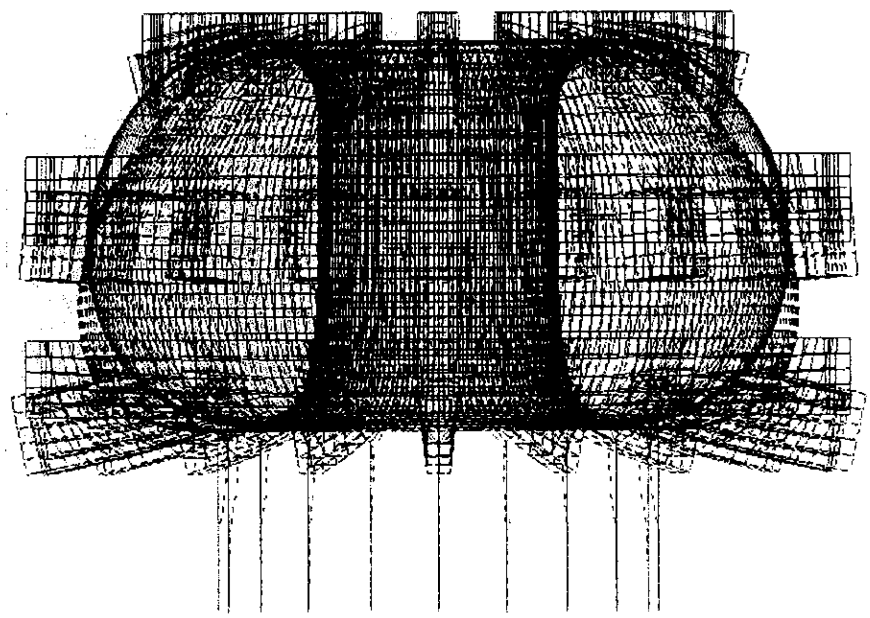
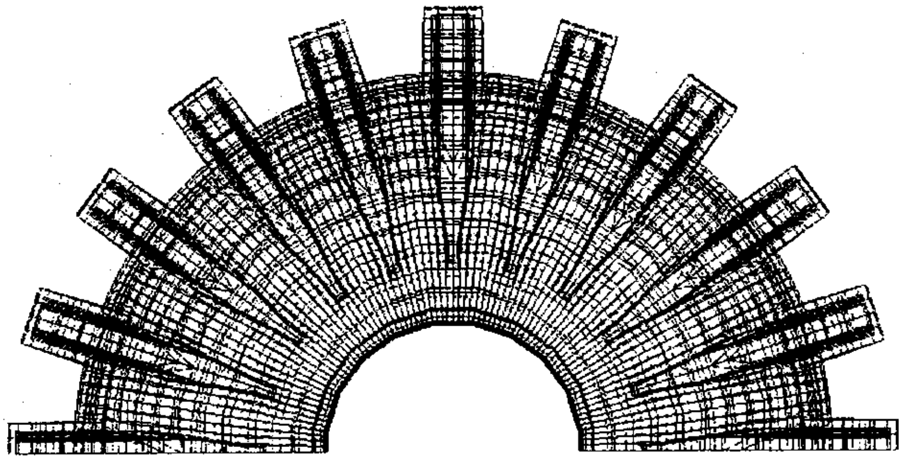


Fig. 3-4: 3rd Mode Natural Frequency of VV Torus Support Sysytem with Separate Type Structure of VV Gravity Support.

ITER 180 MODEL
DEFORMATION: 4-B.C. 0, MODE 4, DISPLACEMENT_4
MODE: 4 FREQ: 17.7049
DISPLACEMENT - MAG MIN: 0.00E+00 MAX: 1.11E-03
FRAME OF REF: PART



ITER 180 MODEL
DEFORMATION: 4-B.C. 0, MODE 4, DISPLACEMENT_4
MODE: 4 FREQ: 17.7049
DISPLACEMENT - MAG MIN: 0.00E+00 MAX: 1.11E-03
FRAME OF REF: PART

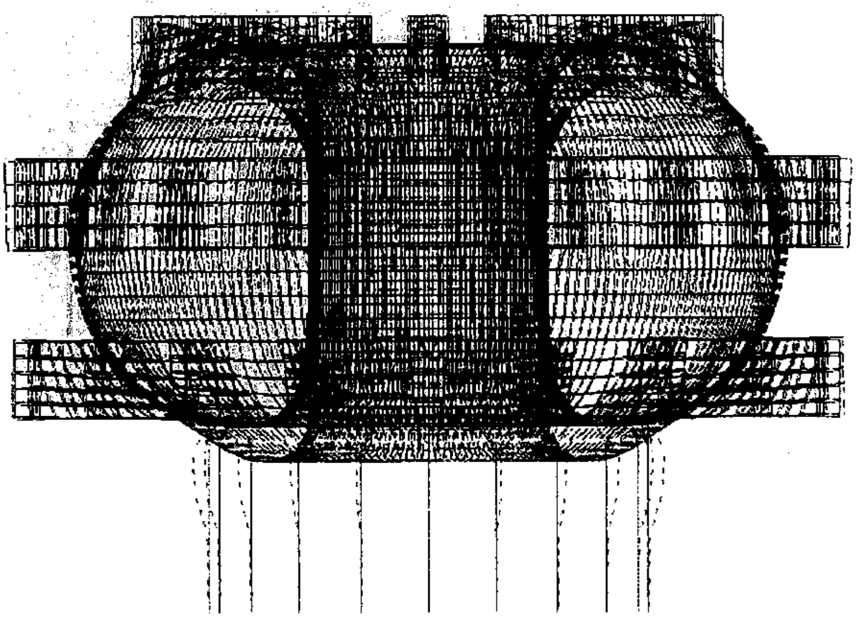
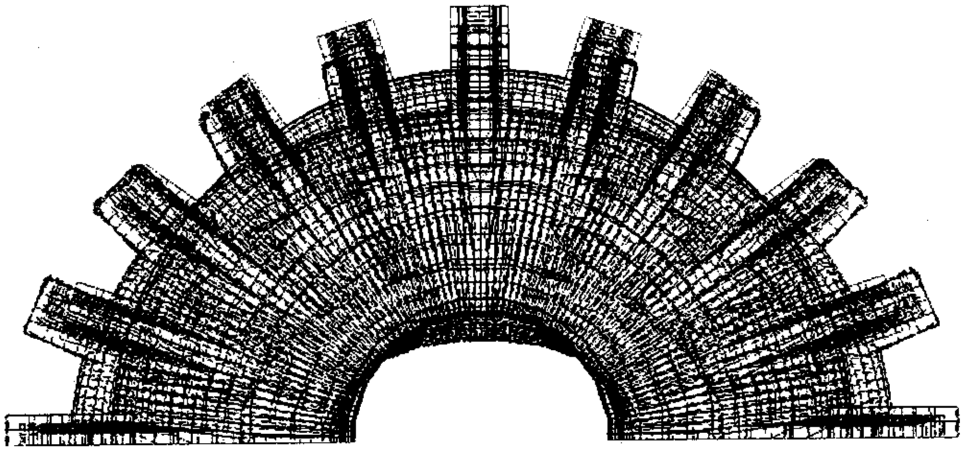


Fig.3-5 : 4th Mode Natural Frequency of VV Torus Support Sysytem
with Separate Type Structure of VV Gravity Support.

ITER 180 MODEL
DEFORMATION: 5-B.C. 0, MODE 5, DISPLACEMENT_5
MODE: 5 FREQ: 18.5316
DISPLACEMENT - MAG MIN: 0.00E+00 MAX: 1.09E-03
FRAME OF REF: PART



ITER 180 MODEL
DEFORMATION: 5-B.C. 0, MODE 5, DISPLACEMENT_5
MODE: 5 FREQ: 18.5316
DISPLACEMENT - MAG MIN: 0.00E+00 MAX: 1.09E-03
FRAME OF REF: PART

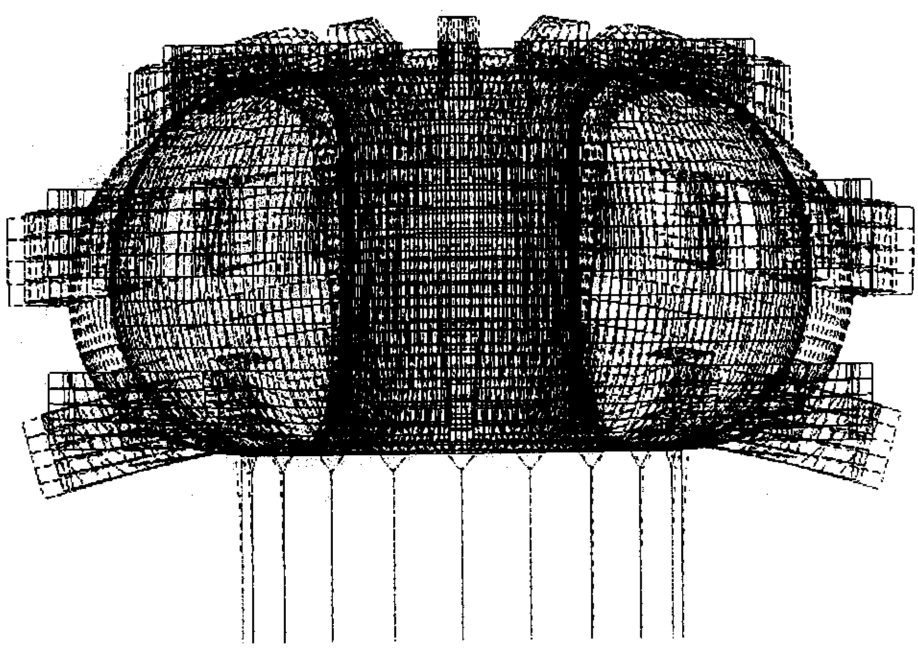
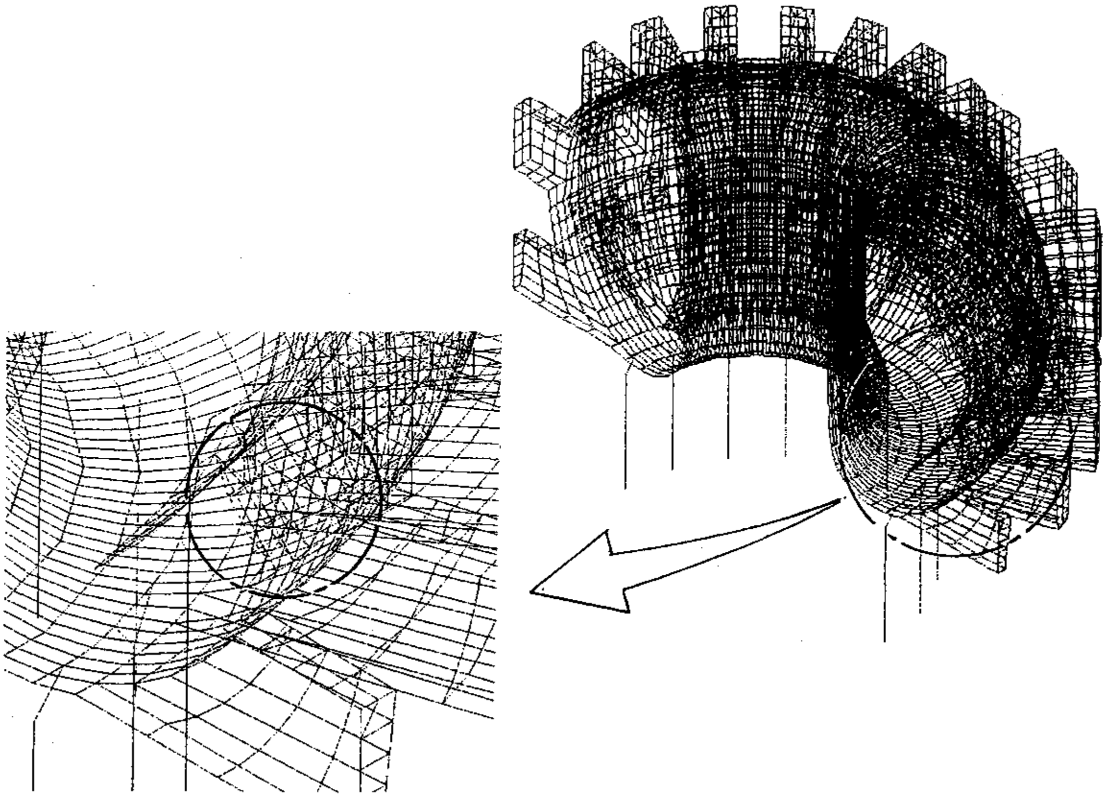


Fig.3-6: 5th Mode Natural Frequency of VV Torus Support Sysytem with Separate Type Structure of VV Gravity Support.



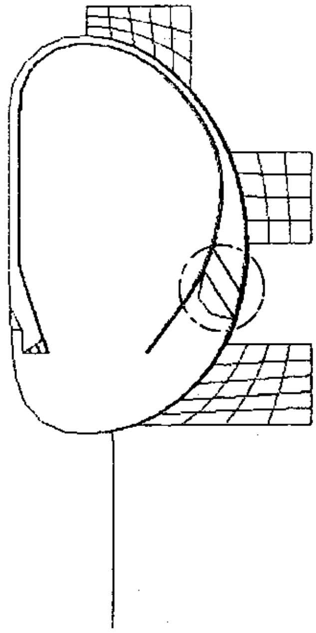


Fig. 3-7 : 1st Buckling Mode of VV Torus Support Sysytem with Separate Type Structure of VV Gravity Support.